MAE 4291: Senior Design Report

Steering System

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1. Introduction

The steering system steers the car. The coordinate system of our car has x pointing in the longitudinal axis towards the front; z is pointing up to the sky; and y is in the direction such that these coordinates form a right-handed coordinate system.

We wanted to change the architecture of the steering system to reduce play, improve assembly, and create a modicum of comfort for the driver. Last year's steering report [3] and wheel mount report [4] were used to determine aspects of last year's design that need to be improved. We explored various architectures as explained in Sections 3.3 and 3.4, and eventually chose the system shown in figure 1. Some of the reasons we decided on this is that it has good 90° planar angles for easier assembly, removes the need for a steering wheel to improve driver comfort and visibility, and is relatively cheap to manufacture.

Figure 1**. Overall Steering System**

With this architecture, we chose the lengths of links (such as the pitman arm, steering column, tie rods, and ackerman lever) to satisfy Ackerman steering by running a MATLAB simulation to solve a system of equations, explained in detail in Section 3.6, describing the dynamics of the car. The inputs to these were some pre-set geometries such as track and wheelbase that were influenced by the chassis architecture, driver height, driver position, and length of the rear compartment. From these results, we compiled table 7 of Section 5 with the final design specifications.

With the geometries decided, we then looked at a tolerance stackup in detail in Section 3.7 and a loaded deflection analysis in Section 3.9. We added the play that would result from manufacturing and other tolerances between component interfaces, and found that the pitman arm was causing 2.44 degrees out of the 3.36 degrees total system play. To improve this, we changed the mounting of the pitman arm on the steering column to use a tapered pin and this reduced reduced the expected play by 73%. We used ANSYS to simulate torsional, bending, and other relevant load cases on each part and predicted the system deflection by summing up the deflection of each component. Our tolerance and deflection specifications were somewhat of a guess this year, because this was the first year that our team had looked at those parameters quantitatively. We believe that these specifications will provide a very helpful baseline for future years as they continue to improve the system and rely on increasingly quantitative data.

We then built a prototype of the steering system to verify these tolerance and deflection specifications and the motion ratio. The prototype is composed of 1:1 real parts that can be used on the car, aside from the wheel mounts, whose geometry we changed slightly after the prototyping. We verified that the no load play in the custom machines parts is unnoticeable for the driver but that the gearbox has 2.3 degrees of play, which is the most noticeable contribution to play. We also verified that our motion ratios are what we pre-calculated for Ackerman steering.

Next year, the steering system will continue manufacturing with some parts like the pitman arm needing only weight reduction, whereas other parts like the wheel mounts and bearing mounts need a complete re-machine. In addition, the steering system needs to finalize the design for mechanical stops to prevent the wheels from hitting the chassis.

2. Design Requirements

Our design requirements have been divided into two categories: external and internal. The external requirements refer to those set by the rules of our competition. These are non-negotiable and cannot be modified. Internal requirements are set by our team and include both interfacing requirements and performance goals.

2.1. External Requirements (from the rules of the competition)

The sections of the 2019 Shell Eco-Marathon rule book which pertain to the steering system requirements of our vehicle are Articles 39 and 42. These are included in appendix 1 in their full text in and are summarized in table 1 below. Adherence to each of these external requirements will be analyzed in the technical inspection which occurs prior to competition. While some of these requirements are very quantitative and are measured directly (wheel base, track, turning radius) others are far more subjective and are evaluated based on the judgement of our inspector (visibility, play). These are far more difficult to prepare for and our interpretation of them is based heavily off of experiences from previous years.

Table 1**. External Requirements**

Parameter	**Requirement**
Wheel Base	$L \geq 1.00$ m
Track	$t \geq 0.50$ m
Turning Radius (Measured at Outer Wheel)	$R \leq 8.00$ m
The driver must have two hands on the steering system at all times.	
Horizontal Visibility	90° to each side, small blind spots permissible
Play	minimal

2.2. Internal Requirements (set by our team)

Internally, our team's additional requirements are founded on the bases of safety, minimizing rolling resistance, interfacing well with other systems of the car, basic ergonomics, feasibility, and the ability to interface with an autonomy system in the future.

While the rules specify a minimum track which is likely in place to prevent cars from rolling over when cornering, our team decided to calculate that risk directly and impose a required safety factor against rollover. The rules are also designed to maximize safety, not performance. In order to improve our performance in this efficiency-based competition, we also wanted to define the geometric components of rolling resistance. This led to the requirement of a maximum Ackermann factor and of zero camber.

As a team, we also set requirements for aspects of the design that would impact play, since that rule is ill defined. Our reasoning for this was that if our internal requirements passed the technical inspection and felt acceptable to the driver, our team would have a more quantitative and better defined requirement for play in the steering system in future years. This was broken down into the unloaded play due to joints and tolerance stackup and the loaded play due to deformation of the system under expected steering torques. While we analyzed our system to make sure it would not reach critical levels of stress (see Section 3.3), we found that we hit our tolerance for deflection much earlier than we hit any critically high stress values.

We also wanted to improve significantly on the ergonomics of the vehicle by ensuring from the start that the steering system would in no way collide with the driver's body when in use (This basic requirement has not been met in the previous two years on our team). We also found that the previous year's mechanical advantage in the steering system (i_s = 1.56) was sufficient for the driver to manipulate the system, and wanted to approximate it again for this year's vehicle.

Our final internal requirement was to make the car capable of much sharper turns than required for our competition. This was motivated largely by the fact that the team often moves the car around inside Upson basement, in other tight spaces on campus, and in tight spaces at our competition and the ability to make tight turns will make the car more maneuverable for those instances.

In future years, one major goal for our team is to go autonomous. This means that we must have software which can sense what is happening and plan a good trajectory, electronics which can gather that sense data and implement the trajectory, and mechanical systems that can accept the electrical input. While we will not have all of those things done in the next year, we have also imposed the requirement that the steering system designed *could* be controlled autonomously by an electronic system. This is something that we believe will benefit our electrical and software teams as they develop and test their autonomy-related projects.

Table 2. **Internal Requirements**

Parameter	**Requirement**
Rollover risk	η (factor of safety) ≤ 1.50
Camber	0°

Ackermann Factor	0.980
Mechanical Advantage	1.065 (this was the final result, we aimed for anything between 1.00 and 2.00)
Turning Radius (Measured at Outer Wheel)	R = 5.00 m
Tolerance Stackup Specification	0.930 degrees (not met because of gearbox)
Angular Deflection Specification	0.934 degrees
System needs to be able to accept input from an electromechanical system.	

3. Support for Design Decisions

3.1 Overall Vehicle and Wheel Mount Geometry

Because of the schedule regarding freshman recruitment (Wheel mounts were a freshman project this year), we took the early work of the wheel mounts project which included research and planning for the geometric requirements of the wheel mounts.

There are no strict requirements for most of these values (requirements for track and wheelbase are set out in the rules, though), so most of them were determined based on our goal of low rolling resistance and some idea of optimal handling and steering stability. The final values of these parameters and the reasoning behind them can be found in table 7 in Section 5.

3.2 Overall System Structure

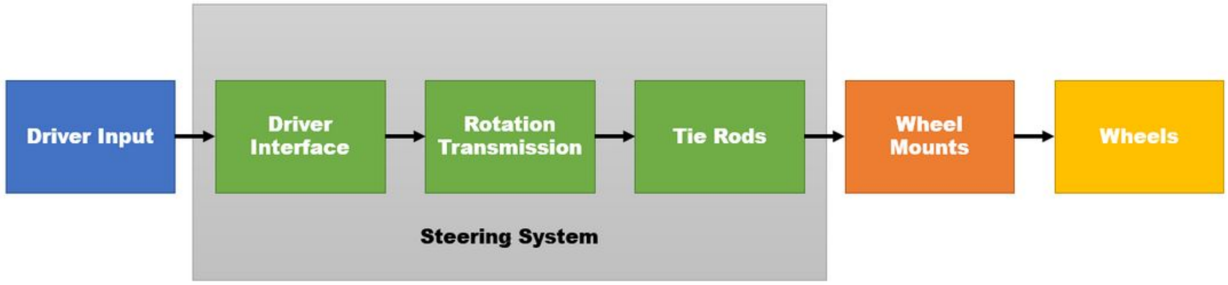


Figure 2**. System Structure**

The steering system has three main subsystems: the driver interface, the rotation transmission, and the tie rods (figure 2). The driver interface is how the driver provides input to the steering system, the rotation transmission moves that rotation input to the tie rods, and the tie rods are how the steering system interfaces with and manipulates the wheel mounts. Both the driver interface and the rotation transmission require their own overall design and need to be compatible with each other. Not pictured above is the autonomous steering input, which will not be in place this year, but we would still like to accomodate for in hopes of future development.

3.3 Driver Interface

3.3.1 Requirements of driver interface

- Two-handed system
- Visibility needs to have only small blind spots

- Needs to be able to mount throttle and horn

3.3.2 Steering Wheel

Figure 3. **Steering Wheel From Last Year Takes Up Space and Blocks Some Visibility**

For both CRR18 and the Kiwi Kruiser, we used a steering wheel which was placed above the driver's pelvis and supported by a cantilevered, angled steering column (figure 3). While this is a viable and traditional method, it has two key issues that make it ill suited to our vehicle. These are the cantilevered steering column and the position of the steering wheel.

When used with a steering wheel, the steering column must be angled to reach from the floor where the rest of the steering system is mounted to the steering wheel above the driver. This means that it is essentially a long cantilevered column which introduces a lot of play and wiggle. Both years that thus was used there was an additional component which was made to support the column. If this component is placed halfway along the shaft (Kiwi Kruiser), it is not particularly effective in reducing the play. If this component is closer to the steering wheel (CRR18) it is only slightly more effective and highly uncomfortable for the driver, as it is essentially in their crotch.

Over the past two years, steering wheel placement has been tricky. The x and y positions of the steering wheel are set by the length of the driver's arms and the position of the driver within the car. The z position must be high enough that the steering wheel does not intersect the driver, but low enough that it does not impede visibility. Over the past two years, we have found that there is no good z-position that meets both of these requirements.

