Open Wheel Racecar Steering

by

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ABSTRACT

The steering system of a rear wheel drive open wheel racecar is the only directional control the driver possesses while driving. Steering linkages must be carefully designed to allow cars to navigate turns without exhausting the driver. Motorsports vehicles are designed to make tight turns while maximizing tire grip to maintain higher velocities in corners. Steering geometry must be optimized not only for car performance, but also to maximize driver comfort and improve the “feel” of the vehicle. In competitive motorsports, the steering system is critical to vehicle performance: an incorrectly designed system can at best cost a few fractions of a second on the track, and at worst cause severe driver injury. In the Formula SAE competition, student teams are tasked with designing and manufacturing all subsystems of a racecar for an annual competition while balancing safety, cost, and performance. This thesis will introduce fundamentals of steering system design, and will document in detail the design, analysis, manufacture, and testing of the 2017 MIT FSAE steering system.

Thesis Supervisor: Amos G. Winter
Title: Assistant Professor
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1. INTRODUCTION

1.1 DESIGN GOALS

To be successful, an FSAE car must be as light as possible, low cost, reliable, and be designed through meticulous iteration. These principles also apply to the design of a successful steering system. Closely linked to suspension performance, a good steering system is essential for a quality racecar. Goals and constraints can be identified to help guide the design process. Potential design goals and parameters for a high performance steering system are summarized in Figure 1. The entries were chosen based on previous system designs and made to align with higher-level vehicle objectives.

| Successful navigation of hairpin turns | 9m outer diameter (2m inner) |
| Optimized tire grip for skidpad event | 15.25m inner diameter turn |
| Minimal system mass | < 3kg |
| Reasonable driver input force | Maximum 20 lbs. per arm |
| Ergonomic positioning and packaging | Driver position jig |

Figure 1 – Table relating design parameters and their values.

These goals were referenced during the design process to help validate decisions. To compete in the autocross event, the car must be able to navigate a minimum inner diameter hairpin turn of 2m with a road width of at least 3.5m. One of the dynamic events at competition is the skid pad, a figure 8 with 2 constant inner diameter turns of 15.25m (Figure 2).

Figure 2 – A schematic of the skidpad event track (2017-18 Formula SAE Rules)
Designing the steering linkage for optimal tire positioning for the skid pad event will maximize grip for a 15.25m turn. This was achieved by iteratively adjusting steering geometry to achieve proper steer angles after accounting for tire slip angles. Orlando Ward of the tire department performed the tire selection and analysis using the Optimum T software (Tech Tip: Steering Geometry). By Newton’s Second Law,

\[ F = ma \]

“Force” in this case can roughly be translated drivetrain output, so minimizing mass will increase acceleration performance. Although the steering system is a small fraction of total car mass, every reduction is valuable when fractions of a second can separate cars in the acceleration event. The endurance race at competition is 22km long, so a car that is hard to turn will likely cause discomfort and fatigue, leading decreased driver performance. An uncomfortably positioned steering wheel will do the same.

1.2 DESIGN CONSTRAINTS

Ideality in practical vehicle design is unfortunately impossible and occasionally compromises must be made. Design constraints and parameters were identified to serve as a checklist of requirements and potential conflicts. These are listed below in Figure 3. Constant reference to this list helps ensure seamless integration and correct performance.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Competition Rules Compliance</td>
</tr>
<tr>
<td>2</td>
<td>Suspension Point Placement</td>
</tr>
<tr>
<td>3</td>
<td>Cockpit Template Clearance</td>
</tr>
<tr>
<td>4</td>
<td>Upright (Knuckle) and Rim Clearance</td>
</tr>
<tr>
<td>5</td>
<td>Production Capabilities</td>
</tr>
<tr>
<td>6</td>
<td>Time</td>
</tr>
</tbody>
</table>

FSAE publishes a new book of rules each competition year. The following rules specifically applied to the steering system during the 2017 season: T4.1, T4.2, T4.6, T5.8, T6.5, T11.1, T11.2 (2017-18 Formula SAE Rules). Suspension point placement is critical to overall vehicle performance. Poor steering point positioning can cause bump and/or roll steer, a condition where vertical tire travel or chassis roll cause the steering wheel to turn, leading to poor handling. The physical steering assembly must integrate into the chassis and not interfere with any structural members while also allowing the cockpit templates (Figure 4) to pass through unimpeded. Clearance between the rim and the steering post on the upright must also be checked for all possible suspension positions. The system needs to be manufactured in the quickest amount of time and for a reasonable cost without sacrificing quality. The MIT FSAE workshop is lucky to have CNC milling and turning equipment in house, so production capabilities are quite high. Timelines are also valuable in the process because they allow for project scheduling, task allocation, and milestone creation. A general project timeline is shown in Figure 5.
In addition to self-created goals and constraints, quantitative figures for system design were adapted from the literature. Most importantly, load cases and general knowledge from previous work were used in the design process. A good upper bound for input torque to the steering system from the driver is 100Nm, including a safety factor (Fox, Steven). This specification was used in all analysis calculations as the maximum possible driver torque input to any component in the system. Additionally, the maximum aligning moment on the tires during cornering was calculated using tire data to be 75ft-lbs.¹ Other component-specific loading scenarios were calculated and will be discussed in the analysis section.

