Honors Research Projects

The Dr. Gary B. and Pamela S. Williams Honors College

Spring 2015

2015 Zips SAE Baja Brakes and Throttle System

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The University of Akron
SAE Baja Zips Off-Road Racing Team
2015 Brakes & Throttle System

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May 2015

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Executive Summary

The SAE Baja student design team at The University of Akron is one of the longest-standing design teams at the university. The purpose of this team is to design, manufacture, test, and race an off-road vehicle within the guidelines of competition established by the Society of Automotive Engineers. The vehicle is made up of a small number of subsystems including frame, drivetrain, suspension, steering, and braking. The following will discuss all aspects of the design process of the braking and throttle system of the 2015 Zips Baja car. This process includes several steps and considerations such as design goals, system performance calculations, stress analysis, and manufacturability. Once the final design is established, the manufacturing process will be outlined. The performance of the finished system will be analyzed and the necessary changes will be made to optimize design, and therefore, performance.

Baja SAE Guidelines and System Requirements

The Society of Automotive Engineers lays out an extensive document of rules and guidelines for the SAE Baja competition. These rules cover all aspects of the vehicle including frame tube thicknesses, safety equipment specifications, allowable materials, and many other details. Much of this document is created for the purpose of driver safety, but the level of regulation given to even the smallest details of the car also serve as a test of engineering precision and attention to detail during the design process. There are several rules and specifications that are directly related to the brakes and throttle systems. The rules pertaining to these systems are the following (The braking section of the official 2015 Baja SAE Series Rules document can be found in Appendix D):

THROTTLE:

- Only mechanical foot-operated pedals are allowed.
- A wide-open throttle stop must be incorporated at the pedal.
- Controls must be designed to return to idle-stop in the event of failure.
- Throttle cable must be sheathed between its forward mounting and the firewall.
- Foot pedals must be positioned so as to avoid foot entrapment in any position.

BRAKES:

- Vehicle must have hydraulic braking system that acts on all wheels and is operated by a single foot pedal.
- Pedal must directly actuate the master cylinder through a rigid link.
- System must be capable of locking all four wheels in both static and dynamic situations on both pavement and on unpaved surfaces.
- System must be segregated into at least two independent hydraulic circuits with their own fluid reservoirs.
- Brakes on the driven axle must operate through the final drive.

- All brake lines must be mounted securely and not fall below any portion of the vehicle (frame, swing arm, A-arms, etc.)
- Vehicle must be equipped with a red LED brake light that is SAE "S" or "U" rated as marked on the lens. Light must be completely illuminated when brake system is actuated and completely extinguished when brake is released.
- Brake light must be mounted minimum of one meter above the ground and must be mounted such that it shines parallel to the ground.

All of these criteria will be extensively checked at the technical inspection that is performed on all cars at each competition. If any of these criteria are not met, the inspection will be failed and the team will not be able to compete unless the necessary changes are made.

There is also an event at the competition that is used to directly test the braking system. In this dynamic brake test, the driver must bring the vehicle to its maximum speed, then engage the braking system to cause all four wheels to lock up. In this test, there is one judge assigned to watch each wheel (four total judges). If any of the wheels continue to spin for any amount of time after the brake system has been engaged, the test is failed. This event is particularly important because no other competition events can be attempted until the brake test is successfully completed. Since there is only one day to complete dynamic events such as suspension test, maneuverability, acceleration, and tractor pull, the time lost with unsuccessful brake tests can lead to "Did Not Finish" status on events leading to zero points for the team. If the brake test is not passed at all, none of these events nor the endurance race can be completed.

Brake and Throttle System Overview

The essential purpose of the brake and throttle systems of the vehicle are extremely simple. The throttle needs to make the car accelerate at the driver's will, and the brake system needs to make the car decelerate at the driver's will. However, the overall capability of the system transcends this simple definition. For example, a low quality braking system will only be able to achieve "no brake" and "full brake." A quality braking system must be able to achieve intermediate ranges of braking that allows the driver to feather the brakes when moving around corners and when approaching a ramp-type obstacle that could send the vehicle airborne.

Other considerations that must be taken into account in a braking and throttle system is are ergonomics and component interference. In terms of ergonomics, the driver must be able to comfortably use the brake and throttle pedals for an extended period of time. The endurance race at the Baja SAE competitions are four hours long, and poorly placed pedals can cause discomfort for extended periods of time. Component interference is also a concern. The pedals and brake assembly features located in the front box must share space with steering system components such as the rack and pinion and the steering column. The rotors and brake calipers located at the wheels can interfere with the connection points of suspension components, steering components, and even with the wheel itself. If potential interferences are not identified during

the design stage, there can be serious problems and significant lost time during the manufacturing stage to make components fit together. This is one of the biggest testaments to the importance of communication between designers of different vehicle components and the absolute necessity of three-dimensional computer modelling of the design.

Major System Components

A typical braking system is made up of only a few main components. These components are the pedal, master cylinder, brake reservoir, brake lines, rotors, and calipers. The throttle system is made up of the pedal, throttle cable, mechanical stopping device, and return spring.

Simple Brake/Throttle System

The main premise of a hydraulic braking system is very simple: driver force applied at the brake pedal is transferred hydraulically to locations at the wheels or axle to reduce rotational speed and therefore vehicle speed. This simple task is completed using the short list of components given in the previous section.

Beginning the process from the driver's effort to push one end of the brake pedal, the movement of the pedal actuates the plunge rod that is attached to the other end of the pedal. This plunge rod moves a plunger inside the master cylinder and decreases the volume within the cylinder, increasing the pressure of the fluid inside. Figure (1) below shows the relationship between the foot force location and the actuation of the push rod.
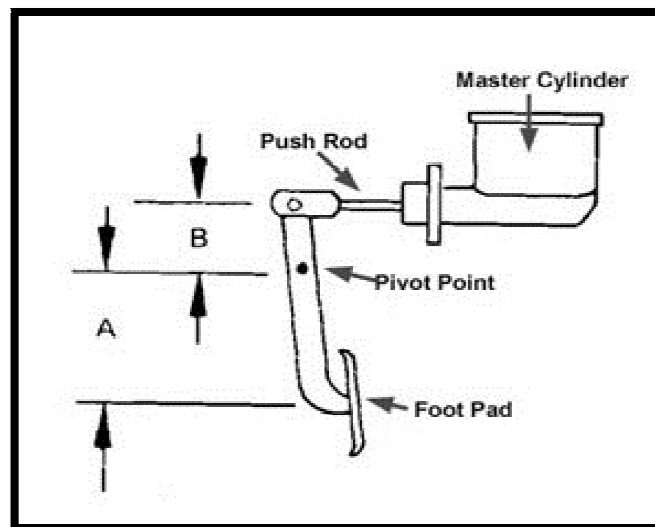
Figure 1: Basic Pedal Diagram.