Pros

- We have done it before and it works
- Relatively cheap

Cons

- No good position for steering wheel
  - Will either have bad visibility or *very* bad ergonomics
- Play from steering column
- Uncomfortable for driver

3.3.3. Under the Butt System

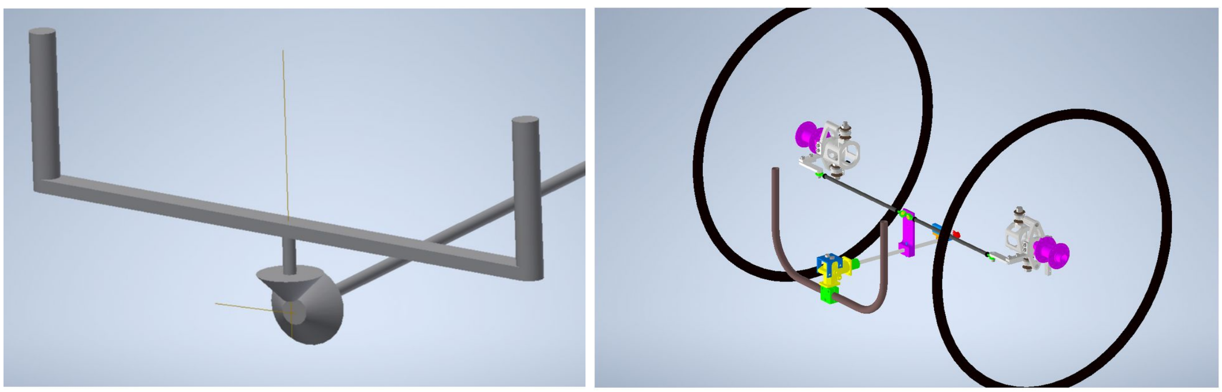


Figure 4**. Under The Butt Steering System**

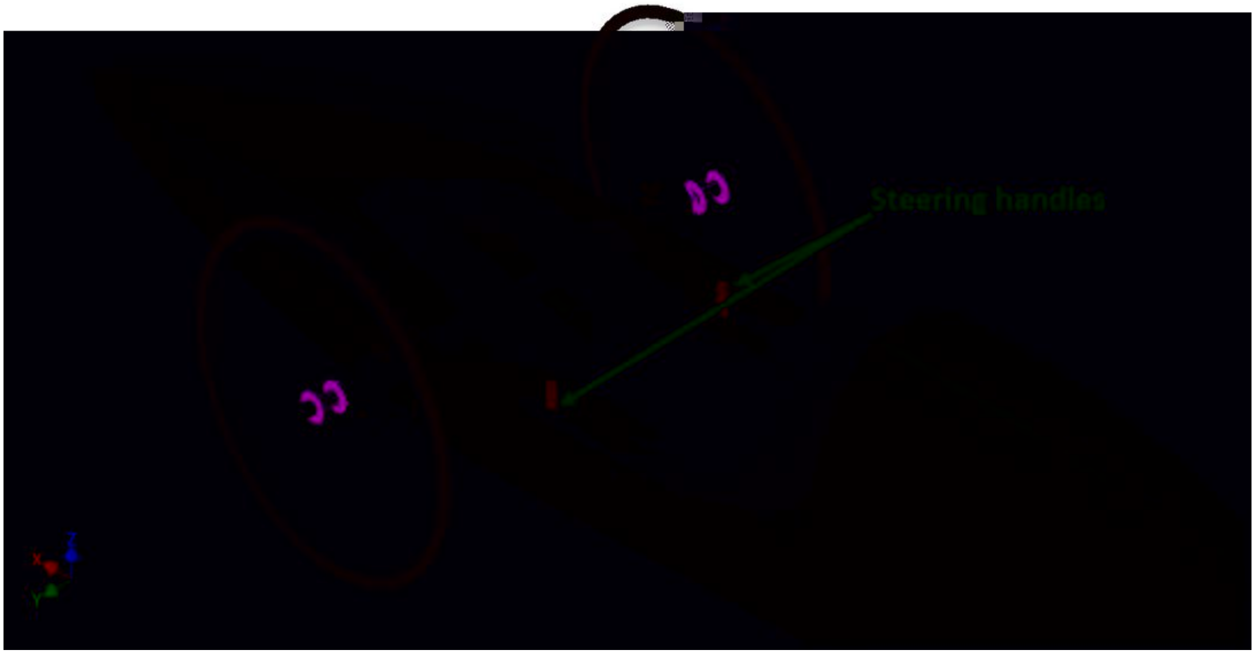


Figure. Under The Butt Steering System

This system relies on a gearbox which allows torque and rotation transmission to turn a 90° corner (this is explained in the terminology section). This will likely be done by an off-the-shelf purchased gearbox which can be expensive. The U-shaped handle rotates about the gearbox underneath the driver's butt. From the driver's perspective, the two vertical segments of the handle move towards and away from their feet.

The fact that the steering column is horizontal and runs underneath the driver, means that it can be supported on both ends instead of being cantilevered. This does, however, also mean that we need space underneath the seat which adds complexity to the chassis, but it can be done.

Doing this allows us to move the driver interface from being on top of the driver (as it would be with a steering wheel) to the sides of the driver which eliminates visibility concerns.

Since this option only has the one gearbox joint, attachment to either a rack and pinion or Pitman arm would lead to an unintuitive steering direction. Pushing the right handle forwards and pulling the left one back would make the car steer to the right. While this will require somewhat of a learning curve, humans are usually pretty good at adjusting to strange user interfaces, so it may not be an issue.

Pros

- Good ergonomics

- Having one steering column makes automation easier
- Horizontal steering column can be easily supported at both ends

Cons

- Requires a seat

- Intuitive direction

Cons

- Requires a seat
- Good gearboxes are expensive
- Lots of joints leads to lots of play

3.4 Rotation Transmission

3.4.1 Requirements of transmission system

- Minimal play
- Motion of steering system should not collide with any part of the driver's body
- Ability to control with either motor or linear actuator for autonomy

3.4.2 Rack and Pinion

Figure 6**. Rack and Pinion System**

In theory, a rack and pinion system is an ideal choice. It provides linear mapping between the input and output of the steering system and is a self contained unit which reduces our ability to introduce errors in manufacturing. However, there are a few key points that make a rack and pinion system unfit for our vehicle.

The most important of these points is that the rack and pinion system requires precise mounting in order to work effectively. Due to our team's inexperience with carbon fiber manufacturing techniques and our inaccurate manufacturing results in the previous year, we don't want to rely on pieces of the chassis being exactly where they are expected to be.

We also operate on a limited budget and quality rack and pinion system prices start at around $650. The rack and pinion system also does not address some of the other requirements of the steering system (ergonomics and long cantilevered steering columns), meaning that it only represents a portion of the budget for the system as a whole.

Pros

- Tried and true system--this is what is used in most cars
- Having one steering column makes automation easier
- Linear relationship between input and output

Cons

- Requires very precise mounting
- A decent rack and pinion system is very expensive
  - If we want a quality one the cheapest we could find was $650

3.4.3 Pitman Arm

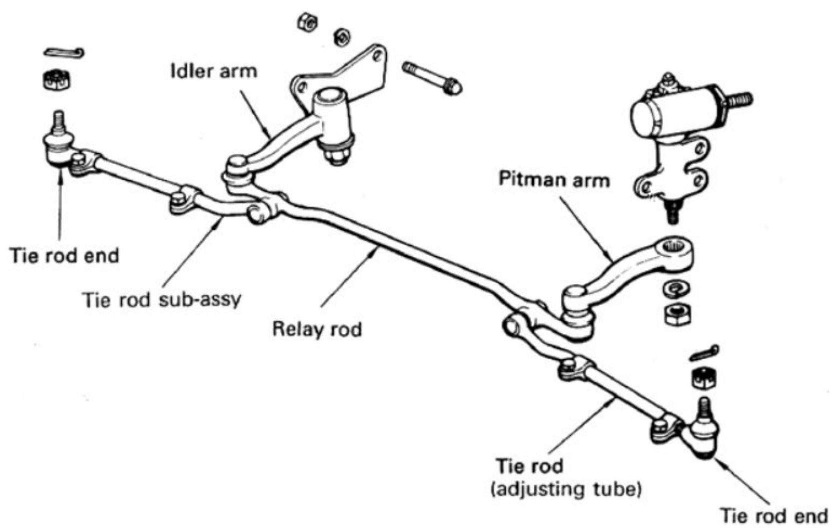


Figure 7**. Pitman Arm Concept in Traditional Vehicles**

This diagram shows the implementation of a Pitman arm in a traditional car. Since our car is simpler and smaller, we don't need to use an idler arm or a relay rod.

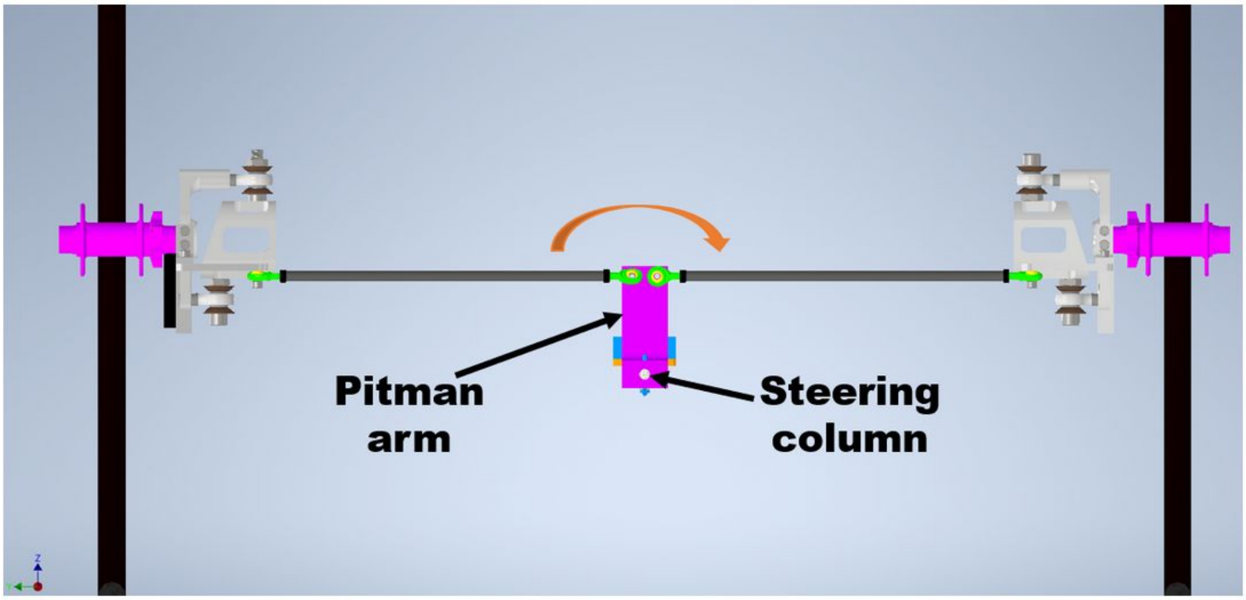


Figure 8**. Pitman Arm in Our Vehicle**

This shows how the pitman arm works in the Cuckoo Caravan design.

A Pitman arm is a potential alternative to a rack and pinion system. It performs essentially the same function: turning the rotation of the steering column into translation of the tie rods. It is much more forgiving of misaligned mounting than a rack and pinion system, but will always contain some nonlinearity due to the fact that the tie rods do not translate strictly along their axes. Instead of strict translation, one end rotates circularly about the steering column which introduces a sine error.

We have done more analysis of the implications of this nonlinearity in Section 3.6.

Pros

- Easy to implement--we've done it before
- Toe-in/out adjustability in tie rods
- Tolerant of inaccurate mounting or chassis manufacturing

Cons

- Nonlinearity in implementation

    - Sine error is small for small angles
- Multiple joints lead to more play

3.4.4 Direct Manipulation

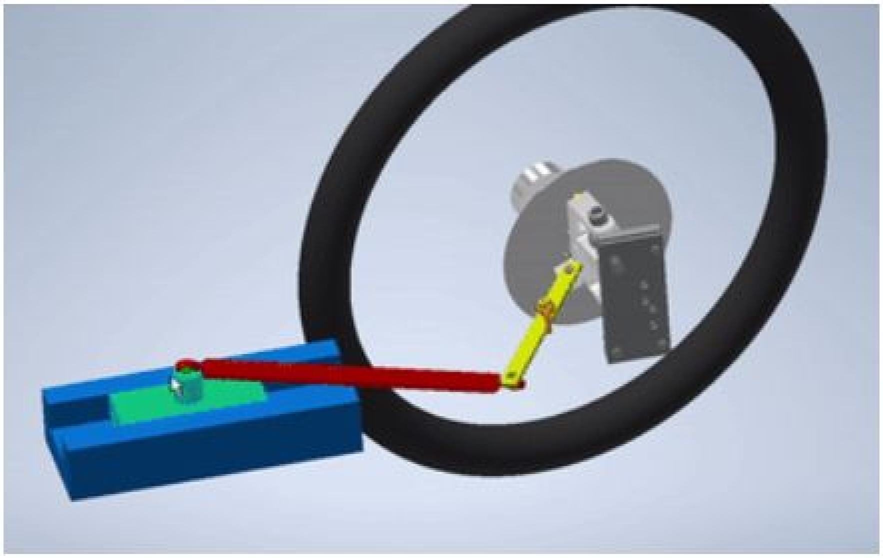


Figure 9**. Direct Manipulation Steering System Architecture**

Another option we considered is to have a more direct manipulation of the wheel mounts. This would consist of two independent linkages for the left and right sides of the car, coupled by one long tie rod. This long tie rod would attach where the yellow and red members meet in the animation above.