¹ Tire data obtained from Formula SAE Tire Test Consortium (FSAE TTC)
1.3 SYSTEM VARIABLES

Several important vehicle variables affect the steering system, and must be considered when designing any car, especially an open wheel racer. These variables are listed in Figure 6 below.

<table>
<thead>
<tr>
<th></th>
<th>Kinematic Point Locations in XYZ Space</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>Caster Angle</td>
</tr>
<tr>
<td>3</td>
<td>Kingpin Inclination (KPI)</td>
</tr>
<tr>
<td>4</td>
<td>Scrub Radius</td>
</tr>
<tr>
<td>5</td>
<td>Mechanical Trail</td>
</tr>
<tr>
<td>6</td>
<td>Toe In/Out</td>
</tr>
<tr>
<td>7</td>
<td>Steering Ratio</td>
</tr>
<tr>
<td>8</td>
<td>Ackermann Percentage</td>
</tr>
</tbody>
</table>

Figure 6 – Table listing steering variables

Kinematic point locations are determined iteratively in the context of the vehicle as a whole to balance packaging and performance goals. Their coordinates determine vehicle handling characteristics. Caster angle is the angle between the steering (kippin) axis and the vertical, measured in the longitudinal (front-rear) direction. Caster angle influences straight-line stability and adds damping to the system, in turn affecting the feel of the car. High positive caster angles are usually avoided as they lead to stiff and unresponsive steering, but low or negative angles can make the system unstable, forcing the driver to fight the car to navigate turns or keep a straight line. The KPI is also the angle of the steering axis against the vertical, but measured in the lateral (side to side) direction. KPI has a similar effect on steering as caster angle and helps return the car to a neutral steering position following a turn. The scrub radius is the lateral distance between the tire contact patch and the intersection of the imaginary kingpin axis with the road. A large scrub radius will make the steering more difficult to turn and can increase tire wear, while no scrub will give the driver little feedback on steering inputs and will give the steering a “dead” feel. Mechanical trail operates similarly to scrub radius but is the longitudinal distance between the KPI intersection with the road and the tire contact patch. Figures 7 and 8 show how these angles are measured.

Figure 7 – Diagram illustrating caster angle and mechanical trail

2 Images sourced from http://www.car-engineer.com/suspension-design-definitions-and-effects-on-vehicle-behavior/
While caster, KPI, scrub radius, and mechanical trail are usually fixed once the suspension is designed, toe can be adjusted to add or remove straight line stabilizing effects. Toe in is usually used in FSAE to improve stability at the cost of some tire wear and increased steering effort (Figure 9).

The steering ratio determines how much the tires will turn with one turn of the steering wheel. Ackermann percentage quantifies the nonlinearity of the steering linkage. As a vehicle rounds a corner, the inner and outer wheels would ideally like to travel in concentric circles of different diameters, so the steer angles should be different. The more difference there is between the inner and outer tire steer angles, the more Ackermann the vehicle has. Most of these parameters must be determined iteratively and examined with intuition. There are no set rules or values for them, as various combinations can be made to work, although there are some general guidelines for acceptable ranges. (Milliken, Douglas L.) Figure 10 summarizes the values of these parameters for the 2017 MIT FSAE Electric vehicle.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Caster</td>
<td>3.5 degrees</td>
</tr>
<tr>
<td>KPI</td>
<td>3.3 degrees</td>
</tr>
<tr>
<td>Scrub</td>
<td>26.5 mm</td>
</tr>
<tr>
<td>Mech. Trail</td>
<td>13.5 mm</td>
</tr>
<tr>
<td>Toe</td>
<td>-2 degrees</td>
</tr>
<tr>
<td>Steering Ratio</td>
<td>4.2</td>
</tr>
<tr>
<td>Ackermann %</td>
<td>129%</td>
</tr>
</tbody>
</table>

Figure 10 – Summary of steering variable values for the 2017 system
1.4 SYSTEM ARCHITECTURE

The 2017 steering system was not designed in a vacuum: previous MIT designs, competitor’s systems, and other motorsports vehicles were researched to come up with a system architecture. The vast majority of FSAE steering systems are unpowered, and most operate with steering ratios usually ranging from 3-5 due to the style of autocross racing that the cars compete in (Farrington, Jock Allen). Tight, quick turns require the driver to rapidly turn the wheel, sometimes from lock-to-lock in a fraction of a second. Powered steering systems are heavy and add design complexity. Additionally, turning forces rarely exceed values that suggest the use of power steering since total vehicle mass is usually in the range of 200 – 350 kg.