As seen above, the driver force is applied at the foot bad. The pedal pivots about the pivot point and the push rod is pushed further into the master cylinder, decreasing the volume occupied by the fluid and therefore increasing the pressure.

Since pressure within the entire system is essentially uniform, the pressure increase in the master cylinder is also experienced in the brake all the way to the calipers. Figure (2) shows the path of fluid between the master cylinder and the calipers of a basic brake system.

Figure 2: Basic brake system fluid route.
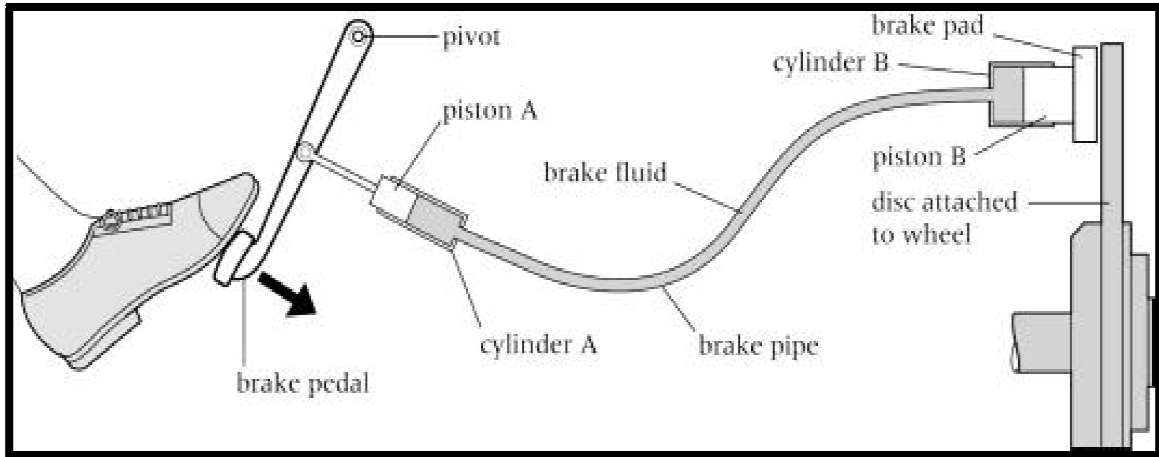
Within each brake caliper there are small piston/cylinder assemblies (the amount of pistons in a caliper varies between models and manufacturers). When the fluid pressure increases in these small cylinders (shown as "cylinder B" in Figure (2)), the pistons travel and push on the brake pads, causing them to push onto the brake rotor from both sides. The friction force created between the brake pads and the brake rotors cause the deceleration of the rotors, which are rigidly attached to either a wheel or a drive shaft. The inner workings of this caliper/rotor assembly are displayed well in Figure (3):
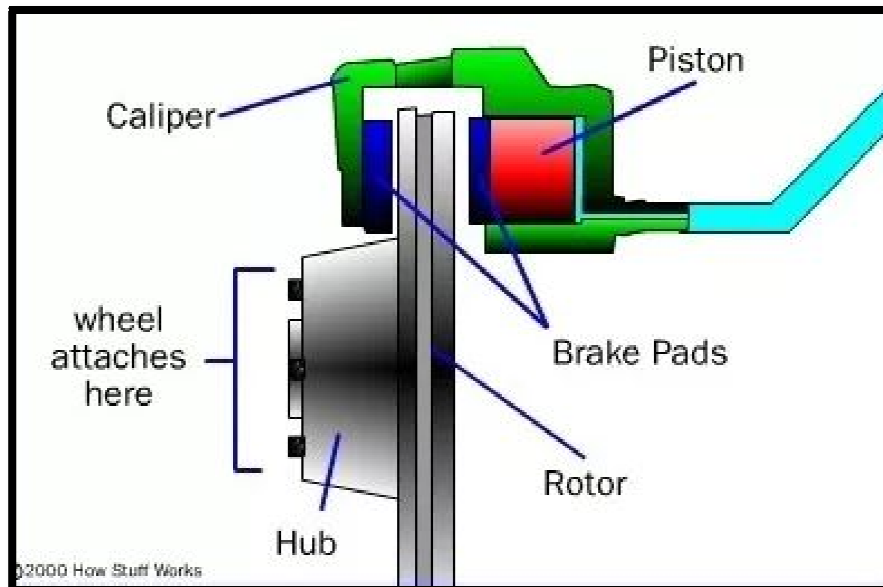
Figure 3: Rotor   Caliper / assembly.

The throttle system is even simpler than a basic braking system. The gas pedal and the throttle lever on the engine are attached via a cable. When the gas pedal is pushed forward, the cable pulls the engine throttle to an open position. When force is removed from the gas pedal, a return spring attached to the engine's throttle arm pulls the arm back to the idle position.

2014 Vehicle Performance Review

The 2014 Zips Baja season was the most successful campaign for the team in almost ten years. However, problems with several components of the car combined to leave a sense of underachievement based on the overall potential of the car. The brake system itself did not experience any failures. Although it took several attempts to pass the brake test at the first competition due to a lack of system testing (the building of the car was finished and assembly of the brake system was done at the competition itself), the test was passed at both competitions. The throttle system experienced more difficulty due to issues with the throttle cable becoming kinked and not returning into the cable sleeve properly.

Although the brake system did not fail, the design was by no means perfect. Many issues needed to be corrected to optimize design and allow the team to move into the upper tier of competitive teams. The biggest issue is that the system was overdesigned in terms of size in almost every way. Rotors, pedals, tabs, gussets, hardware, and cylinder-mounting components were all oversized and were able to endure much more force and many more cycles than the system would ever see. Strength is not a bad thing, but overdesign can take away from other important aspects of the vehicle such as weight and ergonomic considerations.

The issues with the throttle cable also need to be addressed. Throttle problems that potentially lead to not running the engine at full throttle are extremely costly, considering the engine is stock and governed at only ten horsepower. There have been acceleration competitions in the past where the difference between first place and fiftieth place has been only 0.3 seconds. Achieving less than full throttle during this competition would clearly be an enormous disadvantage.

2015 System Design Goals

The most basic level of design of the new vehicle begins with a few main parameters of the car such as weight, wheel base, track, ride height, etc. Table (1) on the next page shows the 2015 design goals compared to the actual values of the 2014 vehicle.

Parameter	2015 Vehicle Goal	2014 Vehicle Actual
Total Weight	360 lbs.	470 lbs.
Front Track	52"	52.4"
Rear Track	48"-50"	43.3"
Wheel Base	60"	59.2"
Ground Clearance	11.5"	12.5"
Tire Diameter	22"	22"

Table 1: Comparison of 2014 actual vehicle parameters and 2015 goal parameters.