While this system is nice in theory because it contains very few joints, it has several issues that will likely occur in implementation. One of the main issues is that separate mounting of the left and right sides is inviting binding and play because of highly-likely inaccuracies in chassis manufacturing. Another issue is that good linear rails are expensive and that buying good linear rails and then mounting them poorly completely defeats the point.

Another important note about this design is that it would be very difficult to integrate with autonomy. We would either need to have two driving systems which provided exactly the same input or only have one side of the system be driven, which would make it very hard to control and very likely to bind.

Pros

- Very few joints

- Fairly simple geometry calculations

Cons

- Requires precise mounting
- Potentially very expensive
- Manufacturing inaccuracies create "lopsidedness" between left and right wheels
- Very difficult to interface with autonomy.

3.5 Autonomous Integration

The complex aspect of autonomous integration is that it must have a mechanical override built into it. This is specified by the rules that Shell lays out and is a safety precaution that we would want to include regardless. For the sake of a functioning, closed loop control system, we would also need to include a sensor which can give the electrical system a reading for current angle of the steering column. This will be a potentiometer regardless of how the mechanical override is designed.

3.5.1 Requirements of autonomous integration

- Easy to operate mechanical override
- Does not impede other systems
- Works in both directions (can turn both left and right)
- Includes a sensor for rotational position

Note: This project was neglected in Fall 2019. In an ideal world, it would be picked up and addressed fully in Spring 2019 so that the electrical automation project can be physically realized. While we believe we left sufficient space to place this system in the existing car, it may be difficult to coordinate all of the details and interfaces.

3.5.2 Mobile idler gear

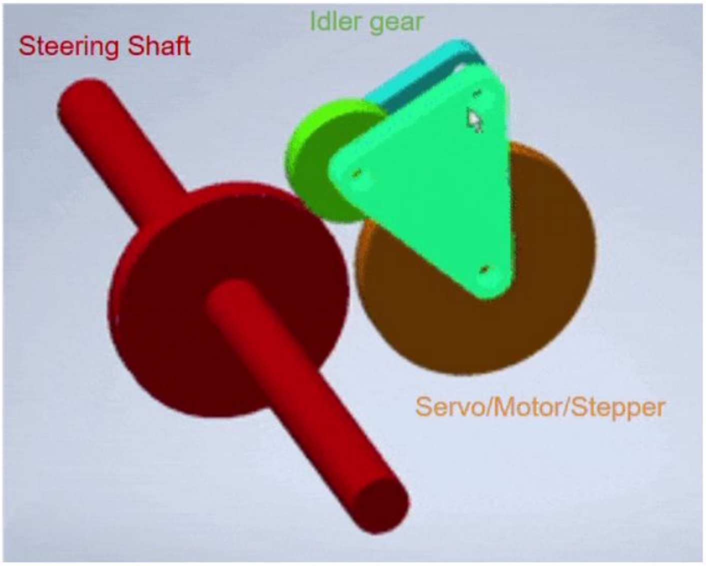


Figure 10**. Mobile Idler Gear Autonomy Architecture**

In this concept, the orange gear represents the input from the servo or stepper motor from the electrical team. The mechanical override control (which would likely be a mechanical brake cable) would move the blue (they are somewhat more green in this image) plates and move the green idler gear into and out of engagement.

In this system, the idler gear and plates would have to be biased towards the steering shaft gear with a spring, likely a torsion spring. While this is not inherently problematic, it does highlight a major issue with this design. When the motor gear is rotating counterclockwise (from the POV shown in the image) the motion of the gears drives the idler gear further into engagement. When it is spinning clockwise, however, it forces the idler gear away from the steering shaft and out of engagement. Balancing the biasing spring with the force from the clockwise rotating motor would be very difficult, if not impossible.

Another important aspect of this design is that once disengaged, the system will lose its reference zero. This means that after disengaging the automation system, the driver would not be able to re-engage it during the same run. While this could be problematic, we believe it isn't much of an issue because the driver would likely not want to re-engage the system if it was making poor driving decisions to begin with.

Pro

- Fairly simple

- Can use a mechanical bike brake handle and cable

Con

- Only works in one direction
- Requires balancing with spring forces and driving forces
- Disengaging loses the reference zero

3.5.3 Translating engagement gear

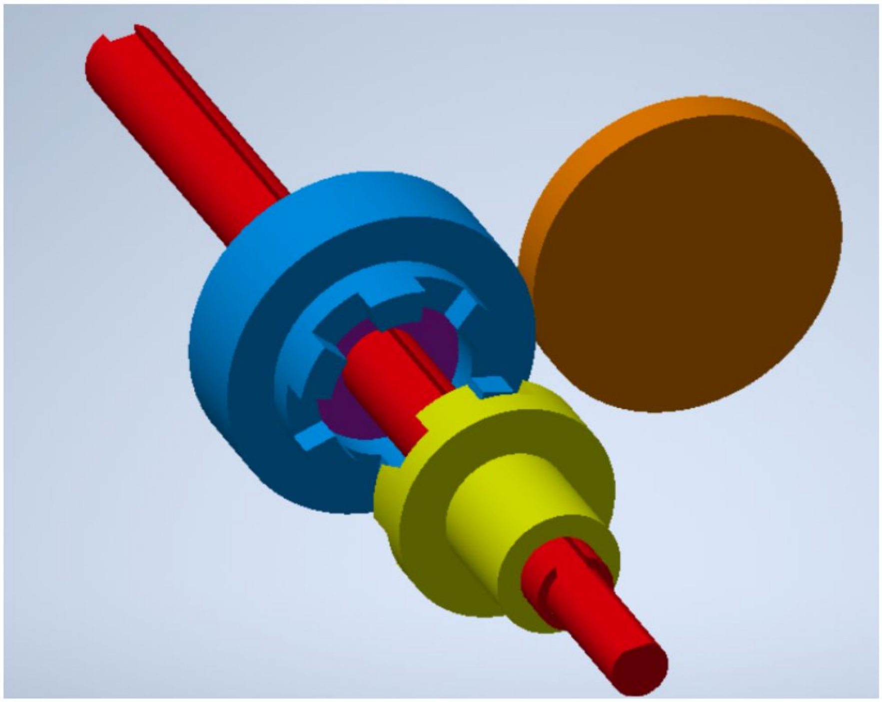


Figure 11**. Translating Engagement Gear Autonomy Architecture**

In this design, the orange gear represents the motor input. The blue gear rotates freely on the red column and the yellow gear is allowed to slide along the red steering column, but not rotate relative to it. The yellow gear is pulled into or out of engagement by the driver. The way in which this happens has not yet been determined. This system would also likely require a biasing spring that would press the yellow and blue gears together.

Unlike the rotating idler gear design, this one works equally well in both directions. It does however, also lose the reference zero in the case of disengagement. It also requires more complex components to determine the sliding and rotating motions and will have more complex machining operations on the steering column.

Pros

- Bidirectional
- Likely very reliable

Cons

- Complicated components/machining

3.6. Motion Ratios and Ackerman Geometry

Ackermann steering is steering in which every wheel is in pure roll, with no lateral skidding, when the car is turning (figure 12). In Ackermann steering, the inner wheel must turn at a sharper angle than the outer wheel because it is moving on a smaller radius circle as shown in the image below. In pure Ackermann steering, the steer angles for the inner and outer wheels can be found through simple geometry as outlined below. This image also contains the equation used to calculate the Ackermann factor. An Ackermann factor equal to 1 has the lowest rolling resistance.

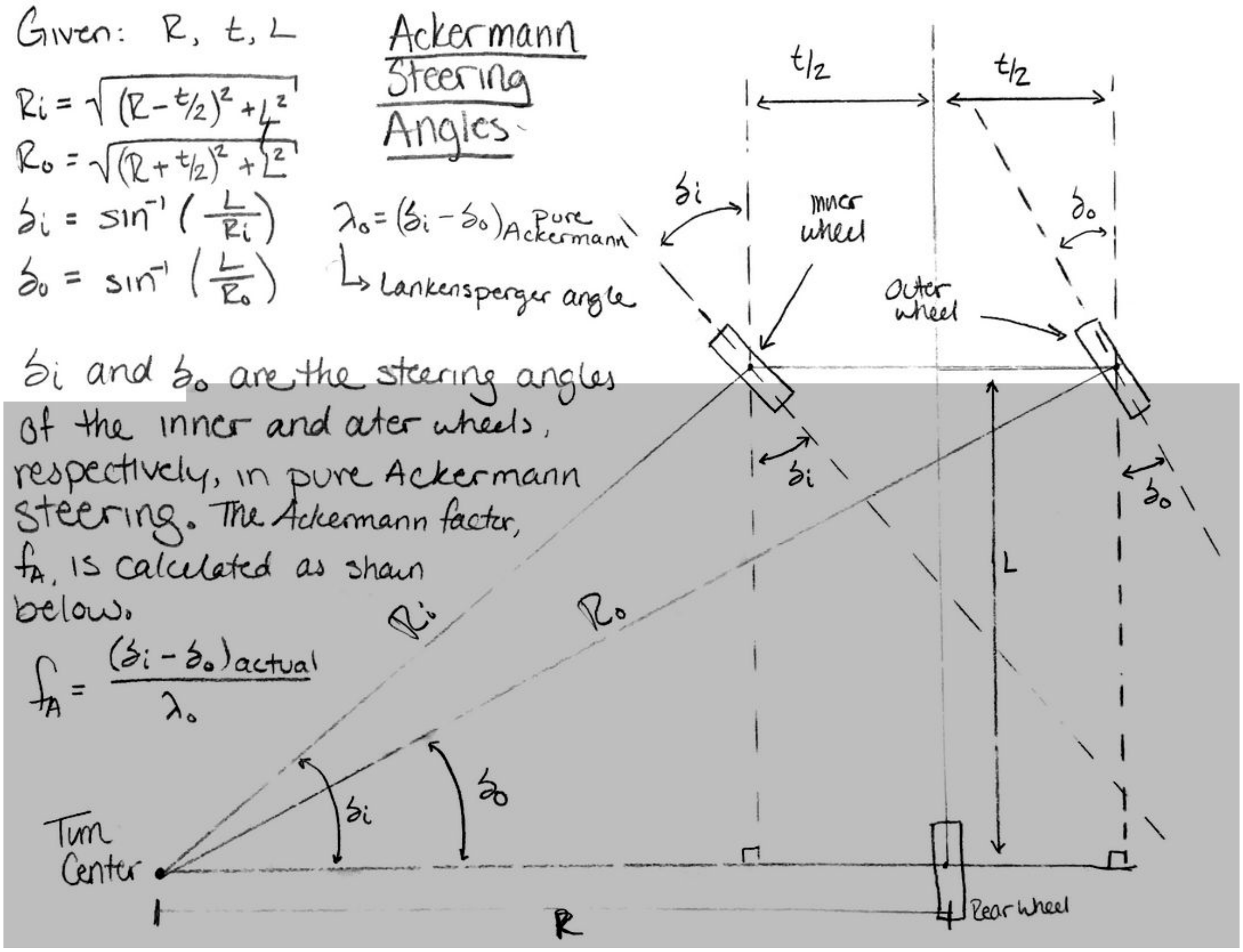


Figure 12**. Ackermann Steering Angles**

We aimed for an Ackerman factor of 1 to minimize slipping and maximize rolling during our (low-speed) turns. From a high level design standpoint, this is done by ensuring that the tie-rod-to-wheel-mount connection point lies on the line which connects the kingpin axis to the center of the rear wheel [1]. Using figure 13 and figure 14, we defined 10 variables and wrote 10 equations of motion from linkage motion. Notably, we used the x, y, and z motion of point C (center of the left and right tie rod connection at the pitman arm) and the x, y, and z motion of point D (the tie rod connection at the ackerman linkage) to define 6 equations.