System architectures vary across cars, but the most common implementation utilizes a steering rack and pinion coupled with a double u-joint construction to transfer steering input from the driver to the tires. A schematic of such a system is shown in Figure 11.

Teams also sometimes use other more exotic systems. These include 90-degree bevel gearboxes, the substitution of a rack with metal wires, direct-drive systems, pitman arms, and others. Figure 12 shows possible variations on more common designs.

Figure 11 – The double u-joint setup used in the 2016 car

Figure 12 – Left: A direct drive steering system that positions the rack above the driver’s legs.
Right: A sketch of a proposed pitman arm configuration

3 Image sourced from https://bradfordwsims.wordpress.com/past-projects/fsae-steering-system-design-and-manufacturing/
Steering supports come in various configurations, ranging from ball bearings to shafts in sleeves (Fox, Steven). Each approach has its drawbacks and strengths, but it is common for more inexperienced teams to overlook such details, even though they translate to serious differences in system performance. I chose to select a bearing to support the steering column instead of a bushing to minimize friction or any chance of binding. Ball bearings do cost more than bushings, but that cost can be justified by the improved performance.

Traditionally, the MIT FSAE team has used the most common double u-joint and steering rack design. Steering systems have been mostly successful in past years using this design, but have had their fair share of issues. The 2016 system was heavy, weighing over 5 kg including mounting. The steering rack chosen had to be very large due to an inefficient linkage design and suboptimal suspension point placement. The 2015 system suffered from excessive play, high driver input, and was heavy. There were also rules compliance issues that had to be resolved very late in the racing season.

Previous systems have used steel columns and steel u-joints. Although stiff in construction, detailed optimization calculations to save mass were never performed. The team has also always used an off-the-shelf aluminum steering wheel not specifically designed for our unique vehicles. With the 2017 design I wanted to challenge these assumptions to identify potential areas for mass reduction, increased system stiffness, smooth operation, and ergonomics.

Most high-performing European Formula Student teams utilize either a direct-drive steering system, where the steering wheel is directly coupled to the steering rack that is either mounted above the driver’s knees or is positioned very far in the front of the car, or a 90-degree bevel gearbox transmission. Most of them also adopt to use a rack and pinion, foregoing more exotic setups for a robust, reliable, and lightweight solution. The drawback to small and lightweight racks is their small lock-to-lock travel, which means steering linkage design becomes more difficult and involved. European teams have been performing at an elite level in the electric racing category since the inception of the competition. I chose to dig deeper into some of their designs to get an idea of what it would take to create a system that could rival theirs.

For our team, a direct drive solution would be nearly impossible to implement, based on our chosen suspension setup. The 2017 car uses a pushrod suspension, which essentially requires the springs, dampers, and bell cranks to be located on the top of the chassis unless the bell crank linkage is very cleverly designed. Many European teams that employ a direct drive steering system have a pull rod suspension setup that naturally positions the springs and dampers lower in the car. Our pushrod and bell crank setup leaves little room for a steering rack to be mounted on the top of the chassis.

---

A floor-mounted steering rack however lends itself nicely to be coupled with a bevel gearbox. A bevel gearbox also greatly simplifies packaging, as the entire column and shaft assembly becomes a right angle instead of a geometric linkage. There is also considerably less friction in a well-lubricated bevel gearbox than in a double u-joint setup. A double u-joint setup has a maximum operating angle above which the u-joints will not function. The angle is usually between 35 and 45 degrees. This constraint can make packaging difficult, possibly requiring sacrificing ergonomics or steering rack placement. These issues are not present with a bevel gearbox because the operating angle is 90 degrees. Lightweight and tough bevel gearboxes are hard to find, and minimizing mass is one of the foundational design goals of racecar construction. Additionally, manufacturing a custom bevel gearbox is a lofty and time-consuming endeavor that our team was not ready to undertake. After much searching, a candidate gearbox was identified. Figure 13 shows a cutaway view of the gearbox. It is specifically designed for use in a Formula Student competition, is rated to withstand 100Nm of input torque, and weighs in at just 0.45 kg. This gearbox is not only easier to design the system around, but it weighs 0.15 kg less than the 2 u-joints that were used in the 2016 design. This component is used by some of the most successful Formula Student teams in the world.