It is nearly impossible to facilitate all subsystem needs while maintaining these goal parameter values. For example, the ideal wheel base for the suspension system may not be the same as the ideal wheel base for the steering system. However, these basic goals serve as a "best prediction" for the final vehicle properties, which is needed in order to begin calculations and analysis used in the design of all subsystems.

The main overall design goals of the 2015 vehicle are based on weight reduction, system integration, and an earlier manufacturing completion date. Weight is by far the most important characteristic in terms of how competitive a vehicle will be. Since every team must run the same stock engine, lighter cars will obviously be able to accelerate faster and have a higher top speed (assuming the drive train is appropriately designed).

System integration is an issue that has been brought up to the team during feedback sessions with competition design judges for several years. The consensus among these engineers has been that it is readily apparent that different individuals designed the different components of the car. For example, the vehicle appears to be five completely independent subsystems (frame, drive train, suspension, steering, and brakes/throttle) that have been bolted together. Instead of this, a well-designed car will look more like one complete vehicle than a group of subsystems. System designers must work together to create more system crossover such as the sharing of mounting tubes and tabs between multiple systems and make design compromises to make the best use of space. These spatial design considerations can also potentially lead to an overall reduction is vehicle size, and therefore vehicle weight.

The manufacturing completion date is also a very important characteristic of the car. Missed design and manufacturing deadlines have plagued the team for many years and has led to habitual mediocrity. Not only do late finishes take away from valuable testing time, but rushed work with competitions looming can also lead to careless manufacturing mistakes. These mistakes can lead to component failure during competition and can also be difficult or impossible to correct between competitions when there is additional time to work on the vehicle. Multiple examples of this have occurred in the past couple of years. In 2014, the guard system for the powertrain was fabricated on-site at the first competition and the result was a puzzle of sheet metal and fasteners that took almost an hour to remove.

When the CVT (Continuously Variable Transmission) had to be adjusted during the four-hour endurance race, almost two hours were lost from simply removing and reassembling the powertrain guard. The 2013 car featured a unique manual transmission that was connected to the engine for the first time at competition. The lack of troubleshooting time led to the drive having to be welded into a single gear for the entire competition. When the competition was over, the damage to the gear and other components was not correctable. These examples show how important it is to set ambitious deadlines for design and manufacturing and to hit these deadlines at all costs. In some cases this meant using purchased components as opposed to custom and also trusting elements of the 2014 car that proved to be reliable.

The overall design goals of the vehicle became the focus of the braking system design goals. The primary goal of the system is to reduce as much weight as possible while still being able to perform the brake test and other events without failure. This would mean decreasing the size of almost all components, especially the rotors and component mounting materials. The biggest potential weight decrease would be from changing the rear braking system to a single inboard caliper compared to two outboard calipers. Although the size of the inboard rotor and caliper would be greater than that of the individual outboard units, the inboard system would be much lighter than the combined rear units.

The main design goals of the system are the following:

- Use single inboard rear brake assembly.
- Use thinnest rotors allowed according to caliper manufacturer specifications.
- Lower profile of assembly into front box further out of driver's vision.
- Mount system without adding any tubes to the frame.
- Customize pedal placement to facilitate driver comfort at most common position (full throttle, no brake).
- Mount proportioning valve in a location easily accessible to the driver.
- Achieve partial braking ability (minimum of approximately 0%, 33%, 67%, and 100%).
- Make direct line of action for the throttle cable (no tight radii in cable sleeve between pedal and engine throttle arm).
- Route brake lines such that they are inconspicuous and protected from moving elements such as suspension components, tie rods, and the driver.

Component Design and Selection

System components such as the calipers, master cylinder, and rotors must be able to provide enough braking force to pass the static and dynamic brake test, as well as provide the response that the driver needs to race competitively. However, there are too many unknown parameters in the system to choose every system characterisic based on calculations alone. In order to reduce the number of unknown paramters, a small number of design decisions and component selections had to be made prior to the calculation process.

Deciding whether a certain system characteristic would be chosen prior to calculations is a tedious process. Initial selections were made based on available components that could best satisfy the design goals of weight, ergonomics, and reliability. Based on these considerations, the front caliper model, rotor diameters, and master cylinder were selected. These decisions are then used to calculate unknown system characteristics and make the remaining design decisions. At that point, the initial decisions can be verified and iterated if necessary. The next few sections outline the design and selection rationale of these system components.
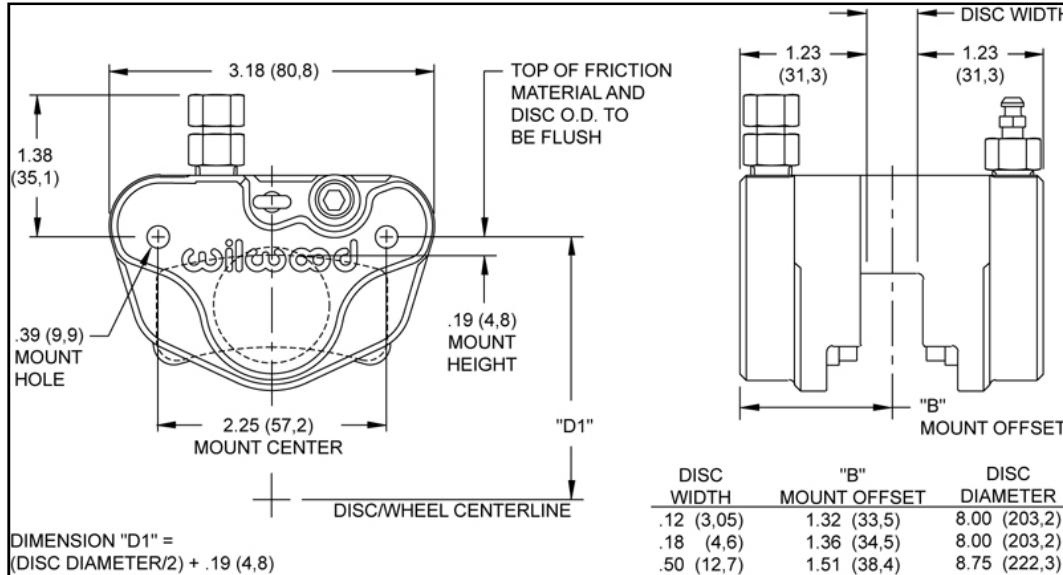

INITIAL COMPONENT SELECTION AND DESIGN

ROTORS

Rotors were a logical component to design in the early stages of the total system design because the range of possible diameters are already limited by other parts of the vehicle. Because the front brake circuit is an outboard system, the assemblies dwell inside the rims of the wheels. The rotor diameter is limited by the inner diameter of the rims and the clearance that the front calipers will need to fit inside the rim with the possibility of a reasonable amount of rim deformation. Within this upper diameter limit, it was a matter of braking ability versus weight savings. A larger diameter rotor would be able to apply a bigger moment to the wheel and provide more braking ability, but a smaller diameter would weigh less. The weight savings were initially assumed to be insignificant compared to providing maximum braking ability, which could be the difference between passing and not passing the brake test at competition. When the front caliper model was selected, the exact weight savings by using smaller rotors could be calculated based on the minimum rotor diameter allowed by the caliper manufacturer specifications (after calipers were selected, the total weight that could potentially be saved by using the minimum diameter was 0.12 lbs. per rotor, which was considered negligible compared to the increased braking ability provided by the larger diameter).