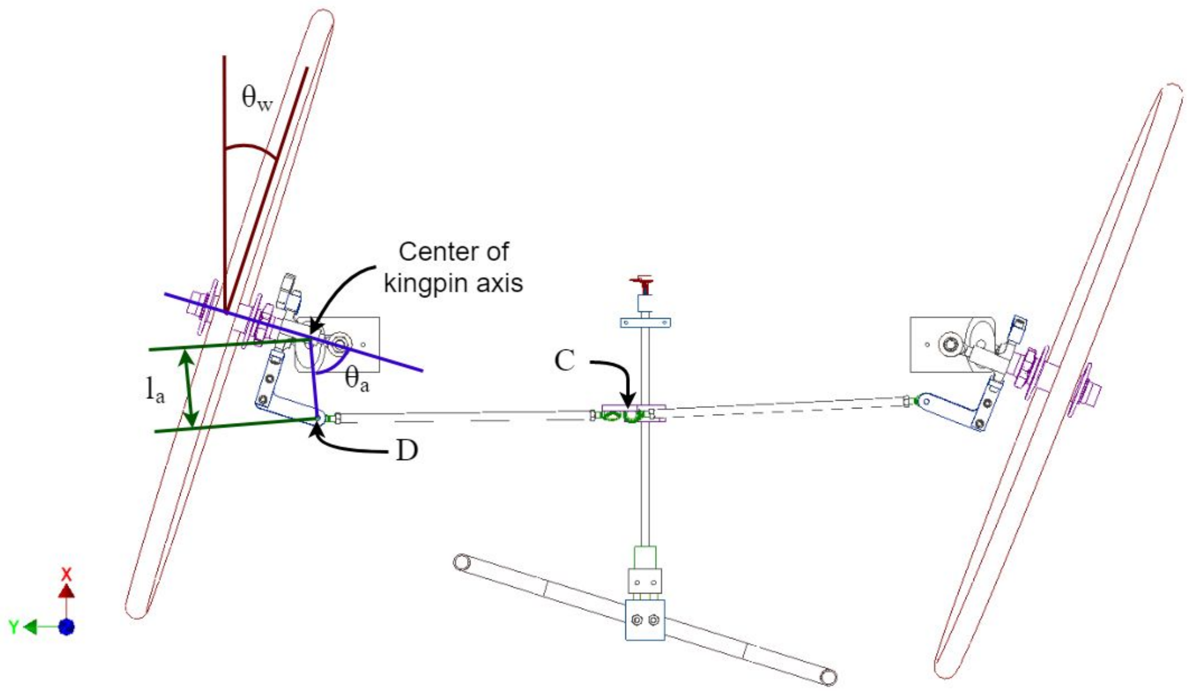


Figure 13**. Dimensions and Angles Used In the Dynamics Equations - Top View**

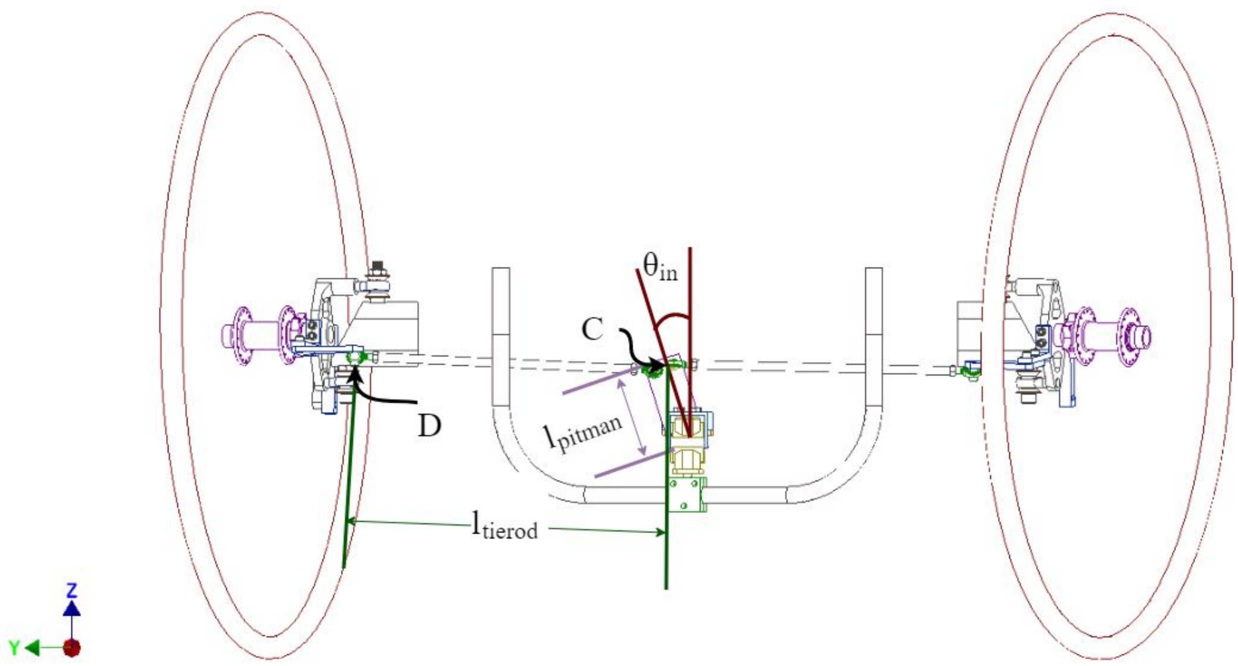


Figure 14**. Dimensions and Angles Used In the Dynamics Equations - Front View**

To define the given parameters, we first calculated what the inner and outer wheel turning angles must be to hit the target turning radius of 5 m (MATLAB file ack_max.m), which ended up being 22.9 degrees for the inner wheel and 18.8 degrees for the outer wheel. We also defined the length of the pitman arm, as this was a geometric constraint from the z-axis direction between the steering column (whose location depends on the seat location) and the mounting location of the tie rods on the wheel mounts. This allows the tie rods to be flat at one z-axis at neutral steering. The wheelbase was set at 1701.8 mm, as this was the minimum length possible with the length of the bulkhead compartment and the height and position of the driver. The track was set at 970 mm, as this was the minimum possible distance between the wheels that will allow them to steer to the angles required for a turning radius of 5 degrees without the wheel covers hitting the sides of the chassis.

Given: track, wheelbase, l_{pitman}, θ_w, θ_{in}

2 Component Length Variables: l_a, l_{tierod}

8 Intermediate Variables: C_x, C_y, C_z, D_x, D_y, D_z, d, θ_a

10 Equations:

$$C_x = 0 \tag{1}$$

$$C_y = l_{pitman} sin(\theta_{in}) \tag{2}$$

$$C_z = l_{pitman}(1 - l_{pitman} cos(\theta_{in})) \tag{3}$$

$$D_x = -(l_a sin(\theta_a + \theta_w) - l_a sin(\theta_a)) \tag{4}$$

$$D_y = l_{tierod} + (l_a cos(\theta_a) - l_a cos(\theta_w + \theta_a)) \tag{5}$$

$$D_z = 0 \tag{6}$$

$$\sqrt{(C_x - D_x)^2 + (C_y - D_y)^2 + (C_z - D_z)^2} = l_{tierod} \tag{7}$$

$$l_a = d - \frac{l_{tierod}}{cos(\theta_a)} \tag{8}$$

$$d = \sqrt{\frac{\text{track}}{2}^2 + \text{wheelbase}^2} \tag{9}$$

$$\theta_a = tan^{-1}(\frac{\text{wheelbase}}{\text{track}/2}) \tag{10}$$

Solving these 10 equations and 10 unknowns in MATLAB gave the result of **83 mm for the length of the ackerman lever** and **462 mm for the length of each tie rod**, as compiled in table 7. The code is in the steering system wiki. We designed our system in Autodesk Inventor with these dimensions, and verified that the CAD was performing at a Ackerman factor of 1 by setting the steering wheel at various angles and plotting the corresponding inner and outer wheel angles in figure 15.

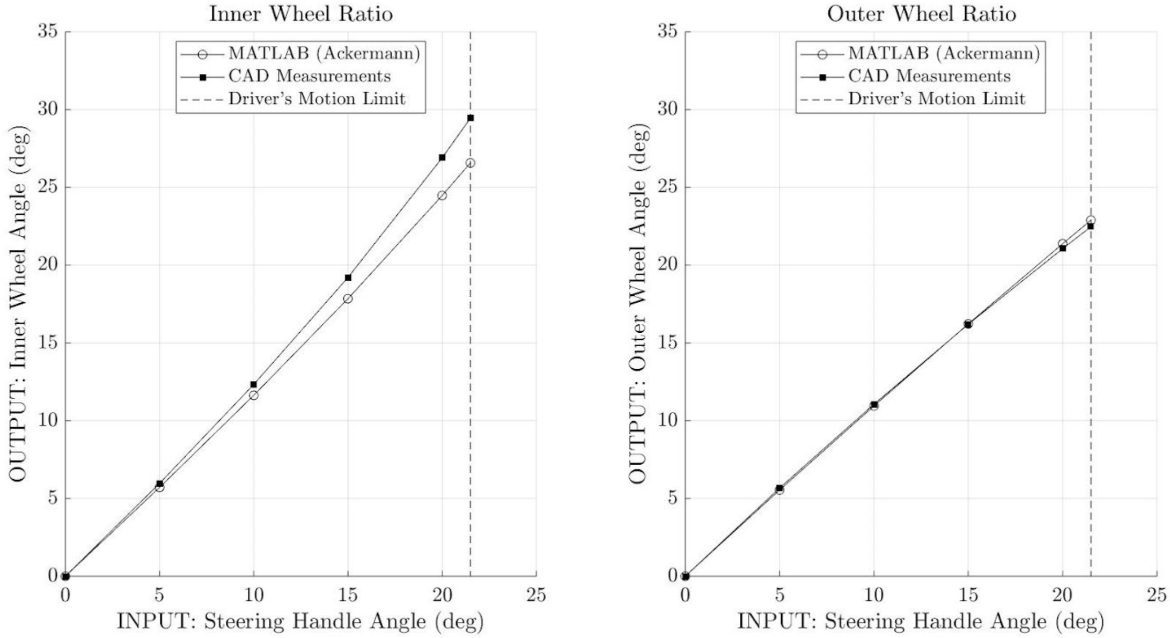


Figure 15**. Motion Ratio Predictions and Comparison With Pure Ackermann Steering**

Figure 15 shows that at the low turning inputs (0-10 degrees) that we expect to see in the competition course, the car behaves as we designed. At high turning inputs (15-22.5 degrees) at the steering handle, the inner wheel actually turns more than we would like it to. These differences are likely because of various small geometric differences from the MATLAB model to the CAD--for example, the ball joints at the pitman arm do not mount onto the same point for both the left and right tie rods because this is physically impossible, but rather mount 0.9 inches away from each other. Small deviations like these likely added up. For the future, it is certainly possible to include these discrepancies in the MATLAB model.