Figure 13 – Photo and drawing of the bevel gearbox used in the 2017 system

1.5 STEERING WHEEL ASSEMBLY
The steering wheel assembly was another area chosen for redesign. Instead of using an off-the-shelf wheel, I chose to manufacture a custom, experimental carbon fiber wheel. By designing it from the outset I was able to create an ergonomic design, a lighter assembly, and to add driver controls easily. The carbon wheel is designed specifically for racing, and in form resembles the high-performance steering wheels of Formula 1. Additionally there are 2 driver control buttons on the wheel: one to control an active aerodynamics package (DRS), and the other to activate launch control. Figure 14 shows a model of the completed assembly.
1.6 WRAP UP

In the following sections I will go into detail about specific aspects of the steering system, starting with the design and analysis, moving to manufacturing and assembly, and finally touching upon static and dynamic testing.

2. PRELIMINARY DESIGN AND CALCULATIONS

2.1 TORSION CALCULATION

To quantify potential changes in design, several calculations were performed to gain a better understanding into whether it makes sense to redesign certain parts of the system. First, a basic torsion calculation was performed to see if changing shaft material from steel to aluminum was possible given the maximum applied input torque. Both solid shaft and hollow tube geometries were considered. Equation 1 is the general form of the torsion equation for a shaft.

\[ T = \frac{J_T}{r} \tau \]

\[ \tau = \text{Shear strength, material property} \]
\[ T = \text{Applied torque, 100Nm of driver input (Fox, Steven)} \]
\[ r = \text{Shaft radius, desired unknown} \]
\[ J_T = \text{Torsion constant, geometric property} \]

Equation 1 can be easily rearranged to solve for the unknown radius. For a solid shaft,

\[ J_T = \frac{\pi r^4}{2} \]

And for a tube,

\[ J_T = 2\pi r^3 t \]

Where \( t \) is the wall thickness of the tube and \( r \) is the inner radius. Substituting for \( J_T \) and rearranging yields equation 2 for a solid shaft and equation 3 for a tube.
The smallest diameter shaft per material is summarized in Figure 15. Possible tube geometry options are summarized in Figure 16. Note: Composite shaft materials were not explored due to team inexperience with carbon fiber design, manufacture, and analysis.

<table>
<thead>
<tr>
<th>Material</th>
<th>Shear Strength [MPa]</th>
<th>Minimum Radius [mm]</th>
<th>Density [kg/m³]</th>
<th>Mass [kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>6061 Aluminum</td>
<td>207</td>
<td>7</td>
<td>2700</td>
<td>0.190</td>
</tr>
<tr>
<td>7075 Aluminum</td>
<td>331</td>
<td>6</td>
<td>2810</td>
<td>0.145</td>
</tr>
<tr>
<td>4130 Steel</td>
<td>335</td>
<td>6</td>
<td>7850</td>
<td>0.406</td>
</tr>
</tbody>
</table>

Figure 15 – Summarizes results of the torsion calculation for solid shafts. The mass value was calculated for a shaft 18 inches long, approximately the combined length of the column + shaft.

<table>
<thead>
<tr>
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<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>6061 Aluminum</td>
<td>207</td>
<td>3</td>
<td>5</td>
<td>2700</td>
<td>0.151</td>
</tr>
<tr>
<td>7075 Aluminum</td>
<td>331</td>
<td>3</td>
<td>4</td>
<td>2810</td>
<td>0.133</td>
</tr>
<tr>
<td>4130 Steel</td>
<td>335</td>
<td>2</td>
<td>5</td>
<td>7850</td>
<td>0.270</td>
</tr>
</tbody>
</table>

Figure 16 – Summarizes results of the torsion calculation for tubes. The mass value was calculated for a shaft 18 inches long, approximately the combined length of the column + shaft.

The results of calculation indicate that the most optimal choice for shaft material is 7075 Aluminum, and the ideal geometry is a tube. Steel shafts are poorly suited for this application due to their high density but similar shear strength. But, considering the added manufacturing complexity of turning a long hollow tube (~12 inches) it was decided to opt for a solid 7075 shaft geometry. The additional mass is negligible when taken in context of the system or vehicle as a whole.

2.2 BEARING LOAD CALCULATION

Since a ball bearing will be used to support the steering column, a bearing load calculation was performed to estimate the amount of loading that a bearing would potentially need to support. Figure 17 shows a free body diagram of the setup with both the radial and axial loading scenarios. Figure 18 summarizes the loads for several different column overhang lengths (ℓ).

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5 Values sourced from asm.matweb.com
Figure 17 – A free body diagram of the steering column used to calculate bearing reactions. The rollers on the left approximate the gearbox support and the pin joints on the right approximate the bearing. 2 pin joints are used because roller bearings are not rated to support moments.