The maximum diameter of the rear rotors were also limited by other components of the vehicle. Since the rear circuit was to utilize an inboard brake system, the brake caliper needed to be mounted to the gear box. The diameter of the rear rotor was limited by the highest location (in the elevation plane) on the gearbox that could be used to mount the rear caliper. Because the design of the gear box and the design of the brake system took place simultaneously, estimates

had to be made by the braking and drivetrain designers as to the final dimensions of the gearbox. As design progressed and more final estimates could be made, the dimensions were updated such that the rear rotor would have the largest possible diameter. The decision to use the largest possible diameter rotor as opposed to a smaller, lighter one was made based on braking ability and the overall weight savings that were made from using an inboard system compared to an outboard system. Although single inboard rotor is larger than each individual outboard rotor from the 2014 car, the inboard rotor is much lighter than the outboard rotors combined. Figure (13) in Appendix (A) shows the weight comparisons of various components of the system between the 2014 and 2015 cars.

FRONT CALIPERS

The initial selection of the front brake calipers were made solely based on reliability and weight considerations. The Wilwood PS-1 caliper has been used on the Zips Baja car for three consecutive seasons and has performed well each season. After cross-checking different models for a product that was significantly lighter, the PS-1 (0.92 lbs. each) was ultimately selected.

Figure 1: Wilwood PS-1 caliper.

This particular model is available with three different bore diameters. Based on braking ability, the largest available bore was initially selected, since a given system pressure will result in more force applied to the rotor with a larger diameter piston. Similar to the rotor diameters, this component selection was re-evaluated and iterated after calculations were executed and overall vehicle design progressed.

REAR CALIPERS

The initial selection of the rear calipers were made primarily based on mounting ability. In the early stages of car design, it was unsure whether the caliper would be mounted on the right or left side of the gearbox. This created a need to find a universal caliper that could be mounted in either orientation so that other components of the car did not have to design around the need for the rear brake assembly to be on a certain side. For ease of manufacturing of the gearbox, it is advantageous for the rear caliper to have a flat face on which it can mount directly to the gearbox without the need for spacers or other parts.

Extra parts not only take more time and money to manufacture, but a tolerance stack-up of multiple parts could lead to rotor misalignment in the caliper. With these design considerations in mind and also the desire to remain with one manufacturer for the calipers and master cylinder, the Wilwood Billet Go-Kart caliper was selected.


Figure 2: Wilwood Billet Go-Kart caliper.

MASTER CYLINDER

The major decision that has to be made when selection a master cylinder is whether to use separate cylinders for the front and the rear circuits or to use one tandem cylinder that serves both. The Baja SAE Rules document states that the front and rear circuits must be completely independent of each other so that if a leak should occur in one circuit, the other one will remain in working order. Both types of systems fulfill this requirement, but the pros and cons of each have to be considered before making a selection.

The first type of system is one with two separate cylinders. This type of system has historically been the most common at past SAE Baja competitions. These cylinders can be very small with reservoirs mounted directly to the top. This is advantageous in terms of weight and ergonomics. One disadvantage is the need for a bias bar system. Since weight is transferred during deceleration, there is generally a need for different braking abilities in the front and rear circuits, leading to the need for different pressures. Since there is only one brake pedal, there must be a way to adjust the amount of pressure being created in each circuit. A bias bar is essentially an adjustable linkage between the brake pedal and the two cylinders. The bar can be angled such that more or less of the overall pressure created by the stroke of the foot is made in each cylinder. Disadvantages of this assembly include difficulty of adjustment, space consumption, and creating extra system components that can be broken during competition.

The second type of system is one with a single tandem cylinder. Although there is only one cylinder, it houses two separate chambers, each with their own reservoir inlet and circuit outlet, that are pressurized by the same plunge rod. Figure (3) shows a schematic of the inside of a tandem master cylinder.
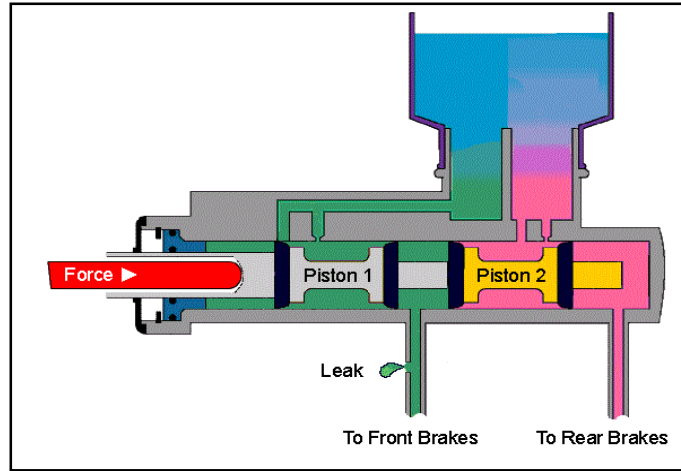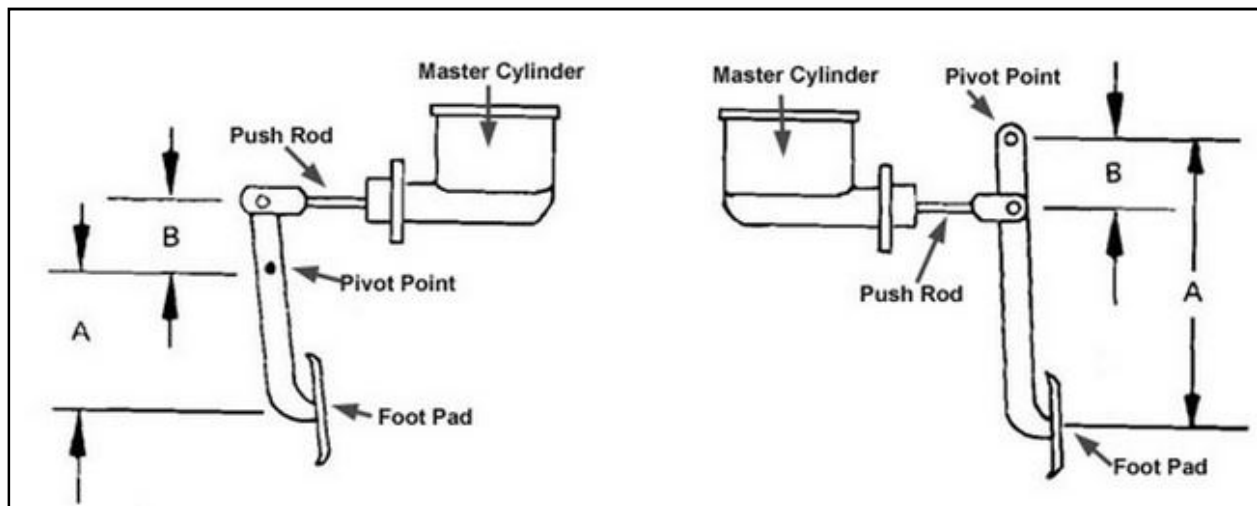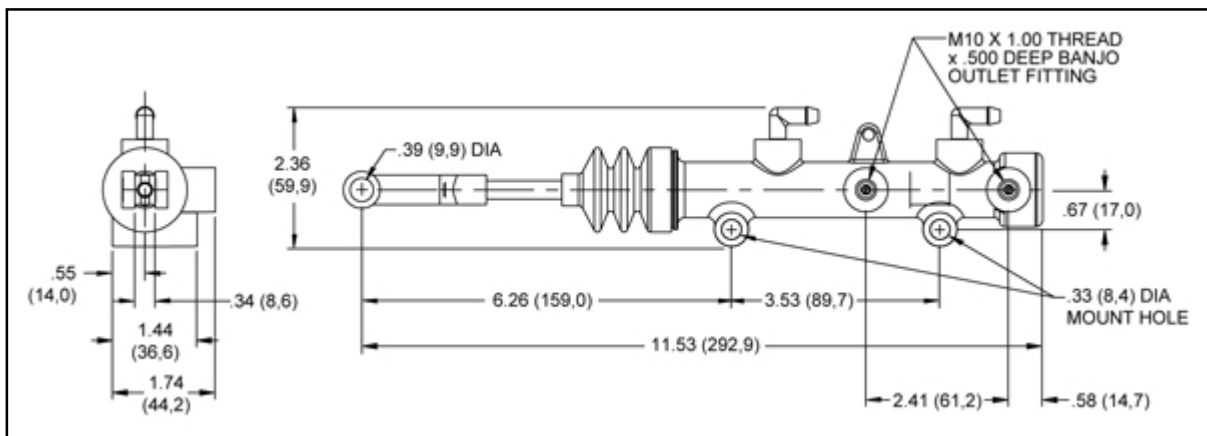