To further verify these results, we manufactured the true-to-size prototype in figure 16. The main functional difference between this prototype and the real car is that the pitman arm still needs weight reduction machining before being used on the real car. The wheel mount was machined with a now-outdated drawing, so this will also need to be re-machined for the real car.

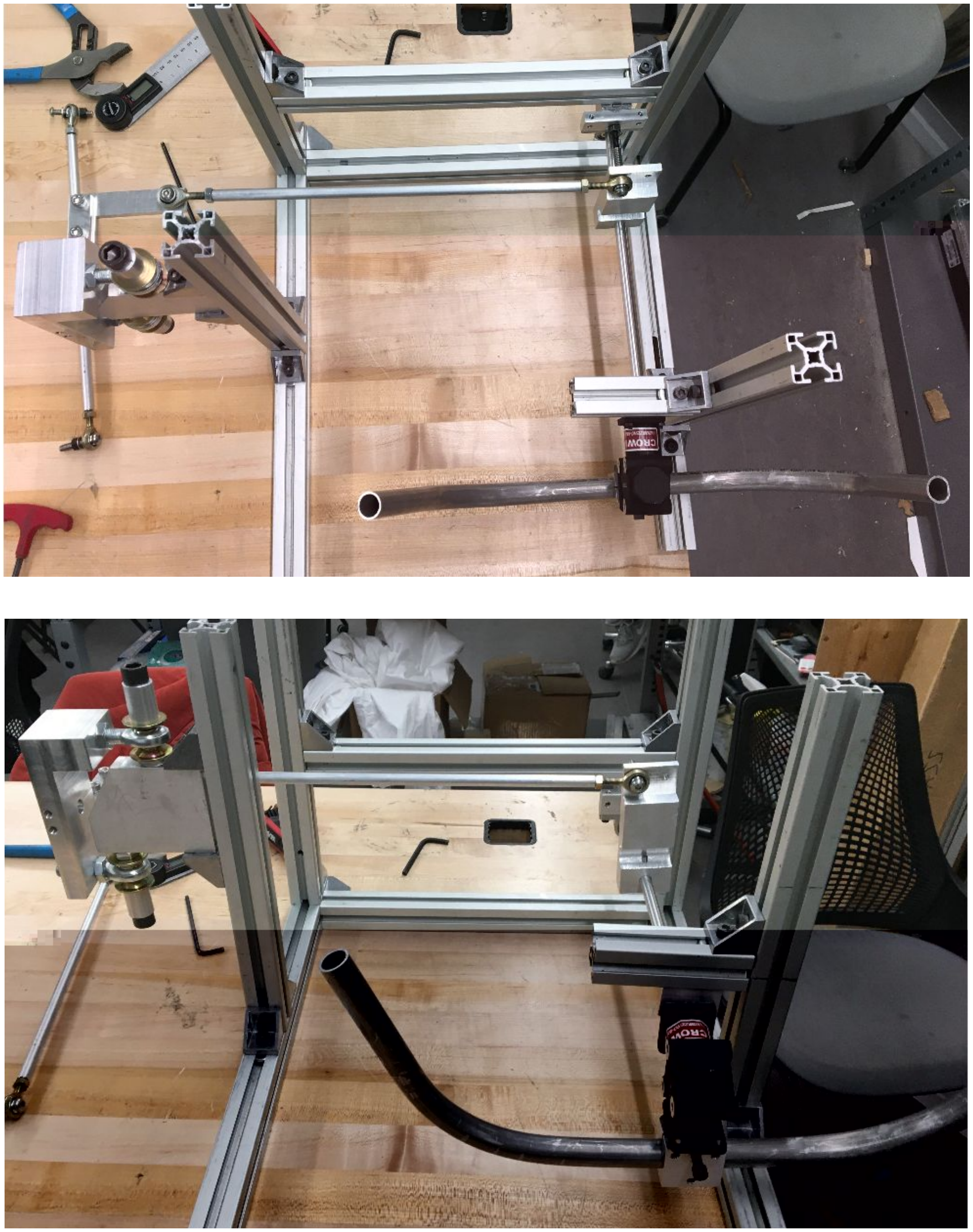


Figure 16**. Prototype Overview**

To measure angles from the prototype, we created the setup in figure 17. On the steering handle side, we braced a flat sheet onto the handle and drew the projection of the line on a fixed piece of paper underneath. On the wheel mount side, we strung a small weight near the end of the bottom ball joint and marked the location of this weight for each angular increment, in addition to marking the pivot point under the rotational center of the ball joint via the same method. After the test, we measured the angles using a digital angle finder. Because of the hand projections and human sight approximations involved in the data gathering, we predict at least a ±1 degree error.

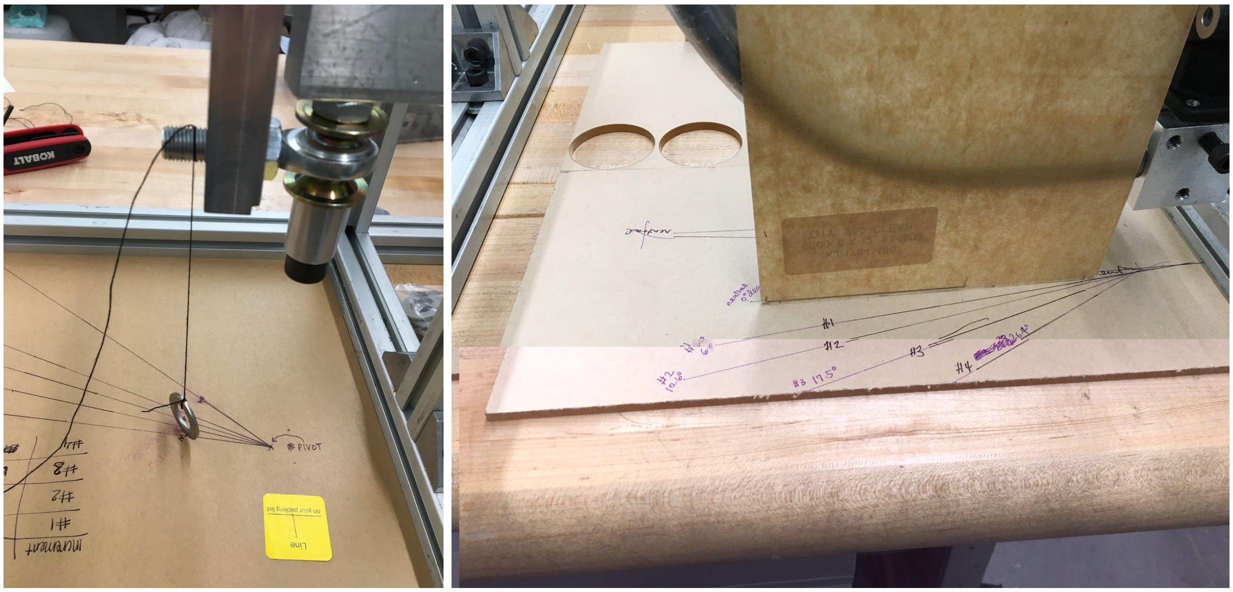


Figure 17**. Prototype Measurement Method**

Table 3**. Prototype Results**

Measurement Increment	**Angle at Steering Handle (degrees)**	**Outer Wheel Angle At Wheel Mount (degrees)**
1	6.2	7.1
2	10.1	10.6
3	16.6	16.0
4	26.4	31.2

Comparing the prototype data with the MATLAB Ackerman Predictions and the CAD angle measurements figure 18. Considering the at least ±1 degree inaccuracy in our data collection method, this is, within that uncertainty, accurate to the CAD predictions, which had an Ackerman factor of greater than 0.98.

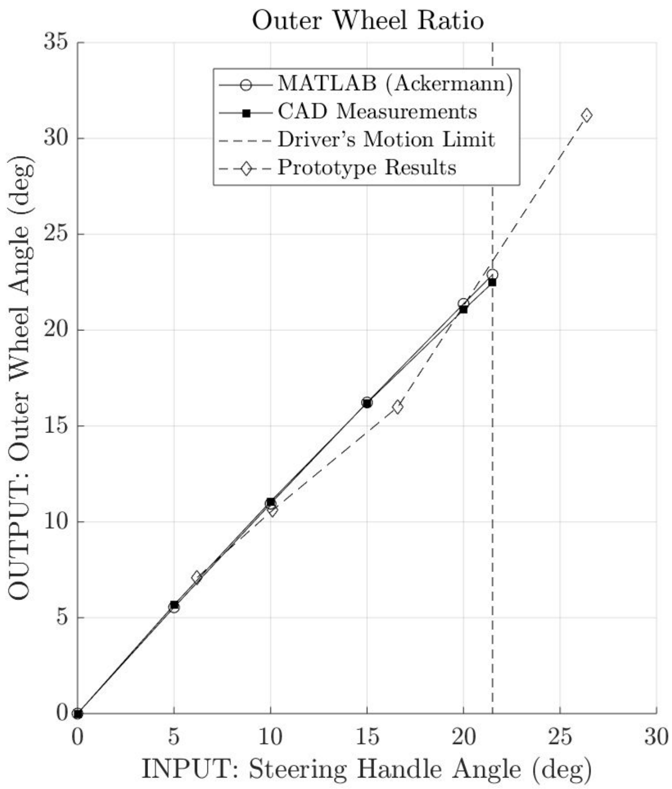


Figure 18**. Outer Wheel Angle Against Steering Handle Angle**

3.7 Tapered Pin

A tolerance stack-up of the steering system showed that 2.44 degrees out of the 3.36 degrees of expected play came from the design of the steering column mounting design to the pitman arm. We were able to eliminate this tolerance source altogether using a tapered pin which reduced the expected play by 73% from 3.36 degrees to 0.92 degrees.

We did a tolerance stack up analysis of angular play with no loads, summarized in table 4. The “Tolerance” column contains the gap in inches between the mounted and mounting component. This defines how much a chosen reference point in a mounted component can move without rigidly moving the mounting component as well. The “Arm Length” column contains the lever arm in inches that will multiply the “Tolerance” column to result in the angular displacements in the direction of the steering column rotation. The most common source of play was the close fit holes on the mounted components that need to be bigger than the screws or pins of the mounting component to be able to assemble the components together.

Table 4. **Original Tolerance Stack Up With Set Screw Through Pitman Arm Design**

	Tolerance	Arm Length	Angular Tolerance	Angular Tolerance	Sources of [illegible] Tolerance
	in	in	rad	deg	
Wheel Mount	0.0050	2.110	0.0024	0.136	Drill tolerance
Bolt @ Wheel Mount	0.0070	2.110	0.0033	0.190	Close fit clearance diameter
Ball Joints @ Wheel Mount	0.0025	2.110	0.0012	0.068	McMaster
Ball Joints @ Pitman Arm	0.0025	2.250	0.0011	0.064	McMaster
Pitman Arm	0.0050	2.250	0.0022	0.127	Drill tolerance
Bolt @ Pitman Arm	0.0070	2.250	0.0031	0.178	Close fit clearance diameter
Pitman Arm @ Steering Shaft	**0.0080**	**0.188**	**0.0426**	**2.443**	**Clearance diameter**
Steering Shaft @ Collar				0.000	Set screw angular tolerance
Collar @ Gearbox				0.000	Set screw angular tolerance
Gearbox Bevel Gears				0.000	No known spec for backlash
T-adapter @ Gearbox				0.000	Set screw angular tolerance
Steering Handle @ Collar	0.0005	0.188	0.0027	0.153	Tapped vs thread diameter
Tolerance Stack Up				**3.36**	

Figure 19 shows views of the new tapered pin mounting design with the tapered pin hammered into a tapered hole through the pitman arm and steering column.