The input forces, $F_1$ and $F_2$ are “worst case scenarios”. In the radial case, $F_2 = 1000$N roughly translates to a driver pushing/pulling sideways 100kg. (220lbs.) The axial loading of $F_2 = 1500$N translates to the total mass of a 100kg (220lbs.) driver braking at 1.5g with loose restraints. Shown next is the calculation of the bearing reaction forces.

\[
\sum F_x = F_a - F_1 = 0 \Rightarrow F_a = F_1 = 1.5kN
\]
\[
\sum F_y = F_2 - F_r + F_b = 0 \Rightarrow F_r = F_2 + F_b
\]
\[
\sum M_o = F_2\ell - F_b(L - \ell) = 0 \Rightarrow F_b = \frac{F_2\ell}{L - \ell}
\]
\[
F_r = F_2 + \frac{F_2\ell}{L - \ell}
\]

<table>
<thead>
<tr>
<th>Overhang Length ($\ell$) [in]</th>
<th>Shaft Diameter ($D$) [mm]</th>
<th>Total Length ($L$) [in]</th>
<th>Axial Reaction ($F_a$) [kN]</th>
<th>Radial Reaction ($F_r$) [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>20</td>
<td>8</td>
<td>1.5</td>
<td>1.33</td>
</tr>
<tr>
<td>4</td>
<td>20</td>
<td>10</td>
<td>1.5</td>
<td>1.67</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>12</td>
<td>1.5</td>
<td>2</td>
</tr>
</tbody>
</table>

Figure 18 – Summary of bearing reaction forces

Figure 19 shows the specification sheet for the bearing selected for the application. It can withstand a static axial loading of 2.5kN and a static radial loading of 8.5kN. Dynamic calculations were not performed because the bearing will not be subjected to continuous rotational operation. The bearing calls for a 20mm shaft diameter. A 7075 Aluminum shaft of 20mm can support an applied torque of 100Nm with a 1.6 safety factor.

---

$^6$ Calculation of maximum allowable loading was performed using the SKF online bearing calculator.
2.3 STEERING RACK AND GEARBOX SELECTION
Instead of purchasing a new steering rack, I opted to re-use an older model that was intended for an earlier car to save money. The Miltera M-Rack (Figure 20) is the lightest steering rack on the market, weighing a total of 0.43kg. The drawback of this rack is the relatively small travel (25mm). To deal with this issue, I chose to opt for a bevel gearbox transmission to simplify packaging. In fact, given our vehicle design, packaging with a u-joint setup would not be geometrically possible, as most u-joints cannot operate beyond 40 degrees. If a u-joint system were used, the front-end suspension points would need to be shifted, and likely driver positioning would need to be sacrificed for cockpit clearance. The bevel gearbox also positions the wheel in a more ergonomic position for the driver. Finally, the bevel gearbox assembly weighs 0.15kg less than a double u-joint setup.

Figure 19 – The datasheet for the bearing selected to support the column

Figure 20 – A CAD model of the Miltera M-Rack

7 U-joint system mass was taken from the 2016 design
2.4 GEOMETRIC STEER ANGLES

To get a realistic estimate for the minimum turning radius and ideal geometric steer angles required of the car, a hairpin turn sketch was created and is shown in figure 21. The angles are not indicative of actual steer angles since tire slip angles have not been taken into consideration. They are however a decent starting point for feasibility calculations. Figure 22 shows a skidpad sketch that approximates ideal geometric steer angles for the skidpad track.

![Figure 21 - Hairpin turn sketch with the shortest path yielding a minimum radius turn of 3.65m](image)

![Figure 22 - Sketch estimating ideal geometric steer angles for the skidpad track](image)

The geometric estimations were confirmed using Equation 4 (Thompson, Dale).

\[
\begin{align*}
\theta_{\text{outer}} &= \frac{WB}{R + T/2} \\
\theta_{\text{inner}} &= \frac{WB}{R - T/2}
\end{align*}
\]

Eq.4
WB = Wheelbase, 1.524m for the 2017 car
R = Turn radius to the car center, 3.65m for hairpin, and 8.2m for skidpad
T = Track, 1.219m for the 2017 car

Geometric steer angles obtained using equation 4 are summarized in figure 23.