Figure 3: Tandem master cylinder schematic.

As shown in Figure (3), the same force pushes on both pistons and pressurized both the front and rear circuits. Although the diagram shows a leak in the front circuit, the rear circuit remains fully functional. The overall size of a tandem master cylinder is comparable to the combined size of two separate single cylinders. One disadvantage of the tandem cylinder could possibly be its orientation, depending on the setup of the front box of the vehicle. Separate single cylinders are able to be mounted side by side and can often fit behind the brake pedal from the perspective of the driver. The tandem master cylinder is long in comparison and cannot fit behind the pedals without severely compromising the design of the front end of the car. The possible disadvantage of a tandem master cylinder's shape could be avoided by orienting it such that the cylinder and the driver's foot acted on opposite sides of the pivot point. Figure (4) below illustrates the possible orientations of the cylinder in relation to the pivot point of the pedal.

A tandem master cylinder would not be practical in a setup where it had to be mounted away from the driver, as seen on the right side of Figure (4). However, moving the push rod connection to the opposite side of the pivot point allows the cylinder to be mounted on the driver side of the brake pedal, as seen on the left side of Figure (4).

The initial selection of the master cylinder was based on reliability and ergonomics. The Wilwood TM-1 tandem master cylinder has been used for the past two seasons by the team and it has performed reliably. With primary team goals being timely design and manufacturing, the decision to implementing a bias bar system without any past experience from any current team members was a great risk.

**Figure 5:** Wilwood TM-1 tandem master cylinder.

The other major consideration is ergonomics. It was decided early in the design process that the overall frame was going to be made to fit only the smallest members of the team in order to save weight. This led to extremely limited space in the front box in which to fit the pedal assemblies, master cylinder, steering rack and pinion assembly, steering column, and the driver's feet. With a tandem master cylinder oriented on the driver side of the brake pedal, it can be mounted as closely to the steering column as possible in the middle of the front box. Considering the limited dimensions of the box, it is necessary to take advantage of generally unused space between the driver's legs. Figure (6) shows the close proximity achieved between the steering column and master cylinder and the space created for the driver's leg and foot.
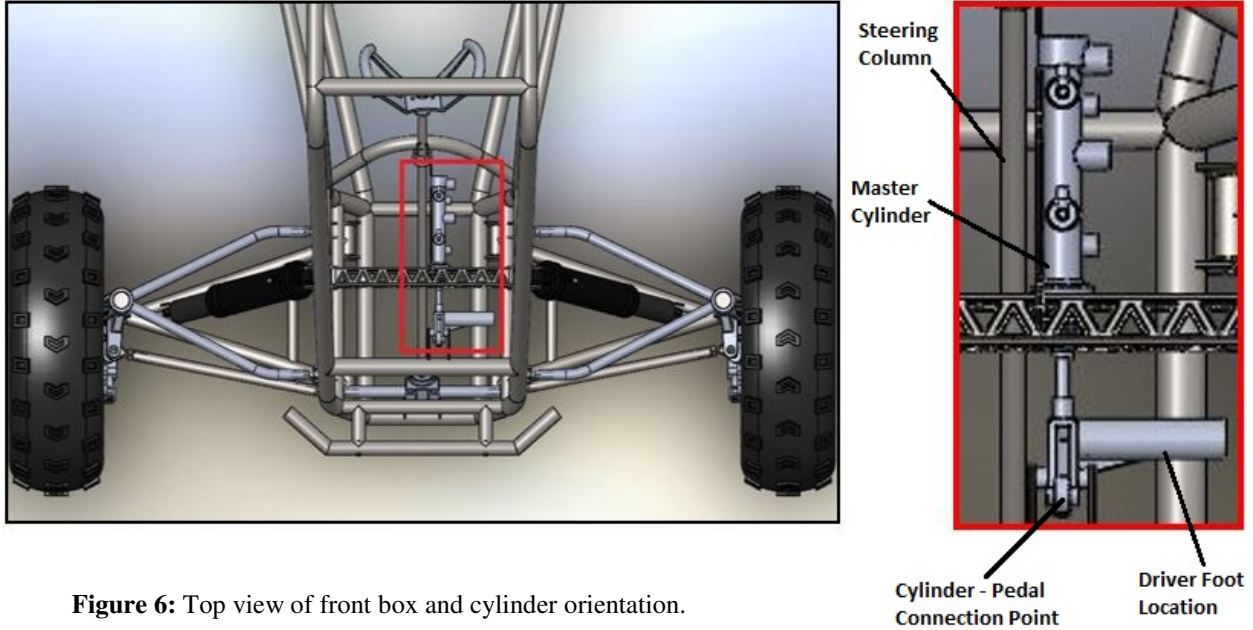
Figure 6: Top view of front box and cylinder orientation.