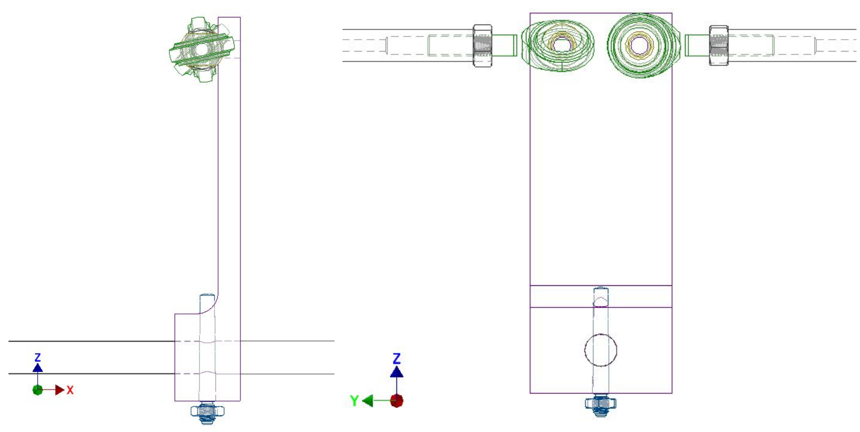


Figure 19. **Views of the Tapered Pin Through the Pitman Arm and Steering Column**

Table 5 shows the updated play contribution from the pitman arm with the tapered pin design, and the corresponding total play.

Table 5. **Tolerance Stack Up With Tapered Pin Through Pitman Arm Design**

Pitman Arm @ Steering Shaft	**0.0000**	**0.188**	**0.0000**	**0.000**	**Tapered pin clamp**
Tolerance Stack Up				**0.92**	

We measured the play in our prototype by fixing the wheel mounts (holding it to ensure no movement) and measuring how much the steering handle moves as shown in figure 20 and found 2.3 degrees of play from the Zero-Max Gearbox. There was likely play from other component mounts as well, as we predicted through table 4 and table 5, but the prototype was not accurate enough to measure less than 1 degree of play. The tapered pin was effective because there is no noticeable play in the pitman-arm-to-steering-column connection.

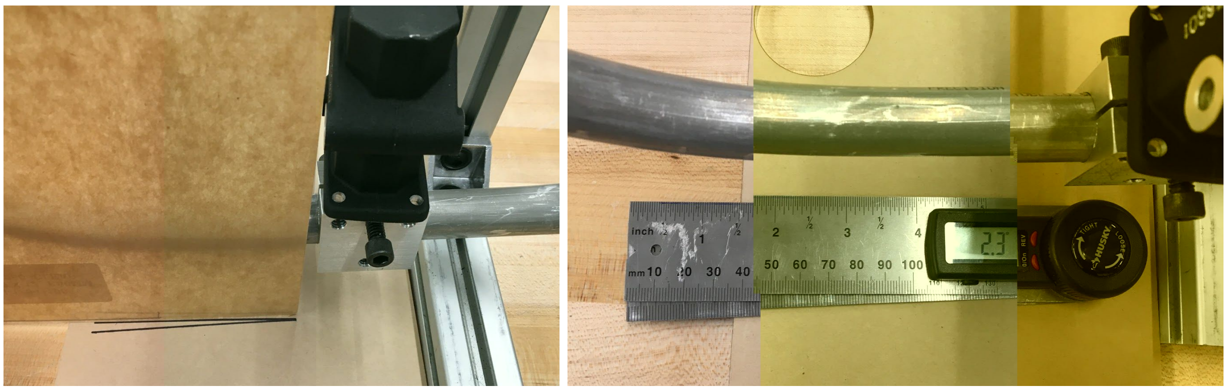


Figure 20**. Prototype Setup to Measure Play**

3.8 Rollover Performance

Rollover risk was not of much concern this year, as the wheelbase remained at a similar range and the track became wider, making the car more stable. So we used the rollover calculation Excel sheet from last year to verify that the car's acceleration will not cause it to rollover. The track and wheelbase are parameters in the equation for the rollover load case, which affect wheel mount analyses, as mentioned in 2018-19 Wheel Mounts Technical Report.

3.9 Loaded Deflection

To do weight reduction, we specified a parameter that we cannot surpass. Usually in classes this is the strength of the material because we don't want to break components. In our case, this is a lesser concern because of the low load cases we encounter in the steering system. It is more likely that we will run into issues with binding or elastic flexure of the steering system far before we start breaking things. 2019-20 was the first time we did weight reduction with deformation as the limit, and we chose the value of 1 degree to begin with. This 1 degree had contributions from each individual component. Our design met this specification.

We can't know whether this specification is too much or too little without the real car and real loads, but once we have the real car assembled, we should test out the steering system to see if components are deflecting enough for the driver to feel the deflections. If it is, then we need to decrease this specification for future years. Hopefully in future years, this parameter can be better defined and can be a helpful design tool.

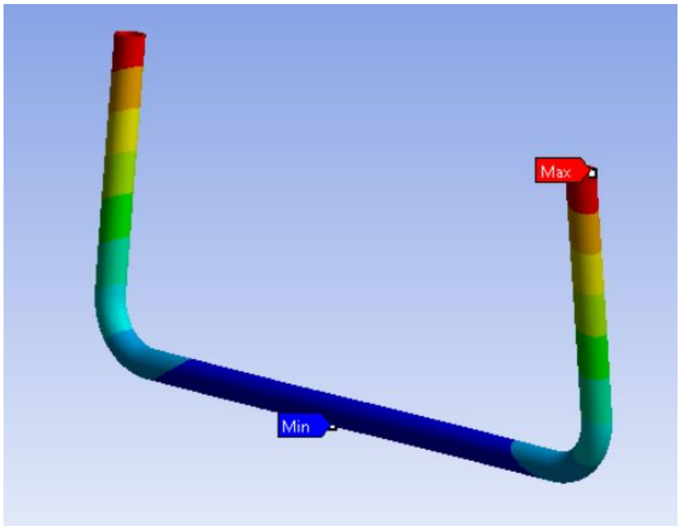
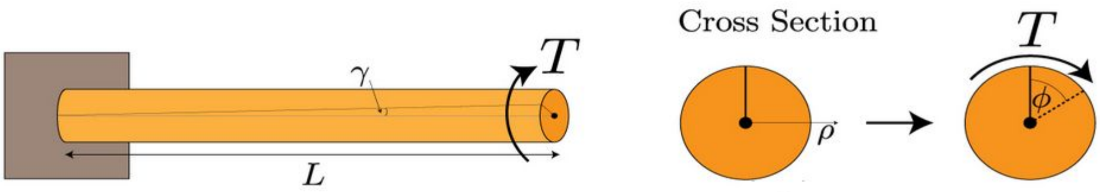


Figure 21**. Steering Handles Deflection in ANSYS**

For the handle, we used ANSYS to simulate bending (figure 21). We split the outer face of the handle at the edges where it would contact the gearbox adapter and fixed that segment of the face. We applied loads to the ends of the steering handles (assuming a worst case scenario) that were equivalent to a torque of 3 Nm. This 3 Nm was measured from the torque needed to turn the wheels on the Kiwi Kruiser, but should be refined in future years. The results from iterating on inner and outer diameters for the handle are stored in the first tab of this spreadsheet.

We assumed that the angular deflection in the gearbox and the Pitman arm would be negligible when compared to the deflection due to the other components. In the Spring semester, weight reduction of the Pitman arm might be worthwhile. The current manufactured Pitman arm could always be used as a pre-machine for a CNC-ed one with weight reduction cutouts.



Max Deflection:

$\phi = LT/(JG)$

$\phi = 32\, LT/(G \pi (D^4 - d^4))$

Max Stress:

$\phi = TR/J$

$\phi = 32\, TR/(\pi (D^4 - d^4))$

Figure 22**. Torsional Loading of a Circular Section Beam**

Calculations for deflection in the steering column can be found in the second tab of this spreadsheet. The steering column calculations are based off of the statics equations in figure 22.

Table 6 (link) adds up all the deformations from the selected components and shows that the anticipated total angular deflection under load is 0.934o.

Table 6**. Loaded Deflection Summary**

Part	Loaded Deflection (degrees)
Handle	0.089
Column	0.748
Tie Rods (x2)	0.0964
Total	0.934

4. Systems Integrations

4.1 Mechanical Interfaces

Mechanically speaking, the steering system interfaces with the chassis, the wheel mounts, and the driver.

4.1.1 Chassis Interfaces

1 **Gearbox** (X2 ¼-20 holes on seat, flat under seat, space inside lower fairing)

2 **Bearing** (X2 ¼-20 holes on seat slope)

3 **Steering column, pitman arm** (1”x2” rectangular-ish hole on seat slope)

4 **Tie rods** (1” holes on L and R walls)

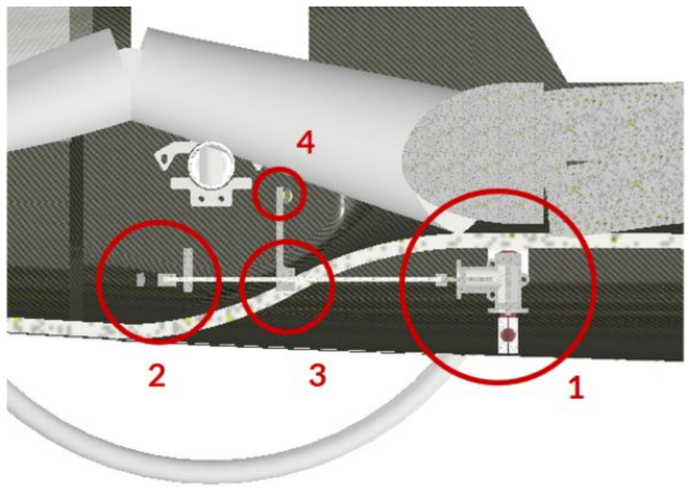


Figure 23**. Chassis Interfaces**

There are 4 places in which the steering system interfaces with the chassis, as shown in the image above.

The first is the mounting location of the gearbox on the underside of the seat. While we don't anticipate it being a problem due to low load cases, it may be worth trying to approximate a hard point in the chassis at that location by replacing that section of core material with an aluminum plate of the same thickness.

The second of these is the mounting location for the bearing. The bearing mount will be underneath the bearing for the real can even though it points upwards into space in the image above. This is because the image above if of the CAD which was used for the prototype and does not reflect the "real car" version of the mount.

The third and fourth of these interface locations are both holes that will need to be cut into the chassis so that the steering column, Pitman arm, and tie rods can pass through it. None of these requires a lot of precision as they just provide space for the steering components and do not involve any bolts or other locating devices.