<table>
<thead>
<tr>
<th>Wheel</th>
<th>Hairpin</th>
<th>Skidpad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner</td>
<td>29 degrees</td>
<td>11.5 degrees</td>
</tr>
<tr>
<td>Outer</td>
<td>20.5 degrees</td>
<td>10 degrees</td>
</tr>
</tbody>
</table>

Figure 23 – Table summarizing the ideal geometric steer angles

2.5 SLIP ANGLES AND TIRE DATA
Geometric steer angles are helpful for initial design considerations but are inadequate for final linkage design. Tire slip angles must be taken into account when finalizing the linkage geometry. Cornering loads impose lateral forces on the tires that either positively or negatively impact physical steering angles. For neutral and understeer vehicles, lateral loading of the tires imposes a counteracting moment on the tires and will steer the car out of the turn. Oversteer vehicles do the opposite; a lateral load will help the car steer into the turn. In a turn, depending on the loading of the wheels, the trajectory of the vehicle and the trajectory of the tires will be offset by a slip angle. Figure 24 shows a diagram of this phenomenon. The physical steering angle, or the angle that the wheels must in reality be able to turn on the car to navigate the turn at a given acceleration is a difference between the ideal geometric angle and the tire slip angle for oversteer vehicles. This difference is approximated as half of the tire slip angle, but varies with CG location and other vehicle dynamics parameters. (Milliken, Douglas L.) Since the 2017 vehicle is built as an oversteer machine (55% rear weight bias) slip angle effects will help the car steer into corners.

Figure 24 – A graphical representation of the slip angle for the outer front right tire of a car navigating a left turn. (Milliken, Douglas L.)
The tire curve for the selected tires for the 2017 vehicle is shown in figure 25. In full cornering the outside tire is under maximum loading, and is thus operating at about a 6-degree slip angle. For the 2017 system, slip angle effects will likely not deviate by more than half of the slip angle from geometric angles so “geometric angle + 3 degrees” was chosen as an appropriate upper bound for physical steer angles. Steering optimization only focused on the slip angle and loading of the outer tire, which does most of the cornering work. That is not to say that inner tire angles were ignored, but they were given a wider range of acceptable values since they will not be loaded as heavily and thus would generate smaller torques.

If the slip angle at a given loading is known, it can be used to optimize the steering linkage to make the tires operate at optimal slip angles during turns, increasing grip. As the vehicle corners, the slip angle of the tires will be oversteering the car in the corner. It is almost impossible to completely optimize tire performance through all turn radii and tire loading conditions, so it was chosen to focus on the outer tire on the skidpad track. Additionally, since slip angles are dynamic variables that change based on a variety of factors and are unique to all 4 tires, it is difficult to quantify exactly how they will affect performance until the car is driven on the track.

2.6 DRIVER EFFORT CALCULATION
Driver effort can be back calculated using maximum tire aligning moment. The aligning moment can be determined by analyzing tire data, and in our case peaked at 75ft-lbs (Figure 24). This data was obtained by Orlando Ward in the tire department using mass transfer calculations and tire compound information. Using this value and other known quantities, an estimate for driver steering effort at maximum tire loading can be calculated and serve as an upper bound. Figure 25 shows a free body diagram and the equation derived for driver input force per arm.
Figure 24 – Tire aligning moment as a function of tire slip angle generated with tire data.

Figure 25 – Free body diagrams of a simplified steering linkage and steering wheel. The lower FBD can be imagined to connect to the center of the rack gear.

\[ \tau_{tire} = \text{Tire aligning torque, 75ft-lbs} \]
\[ \tau_{input} = \text{Driver torque input, equal to } \tau_{tire} \]
\[ r_{gear} = \text{Radius of rack gear, 0.5in} \]
\[ L = \text{Steering arm length, 2.4in} \]
\[ l_{steer} = \text{Length of steering rack + Tie rod, 21.5in} \]
\[ r_{wheel} = \text{Steering wheel radius, 4.5in} \]
\[ F_{rack} = \text{Rack reaction force} \]
\[ F_{tire} = \text{Tire reaction force} \]
\[ F_{input} = \text{Driver input force from 1 hand} \]

\[ F_{input} = \frac{r_{gear}}{2r_{wheel}l_{steer}} \tau_{tire} = 91N^8 \]
2.7 STEERING POINT LOCATIONS
A final obstacle to linkage design is the influence of steering kinematic points on overall vehicle handling. Both the location of the steering rack and the location of the outboard steering points have significant effects on suspension design, mainly in the form of bump and/or roll steer. Bump/roll steer is highly undesirable in any vehicle, as it would cause the car to turn during suspension travel or chassis roll without driver input. This is due to geometric misalignments between the steering linkage and suspension arm travel arcs. Using advanced software like WINGEO allows our team’s suspension department to iteratively adjust point locations to mitigate such unwanted driving effects. The suspension department can get instant feedback when changing suspension kinematics and make required adjustments as needed.