PROPORTIONING VALVE

The decision to use a tandem master cylinder necessitated the use of a proportioning valve. A proportioning valve essentially regulates the fluid flow in a circuit by partially blocking the fluid passage.
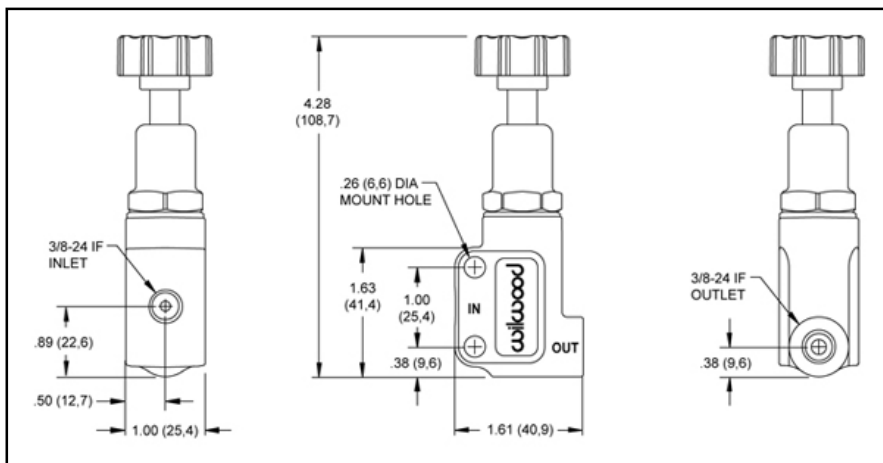
Figure 7:   Wilwood
Adjustable Proportioning Valve.

Since a proportioning valve can only be used to decrease the amount of pressure in a line, it must be integrated into the rear circuit because it will require less pressure than the front circuit. This particular model is able to decrease pressure to 57% of the full circuit pressure. When full calculations were performed, the final estimated front and rear pressures were checked to make sure that the rear pressure value was at least 57% of the front pressure value.

SYSTEM PERFORMANCE CALCULATIONS

Once the necessary initial design selections were made, a series of calculations were performed based on the competition brake test. In this scenario, the vehicle is theoretically decelerating instantaneously from its top speed to a complete stop. This scenario represents the maximum demand seen by the brake system, excluding unpredictable impact cases such as collisions with other vehicles or course obstacles.

There are multiple ways that the system demand calculation process could be approached. For example, analysis could begin at the brake pedal with a desired amount of driver effort. The result of this force (system pressure) could then be analyzed at the caliper assemblies to determine properties such as necessary rotor diameter or caliper piston bore. Since the rotor diameters and calipers were initial selections, an alternative approach would be to calculate the necessary brake force at the wheels and trace this back through the system to determine the necessary force on the master cylinder piston. This allows the brake pedals to be designed based on the expected force that the driver will need to apply to stop the car. If this force or the necessary system pressures turn out to be impractical, initial decisions can be iterated to optimize the system.

The full derivation of equations and associated theory can be found in Appendix (C). In order to assist in the iteration process, the derived system of equations was put into an Excel spreadsheet. This also allowed vehicle characteristics such as weight and ride height to be updated as total design progressed without having to re-do calculations by hand. Table (2), (3), and (4) show the results of the analysis:

CHARACTERISTIC	SYMBO	UNIT	VALU
Gravity	g	ft/(s^2)	32.2
Friction Coeff. - Tire/Road	μT	-	0.73
Friction Coeff. - Pad/Rotor	μR	-	0.55
Weight	W	lbf	550
Height of Unsprung Mass	H	in	22
Wheelbase	L	in	58
C.O.G. to Rear Axle	Lr	in	22.5
C.O.G. to Front Axle	Lf	in	35.5
Front Tire Radius	Rft	in	11
Rear Tire Radius	Rrt	in	11
Max Foot Stroke	Sfootmax	in	
Master Cylinder Piston Area	Amcp	in^2	0.306796
Master Cylinder Stroke	Smcp	in	1.3
PS1 Caliper Piston Area	Acp	in^2	0.7854
PS1 Stroke	Scp	in	0.02
Front Rotor Diameter	Dfr	in	7
Wheel Center to Front Caliper Piston	Rfcp	in	3
Billet Caliper Piston Area	Arcp	in^2	0.7854
Billet Caliper Stroke	Srcp	in	
Rear Rotor Diameter	Drr	in	7.625

System

CALCULATED VALUES	SYMBOL	UNITS	VALUE
Front Normal Force	Fnf	lbf	365.66
Rear Normal Force	Fnr	lbf	184.34
Front Caliper Force	Ffcp	lbf	978.74
Front Circuit Line Pressure	Pf	lbf/in^2	566.44
Master Cylinder Force (Front)	Fmcf	lbf	173.78
Master Cylinder Force (Rear)	Fmcr	lbf	173.78
Shaft Center to Rear Caliper Piston	Rrcp	in	3.31
Rear Caliper Force	Frcp	lbf	446.88
Rear Circuit Line Pressure	Pr	lbf/in^2	517.26

parameters from Excel calculation spreadsheet.

Total Braking Force Needed	Fb	lbf	401.5			
Front Braking Force Needed	Fbf	lbf	266.9282759		66.48276	%
Rear Braking Force Needed	Fbr	lbf	134.5717241		33.51724	%

Table 3: Calculated results from Excel calculation sheet.

Table 4: Braking force results from Excel calculation sheet.

DESIGN ITERATION AND OPTIMIZATION

Once results are generated, it is necessary to check several values in order to verify initial design decisions. The primary values that need to be checked are the front and rear circuit pressures. Due to weight transfer during deceleration, the braking force needed in the rear circuit of vehicles is always less than the force needed in the front. Since proportioning valves can only be used to decrease circuit pressure, and therefore decrease braking ability, they are almost exclusively installed in rear brake circuits. If the rear circuit pressure is greater than the front circuit pressure, the proportioning valve would have to be installed in the front circuit. Installing this valve in the front circuit would most likely draw criticism from competition judges and is not generally good engineering practice.