4.1.2 Wheel Mount Interfaces

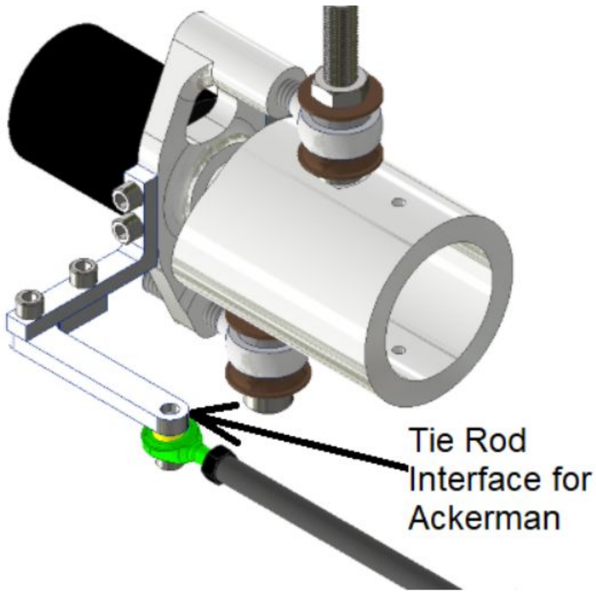


Figure 24**. Wheel Mount Interfaces**

The wheel mount interface is important because it defines a lot of the steering geometry. In order to implement Ackermann steering, the point where the tie rods connect to the wheel mounts needs to lie on the line which goes from the kingpin axis to the rear wheel center [1]. Since the kingpin axis is tilted, we made it so that the tie rod would connect in the XY plane which intersects the wheel centers ($z = 0$) and that line is taken from where the kingpin axis intersects the $z = 0$ plane.

4.1.3 Driver Interfaces

The steering handle is manipulated directly by the driver and therefore must be in a location that they can reach. The range of motion of the driver must correspond to the range of motion of the steering handle. In order to determine what this range was, we asked our driver, Kathleen, to put on the helmet and lie on the floor in our anticipated driver's position. We used a filing cabinet to represent the bulkhead that she leaned her head up against. From this, we used masking tape to mark the points on the floor which corresponded to the maximum and minimum x positions that she could reach with her hands without bending her elbows outwards. We then measured these points relative to the bulkhead and put them into a CAD sketch. The 4 points formed the corners of a rectangle which is shown in the image below. The center of this rectangle is where the gearbox must go and the y-direction width of this rectangle shows how wide the U-shaped handle must be.

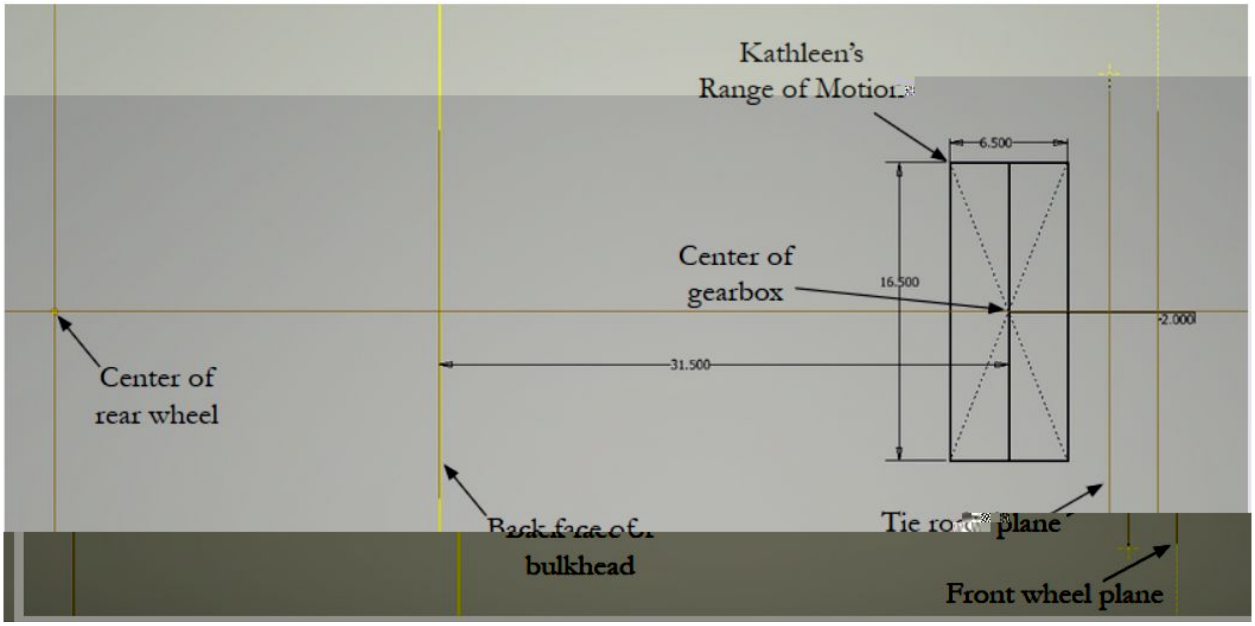


Figure 25**. Driver in Car Geometry**

4.2 Electrical Interfaces

4.2.1 Autonomy Interfaces

The steering system interfaces with the electrical autonomy project because the autonomy project needs a way to drive the steering system. Very little work was done on this interface throughout the Fall semester, but the general architecture of the steering system was chosen such that it is compatible with autonomy. We believe that the autonomy motor will somehow drive the steering column in a way that can be mechanically overridden. This has not been fleshed out and the details have not been finalized, although we do have a few ideas which can be found in Section 3.5.

4.2.2 Driver Control Interfaces

When driving, the driver must manipulate a few electronic controls which include the throttle and the horn. The mounts for both of these components have not yet been designed, but they should be fairly straight forward 3D printed parts. We believe that both of them will go on the steering handles. The general shape of the throttle from last year is well suited to mounting this way and the horn button will require a 3D printed mount regardless of where it is placed.

5. Final Design Specifications

5.1 Design Overview

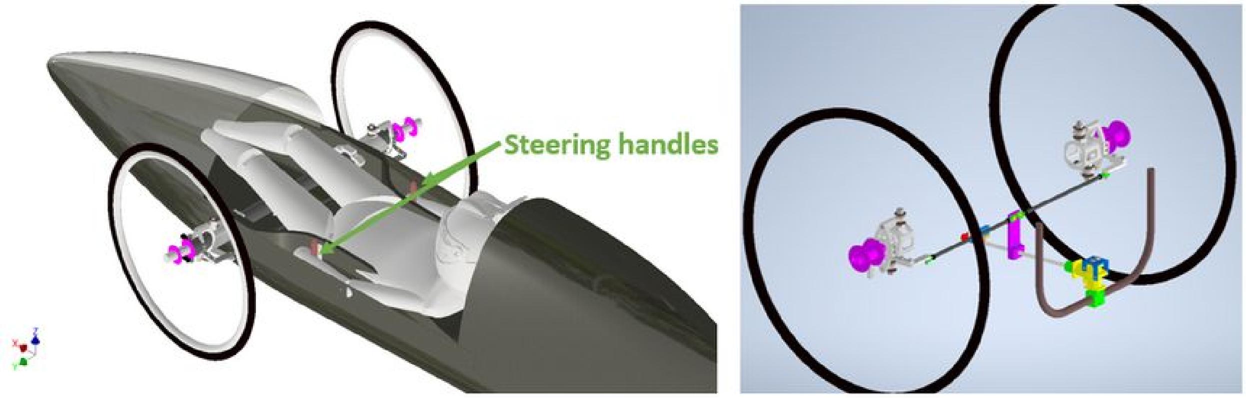


Figure 26**. Overall Steering System**

The architecture for the 2019-2020 steering system is a U-shaped steering handle with a 90° gearbox, a pitman arm, and tie rods. There have not yet been any definitive plans for autonomy integration. The geometric parameters are tabulated below.

Table 7**. Final Overall Car Design Specifications**

Parameter	Cuckoo Caravan Value	Why?
Track	970 mm	Needs enough space to accommodate wheels turning without hitting chassis
Wheelbase	1701.1 mm	Although our driver got shorter, the amount of space needed for rear assembly grew
Camber	0°	0 camber has the lowest rolling resistance
Kingpin	17.82°	positive kingpin increases steering stability without affecting rolling resistance
Scrub Radius	0 mm	high scrub radius increases rolling resistance when turning
Castor	0°	allows us to align the wheel mounts when assembling the wheel mounts
Toe in/out	0° (adjustable)	zero toe has the lowest rolling resistance but a slight toe in can be helpful in preloading and therefore reducing the play in the steering system

Table 8**. Final Steering System Design Specifications**

Component	**Length (mm)**
Each Tie Rod	462.24
Pitman Arm	89.77
Ackerman Lever	83.02

Table 9**. Compilation of Masses of Each Component of the Steering System**

Component	**Mass (kg)**
Ball joints (x4)	0.0363
Tie rods (x2)	0.0286
Pitman Arm	0.111
Steering Column	0.0567
Gearbox Adapter (both halves)	0.0648
Handle	0.213
Gearbox Mount	0.0286
Bearing Mount	0.0304
Bearing	Not Listed
Gearbox	0.340
Shaft Coupler	Not Listed
Total (neglecting “Not Listed” items and fasteners not included above)	**0.910 kg**

For the driver interface we have selected the simple under the butt design because it is the best way to ensure basic levels of ergonomic comfort while maintaining full visibility. Although it does move in an unintuitive direction, we believe that this can easily be addressed by having the driver practice driving the car before competition. This design also makes the timed exit test rather trivial (the hardest part has always been maneuvering around the steering wheel), which is nice.

For rotation transmission, we have selected the Pitman arm design because of its tolerance for inaccuracies in mounting and it's lower cost. Since we are spending a decent amount of money on the 90° gearbox, we do not want to buy an expensive rack and pinion system. We also want our system to be compatible with autonomy, which rules out the direct manipulation design. We have also analyzed the effects of the nonlinear nature of a Pitman arm system and have deemed them acceptable.

5.2 Components

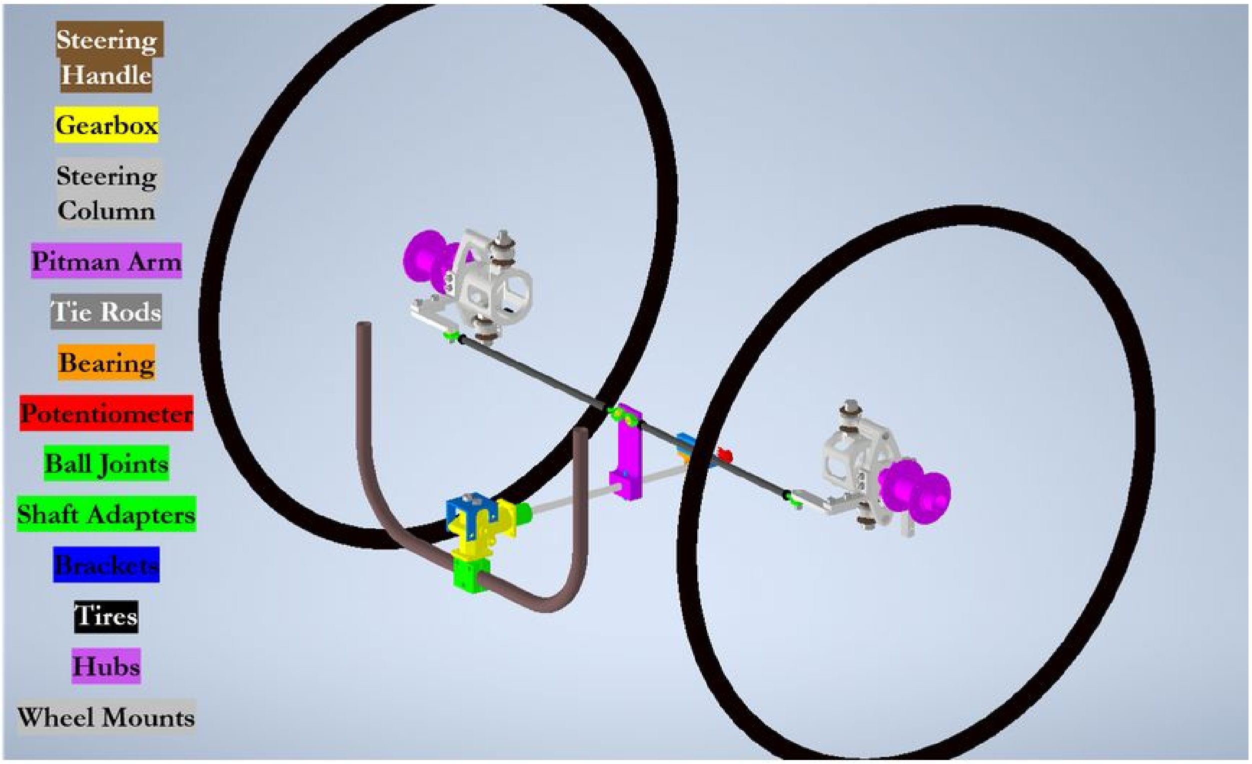


Figure 27**. Labelled Components of the Steering System**

5.2.1 Off-the-Shelf Component Selection Process

The off-the shelf components are tabulated in the bill of materials section in table 12.