2.8 2D LINKAGE SKETCH
Next, a preliminary 2-D steering linkage was modeled in SolidWorks to approximate real inner and outer steering angles taking packaging concerns into consideration. It also served as a preliminary indicator of design feasibility. The preliminary sketches are shown in figures 26 and 27. These sketches are 2-dimensional approximations that do not account for 3D geometric or dynamic effects.

![Figure 26](image1.png)

Figure 26 – The sketch used to geometrically approximate steer angles shown here in the minimum radius configuration. The outer wheel will be operating at optimal geometric angles while the inner wheel will be experiencing an understeer torque, an undesirable effect but unavoidable in this linkage configuration.

![Figure 27](image2.png)

Figure 27 – The sketch used to geometrically approximate steer angles shown here in the skidpad configuration. The outer wheel will be operating at optimal geometric angles while the inner wheel will be experiencing an understeer torque, an undesirable effect but unavoidable in this linkage configuration.

* Equal to lifting about 9kg (20bs) in each arm
3. DETAILED DESIGN

3.1 3D CAD MODEL
Having validated the feasibility of the linkage design, a 3D CAD model of steering components was constructed and integrated with the suspension, wheel package, and chassis models (Figure 28). More accurate steer angles can be predicted using this higher order model. It allows for more design iteration and greater performance optimization. Additionally, a 3D model will account for non-planar effects to the linkage including camber angle and suspension travel. Using this model, linkage design was finalized. Figure 29 summarizes the parameters of the 2017 linkage. While cornering, the outer wheels of the vehicle are receiving the highest loading due to mass transfer. Optimizing outer wheel steering will yield the highest benefits for vehicle cornering performance. The 2017 steering system was optimized primarily for maximizing outer tire grip during the skidpad event. Secondary goals were to maximize grip in hairpin turns.

![Figure 28 – Two views of the 2017 steering system CAD model](image)

<table>
<thead>
<tr>
<th>Wheel</th>
<th>Hairpin</th>
<th>Skidpad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner</td>
<td>25.5 degrees</td>
<td>10.7 degrees</td>
</tr>
<tr>
<td>Outer</td>
<td>23.6 degrees</td>
<td>10.5 degrees</td>
</tr>
</tbody>
</table>

Figure 29 – Final achievable physical steer angles for the 2017 system

3.2 ERGONOMIC WHEEL POSITIONING
To position the steering wheel ergonomically, a frame jig was constructed to approximate driver posture in the chassis. This jig was used to position the steering wheel in a comfortable location relative to the front and rear roll hoops (Figure 30).
3.3 COMPONENT CONNECTIONS

Connections between components in the steering column and shaft assembly needed to withstand required input loads. Although the components in question (quick release coupler into the steering columns, and the steering rack coupler into the steering shaft) were machined to a 0.002" interference fit and hydraulically pressed into each other, a shoulder bolt mechanism was adapted to provide additional safety and redundancy to the system (Fox, Steven). (Figure 31)

Figure 31 - A close-up view of the shoulder bolt mechanism used to secure the components together. The bolt is rated to a tensile strength of 140,000psi.
3.4 WRAP UP
Finally, template, chassis, and suspension clearances were checked for all possible orientations to ensure no interference. A minimum of 0.1 in of clearance between components is used to allow for some inevitable manufacturing errors. This also allows for experimentation using a purchased carbon fiber tie rod that has a larger outer diameter than the steel one currently used. Rules compliance was also checked.

4. ANALYSIS

4.1 FEA
FEA analysis was performed on all components to validate calculations and check for safety. The figures below show the results of the analysis on the different steering components.

![Figure 31 - Torsion FEA on steering column applied at the connection to the steering wheel and fixed at the connection to the gearbox](image)

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9 Analysis was not performed on purchased components when specifications were provided by the manufacturer. These include the steering rack, bevel gearbox, and quick release assembly.
Figure 32 – Torsion FEA on steering column applied at the connection to the gearbox and fixed at the shoulder bolt connection.

Figure 33 – Close up view of figure 32
Figure 34 – FEA of the steering column in bending, applied at the interface of the quick release coupler and fixed at the bearing surface.

Figure 35 – FEA of the column support, fixed at the top surfaces welded to the chassis with an applied load of 1500N on the bearing surface.
Figure 36 – FEA of the column support due to a sideways loading of 1500N applied at the bearing surface.

Figure 37 – FEA of the gearbox support mount subjected to a 100Nm input torque about the axis of the vertical bevel gear.
Figure 38 – Torsion FEA of the steering shaft, applied at the gearbox connection and fixed at the shoulder bolt connection.

Figure 39 – Torsion FEA of the steering shaft, applied at the steering rack coupler surface and fixed at the gearbox connection.
Figure 40 - FEA of the steering rack clevis, an applied 1500N load at a 10 degree offset to the spherical bearing interface, fixed at the bolt location.