After the first iteration of calculations, the rear pressure was indeed greater than the front pressure. This was primarily due to the limitation of the rear rotor diameter. Since the rear rotor diameter was already at the maximum value allowed by other subsystems, the rear line pressure had to be reduced relative to the front line pressure another way. Because of the limited options in the rear system components, the front system would have to be changed to fix the problem. Since the rear circuit pressure needed to decrease *relative* to the front circuit pressure, the same result could be achieved by leaving the rear system unchanged and increasing the pressure in the front circuit. There are two main changes that can be made to increase the necessary pressure in the circuit. One way is to decrease the rotor diameter. Since the moment arm between the caliper and the center of the wheel would be decreased, the force applied to the rotor would have to be increased by increasing the pressure.

However, at this point in design, the upright onto which the caliper mounts was in the late stages of design, and changing the caliper mounting hole location due to a change in rotor diameter would add weeks to the design process. In the interest time, the circuit pressure had to be increased another way. This alternative way to increase necessary circuit pressure is to decrease the caliper bore size. A decrease in the piston area would require more pressure in order to achieve the same caliper force. The initially selected front caliper (Wilwood PS-1) is available in several different bore sizes while maintaining the same outer dimensions and mounting hole locations. Tables (2), (3), and (4) show the results of the calculations with the new caliper. The cell highlighted red in Table (2) shows the caliper piston area of the new caliper, which has a bore diameter of 1.00 inches compared to the initial choice caliper that had a bore diameter of 1.12 inches. The cells highlighted blue in Table (3) show the pressures in the front and rear circuits. The rear pressure is lower than the front pressure, and the proportioning valve would then be installed in the rear circuit provide the lower pressure. It is also seen that the rear pressure is at least 57% as high as the front pressure, which justifies the particular model of valve that was initially selected and is shown on Page 13.

BRAKE & THROTTLE PEDAL DESIGN

Once the system calculation process was complete and the necessary force applied by the master cylinder's plunge rod is known, the brake pedal could be designed. Two performance characteristics were considered during the design process. One is the force that needs to be applied by the driver. A very high force is inconvenient for the driver and can become exhausting over the course of a four-hour endurance race. Also, because of the placement of the master cylinder (towards the center of the front box), the pedal had to be designed in an "L" shape such that the driver's foot could be oriented in a natural position (See Figure (10). This pedal shape will cause bending about multiple axes and would require a very robust design if large forces were required every time the driver wanted to stop the car. The other performance consideration is the foot stroke. One criticism of last year's car was that the brake pedal had a

stroke of almost zero, and this was the biggest contributor to the "all or nothing" braking ability of the car. If the design goal of having intermediate braking capabilities was to be achieved, the pedal stroke needed to be increased.

Both of the performance characteristics could be achieved by selecting a favorable mechanical advantage. A brake pedal such that the location of the driver's applied force and the attachment to the master cylinder are on opposite sides of the pivot point is shown in Figure (8).
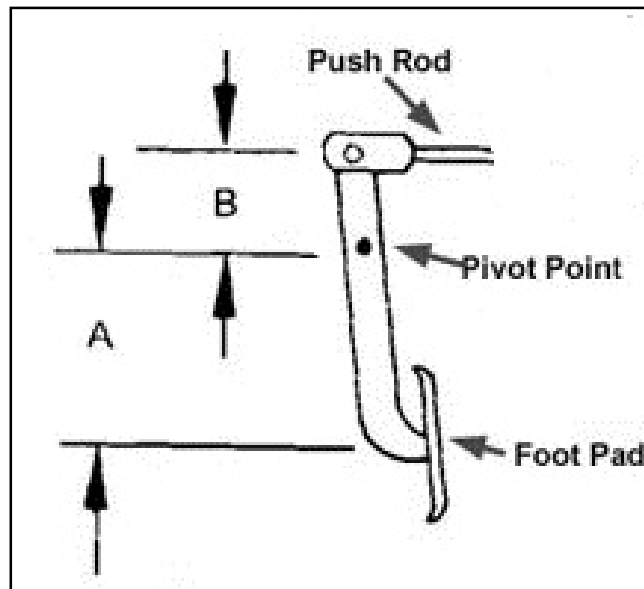
Figure 8: Schematic of generic brake pedal.

Taking moments about the pivot point and manipulating yields the following:

$$\frac{A}{B} = \frac{F_{MC}}{F_{foot}}$$

The ratio given above is the mechanical advantage of the pedal, or pedal ratio. The inverse of this ratio is also the ratio of the stroke of the foot to the stroke of the cylinder:

$$\frac{A}{B} = \frac{S_{foot}}{S_{MC}}$$

It can be seen that a higher pedal ratio would be advantageous for both the driver's force and the pedal stroke. A higher ratio will result in lower driver force and longer foot stroke. Using the calculated master cylinder force and a desired driver force, the ideal ratio is found. With this number, any combination of *A* and *B* dimensions can be chosen to satisfy the pedal ratio based on the available space in the vehicle.

A few considerations had to be made when selecting the *A* and *B* dimensions. One is the vertical space limitations. One of the system design goals was to have the entire brake assembly sit lower in the front box in order to improve the driver's view out of the front of the car. This limits the location of the of the attachment point between the pedal and the master cylinder in the

elevation plane. The bottom of the pedal is constrained by the size of the driver's foot. The contact point between the driver's foot and the pedal must be at a comfortable height for the driver such that the appropriate amount of force can be applied easily for extended periods of time. At the beginning of the design process, a replica of the frame was made out of foam members to ensure that the design would be comfortable and to check certain frame dimensions relative to the driver in compliance with official Baja SAE regulations. Using this replica, the ideal position of the brake and throttle pedals and the ideal strokes were measured while the drivers were sitting in the frame replica.
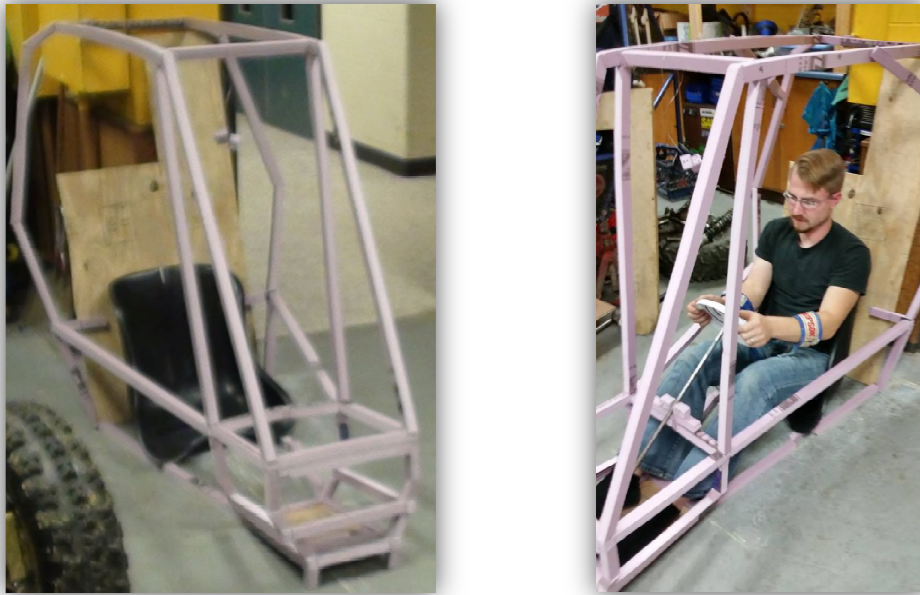
Figure 9: Foam frame replica used in early design process.