We selected our gearbox for it's light-weight, ability to handle the anticipated 3 Nm of torque, smooth bearings and low play. We wanted to order an off-the-shelf gearbox instead of making one ourselves because we believed we would get a much better result with much better alignment that way. It turns out that the gearbox we order likely has more play than we needed. We plan on reaching out to the manufactured to see if there is another gearbox that they sell that may more closely fit our needs. Zero-Max Gearbox

The shaft adapter was selected because it is the simplest way to rigidly couple the ends of two shafts with little to no play. The one we selected also has some bend to it which means it is tolerant of small amounts of both angular and locational misalignment. This is helpful for us when we are mounting to the chassis which is always uncertain. Flexible Shaft Collar

The LH and RH (left handed and right handed threads) ball joints were selected for their small amounts of play. They are sold by Aurora. Each tie rod will have a left handed ball joint on one and a right handed ball joint on the other. Each of these ball joints will require locking nuts to secure them in place when the car is moving and therefore rattling around. Aurora SPM-4 and SPB-4

The bearing was selected because it is of high quality and matches the diameter of the steering shaft, the shafts on the gearbox, and the shaft adapter. Steering Shaft Bearing

6. Manufacturing

6.1 Prototype Progress Tracking

While we were in the prototype process we used this chart to track the status of each part.

Table 10**. Progress Tracker While Prototyping**

Part	DWG?	Stock?	MFG?	Inspected?	Can it be used for real?
Handle					NO - Get Clark to bend tubes?
Handle to GB A					YES BUT - Could fix hole locations & dim
Handle to GB B					YES BUT - Could fix hole locations & dim
Steer column					YES
Pitman arm					YES BUT - Take out material; update dims
Gearbox mount					YES
Bearing mount					NO - Redesign to mount to bottom of car
Tie rod L					YES
Tie rod R					YES
80-20 frame					NO - For prototype only
Tie Rod Interface1 L					YES BUT - Take out material
Tie Rod Interface1 R					YES BUT - Take out material
Tie Rod Interface2 L					YES BUT - Take out material
Tie Rod Interface2 R					YES BUT - Take out material
Mount - Wheel Side L					PROBABLY NO - dimensions slightly modified?
Mount - Wheel Side R					PROBABLY NO - dimensions slightly modified?
Mount - Car Side L					NO - For prototype only
Mount - Car Side L					NO - For prototype only

Part	Here?
gearbox	
shaft collar	
bearing	
ball joints (x4)	
pin(s)	
tapered reamer	

6.2 Manufacturing Plan

Table 11. **Manufacturing Plan**

Part	Quantity Needed	Material	Op 1	Op 2	Op 3	Estimated # of Machining Shifts	Design Finalized?	Stock/tools ordered?	Drawing Inspected?	Done?
Handle	1	Al 6061	Bend (Clark?)	NA	NA	NA	yes	yes	yes	no
Gearbox Adapter A	1	Al 6061	manual mill	NA	NA	1	yes	yes	yes	yes
Gearbox Adapter B	1	Al 6061	manual mill	NA	NA	1	yes	yes	yes	yes
Steering Column	1	Al 6061	manual lathe	manual mill	NA	1	yes	yes	yes	yes
Pitman Arm	1	Al 6061	manual mill	NA	NA	1.5	yes	yes	yes	yes
Tie Rods	2	Al 6061	manual lathe	NA	NA	1	yes	yes	yes	yes
Bearing Mount	1	Al 6061	manual mill	NA	NA	1	no	yes	no	no
Bearing Mount Chassis Shim	1	ABS	3D print (RPL)	NA	NA	NA	no	NA	NA	no
Gearbox Mount	1	Al 6061	manual mill	NA	NA	1	yes	yes	no	no

These parts will comprise the custom manufactured aspects of our design. They are primarily manual mill parts and none of them requires the use of a CNC.

The bearing mount chassis shim will be a 3D printed part made to fit in the space between the organically contoured chassis and the square bearing mount. The bearing mount and this shim piece will have to be redesigned or designed in the Spring semester as the versions which are currently in CAD reflect the prototype mounting which is slightly different. It will also likely be helpful to finalize and machine these parts after chassis manufacturing is somewhat complete so that inaccuracies in chassis manufacturing can be accounted for.

6.3 Assembly Plan

Our current alignment plan is to align the steering system based off of the front tube. Using a profile of XY plane of car as a jig (lasercut) with perpendicular attachments for hole patterns and tube alignment, we can determine mounting locations for the gearbox and bearing. The wheel mounts will be based off of front tube (and wheels) so it makes sense to use that a reference and not the bulkhead. It does not matter whether wheel mount alignment or steering alignment happened first.

6.4 Segmented BOM

Total Spent in Fall 2019: $486.79

6.4.1 Off-the-Shelf Components

Table 12**. Segmented BOM of Off-the Shelf Components**

Part	Vendor	Quantity	Specs/Part Number	Cost (for all)	For prototype, real car, or both?
Gearbox	MFG: Zero-Max C138801; Vendor: Kaman Industrial Tech	1	ZER C138801_BEVEL GEAR DRIVE 2WAY 1:1 3/8 (CROWN)	$254.00	both
Shaft Adapter	McMaster	1	9861T619	$39.06	both
LH Ball Joints	Aurora Bearings	4 (need 2)	SPM-4	$3.19	both
RH Ball Joints	Aurora Bearings	4 (need 2)	SPB-4	$3.19	both
Bearing	McMaster	1	8600N5	$40.16	both

Total Off-The-Shelf Components Cost: $339.6

6.4.2 Hardware and Fasteners

Table 13**. Segmented BOM of Hardware and Fasteners**

Part	Specs	Quantity	McMaster Part Number	Cost (for all)	For prototype, real car, or both?
Gearbox Adapter Bolts	1/4-20 L: 0.75	100 (need 4)	91251A540	$11.38	both
Gearbox Set Screw	1/4-20 L: 0.25	50 (need 1)	91375A533	$9.22	real
Tapered Pin	#2	5 (need 1)	Grainger: 92281A221	$5.91	both
LH nuts for tie rods	1/4-20	2	Already have	$0	both
RH nuts for tie rods	1/4-20	2	Already have	$0	both
Bolts to secure tie rods	1/4-20L: 0.75	8	91251A540 (incl. above)	$0	both
Bolts for bearing mount	10-24 L: 0.75	100 (need 4)	91251A245	$12.07	both

Total Hardware Cost: $38.58

6.4.3 Tools

Table 14**. Segmented BOM of Tools**

Tool	Specs	Quantity	Vendor	Part Number	Cost (for all)
Tapered Reamer	#2	1	Grainger	20D555	$50.50

Total tools cost: $50.50

6.4.4 Custom Components (Stock)

Table 15**. Segmented BOM of Custom Components**

Part	Material	Quantity	Stock Size	Specs	McMaster Part Number	Cost (for all)	For prototype, real car, or both?
Handle	Al 6061	2 (need 1)	L = 2 ft	OD: 0.75"; ID: 0.62"	9056K71	$38.08	both
Gearbox Adapter A & B	Al 6061	1	lamp stock	NA	NA	$0	both
Steering Column	Al 6061	2 (need 1)	L = 1 ft	OD: 0.375"	8974K24	$5.00	both
Pitman Arm	Al 6061	2 (need 1)	lamp stock	lamp stock	lamp stock	$0	both
Tie Rods	Al 6061	2	L = 2 ft	OD: 0.375", ID: 0.145"	1658T32	$3.00	both
Bearing Mount	Al 6061	2 (1 proto, 1 real)	lamp stock	lamp stock	lamp stock	$0	1 proto, 1 real
Bearing Mount Chassis Shim	ABS (3D print)	1	NA	NA	NA	$0	real
Gearbox Mount	Al 6061	1	1.5"x0.5"x3.25"	1/8" wall thickness	6546k61	$12.03	real

Total Stock Cost: $58.11

7. References

[1] Dixon, John. *Suspension Geometry and Computation.* John Wiley & Sons Ltd. 2009.

[2] Resistance Racing. *Steering System.* Resistance Racing Wiki. 2019. https://github.coecis.cornell.edu/Resistance-Racing/Resistance-Racing-Wiki/tree/master/mechanical/subteams/frontassembly/steering

[3] Resistance Racing. *2018 Steering System Technical Report.* Resistance Racing Wiki. 2019. https://github.coecis.cornell.edu/Resistance-Racing/Resistance-Racing-Wiki/blob/master/mechanical/subteams/frontassembly/steering/FA18_TechnicalReport_Mechanical_Steering.pdf

[4] Resistance Racing. *2018 Wheel Mounts Technical Report*. Resistance Racing Wiki. 2019. https://github.coecis.cornell.edu/Resistance-Racing/Resistance-Racing-Wiki/blob/master/mechanical/subteams/frontassembly/steering/FA18_TechnicalReport_WheelMounts.pdf

Appendix 1. SEM 2019 Relevant Rules for Steering

ARTICLE 39: DIMENSIONS

a) The vehicle maximum height must be less than 1000 mm.

b) The vehicle track width must be at least 500 mm, measured between the midpoints where the tyres of the outermost wheels touch the ground.

c) The ratio of height divided by track width must be less than 1.25.

d) The vehicle wheelbase must be at least 1000 mm.

e) The maximum total vehicle width must not exceed 1300 mm.

f) The maximum total length must not exceed 3500 mm.

g) The maximum vehicle weight, without the Driver is 140 kg.

h) None of the body dimensions above must be achieved by design singularities such as 'stuck-on' appendages or cut-outs.

ARTICLE 42: TURNING RADIUS AND STEERING

a) Only front wheel steering is permitted. If the Organisers are not satisfied with the effectiveness and/or control of a vehicles steering system, this vehicle will be removed from the competition.

b) The turning radius must be 8 m or less. The turning radius is the distance between the centre of the circle and the external wheel of the vehicle. The external wheel of the vehicle must be able to follow a 90° arc of 8 m radius in both directions. The steering system must be designed to prevent any contact between tyre and body or chassis.

c) Electrically operated indirect steering systems are permitted providing they are operated by a steering wheel or similar (rotary potentiometer), joystick operation is not permitted. If electronic steering systems are used, in the event of system failure, the vehicle must be equipped with manual steering override.

d) The Organisers reserve the right to set up a vehicle handling course to verify the following when the vehicle is in motion: driver skills, turning radius and steering precision. For example, the Organisers will verify that steering is precise, with no play.