Figure 41 - FEA of the chassis mounting tab, fixed at the weld locations and a 1500N applied load across the steering rack interface.
4.2 CARBON WHEEL TESTING

To analyze the carbon wheel, a prototype was manufactured and physically loaded to simulate worst-case scenarios. The two failure modes examined were a driver input torque of 100Nm and a frontal loading of 350lbs, supporting an unrestrained driver under full braking. The experimental setups are shown in the Figure 42.

Partial yielding of the core material was observed in both experiments, but no damage to the outer carbon fiber shell was observed. Partial yielding occurred at 125Nm of applied torque and 600lbs of frontal loading.

5. MANUFACTURE AND INTEGRATION

5.1 MACHINING

Components that were manufactured in-house include the steering column, steering shaft, steering wheel and grips, and chassis mounting. The steering rack, quick release, and bevel gearbox were purchased off-the-shelf. The steering shaft and steering column were turned on a lathe and milled to create the splined connections. See Figure 43 for a picture of the machining setup used. The rack mounts were milled, and the chassis mounting and steering wheel were waterjet cut. Figures 44 to 51 show component drawings. Drawings are not to scale and all dimensions are in inches.

Figure 43 – Lathe setup for turning steering column and shaft. Splining was done on a 4-axis mill.
5.2 COMPONENT DRAWINGS

Figure 44 – Drawing of the steering column

Figure 45 – Drawing of the steering shaft
Figure 46 – Drawing of the column support

Figure 47 – Drawing of the gearbox support
Figure 48 – Drawing of the steering rack coupler

Figure 49 – Drawing of the steering rack mounting

Figure 50 - Drawing of the steering rack clevis
Figure 51 – Full assembly drawing
5.3 JIGGING AND ALIGNMENT
To align the system relative to the other suspension points, a positioning jig was constructed. The steering chassis tabs were attached to the jig, then welded to the frame. Next, the steering assembly was bolted to the chassis, and aligned by measuring the level of the gearbox. After welding the gearbox support plate, the column support was aligned and welded. Figure 52 shows the jigging setup. Careful manual grinding made precise adjustments to the geometry of the mounting pieces to account for chassis manufacturing errors. Finally, template clearance and rules compliance were checked.

Figure 52 — The jigging setup for the 2017 system

6. TESTING, VALIDATION, AND RESULTS
Once assembled, the system was ready for testing. Although dynamic testing cannot be done until the vehicle is fully operational, static tests indicate this steering design to be one of the most successful in the team's history. There is very little backlash or compliance in the system, the wheel positioning and shape is comfortable, and very little effort is required to turn the steering wheel from lock to lock when the car is on the ground. Only 1 finger is required to spin the wheels from lock to lock when the nose is off the ground.

Dynamic testing on the track will truly indicate whether this system is able to perform at the level required. In preliminary, low speed testing, the system functions as expected and is able to navigate minimum radius turns. It is expected to perform equally well at higher speeds, but that capability will need to be confirmed.

Another portion of the competition tasks teams with costing the production of their vehicle. The itemized BOM for the steering system is given in figure 53. Each item on the BOM also has a separate itemized part cost sheet detailing materials and processes used in manufacture, although those are not included for brevity. The final cost of the 2017 steering system is $305.88, lower than the previous 2 systems, but slightly higher than other teams. The performance benefits provided by this system, however, justify the increased cost.

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## 7. CONCLUSION

The 2017 MIT Formula SAE steering system embodies the essence of the engineering design process. Starting with hand calculations, geometric sketches, and literature analysis, and using high-level goals, a preliminary system layout was envisioned. Then, using computer modeling and simulation software, the initial design was verified and integrated into the overall vehicle model. Finally, the system was manufactured and tested to validate calculations and simulations before physical integration into the car.

There are areas for potential future improvements in an FSAE steering system. More advanced simulation of slip angle effects can be made to better understand the requirements of the steering linkage imposed by the tires. Future system designs should explore the use of composite steering columns to save mass and increase stiffness. More sophisticated bearing analysis can be performed, and a more suitable bearing arrangement can be chosen for the application, for example, a combination needle-roller bearing. Instead of using a 2D linkage sketch, a 3D linkage should be modeled from the beginning. A 2D setup does not give very precise estimates for steer angles, and a 3D model much better approximates the final system. The 2017 system was optimized for the skidpad event, but future designs could optimize for the endurance competition by designing the linkage to perform best on that specific track. Finally, the jigging setup used for the 2017 system is less than ideal. A more sophisticated welding fixture should be designed to avoid any misalignments.
8. References