These measurements gave the location of the bottom of the brake pedal in the elevation plane and also a desired value for the foot stroke (approximately one inch). Using the 2014 car, the stroke of the master cylinder when the system was fully pressurized was measured to be 0.375 inches. Using this as an estimate for the 2015 car, the necessary pedal ratio according to stroke preferences was approximately 2.67. Before this could be finalized, the ratio needed to be checked in terms of forces. With the estimated master cylinder force being 173.78 lbs. (see Table 3) a pedal ratio of 2.67 would result in a necessary driver effort of approximately 65 lbs. This driver force is low enough to be easily applied in a full-brake situation and high enough that the driver can still have a feel for the pedal's location during intermediate braking situations. With an appropriate pedal ratio and ideal locations for the top and bottom of the pedal identified, the final distances between the foot location, pivot point, and master cylinder connection were established.

The final step of pedal design was to find the necessary dimensions of the pedals through stress analysis. This analysis consisted of several FEA trials until an appropriate safety factor was reached with respect to yielding when using a worst-case emergency brake scenario of 300 lbs. of force applied by the driver. Adjustments between trials consisted of adjusting thickness
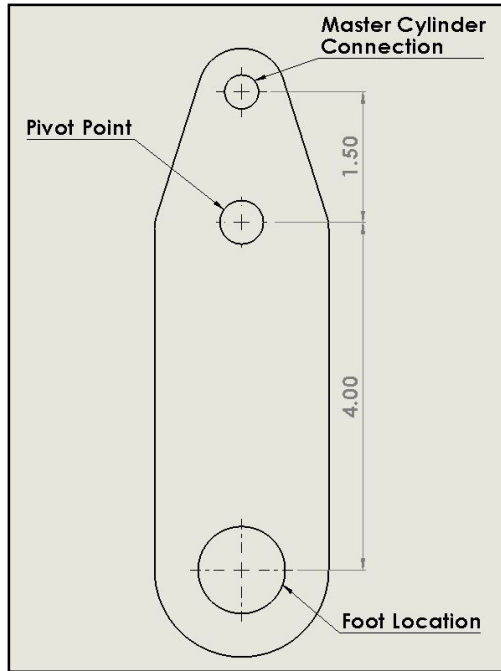
and depth of the main arm of the pedal as well as adjusting the sizes and spacing of weight-saving cutouts. Figure (10) shows the initial pedal geometry and the final design of the brake pedal. Full FEA analysis results can be seen in Appendix (B).
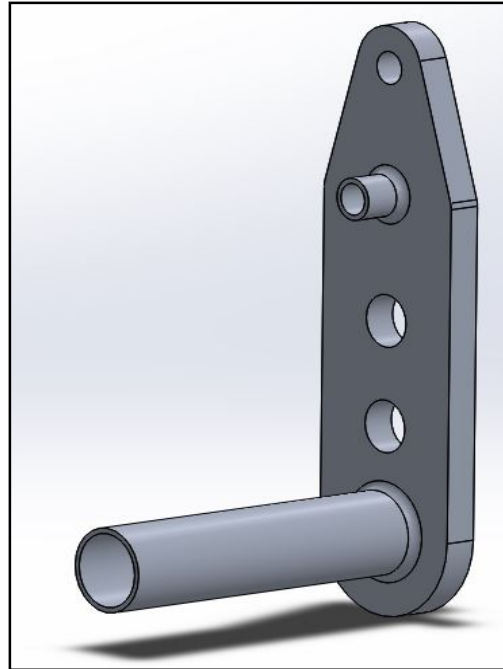
Figure 10: (LEFT) Initial dimensions based on pedal ratio. (RIGHT) Final brake pedal design.

The design of the gas pedal was much more straightforward than the design of the brake pedal. For driver comfort, the foot elevation was made identical to that of the brake pedal. The pivot point was made at the elevation of the frame tube onto which the pedal tabs would be mounted in order to keep the bending moment on the tabs to a minimum. The FEA analysis was done expecting a maximum foot force of 65 lbs. although only 15-20 lbs. of force are needed to go full-throttle. Changes made to the design between FEA trials were the thickness and width dimensions, as well as the dimensions and spacing of the weight-saving cutouts. Final throttle pedal design is shown in Figure (11).

Figure 11: Final throttle pedal design.
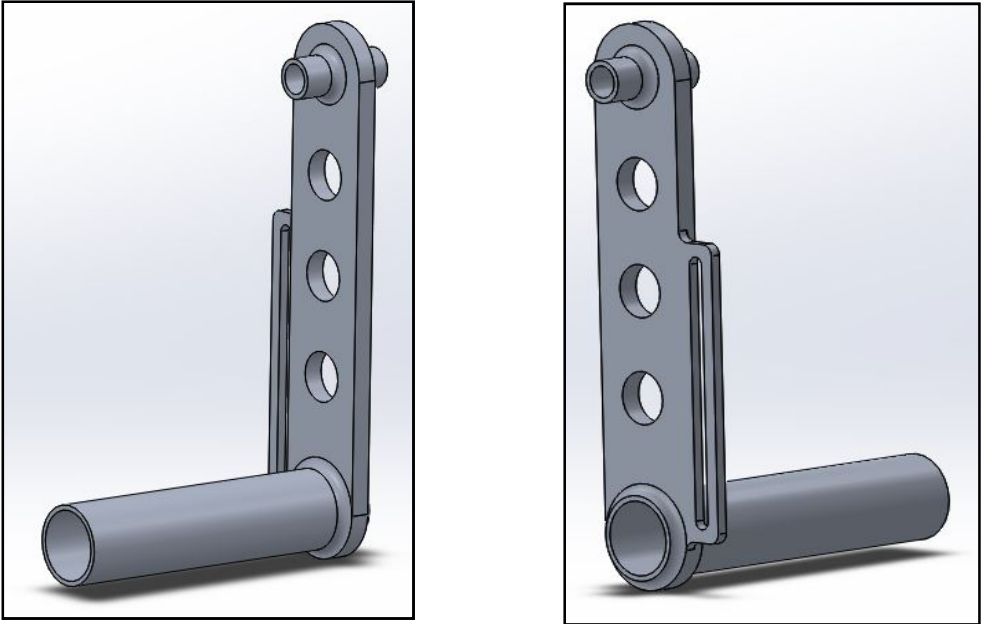
Competition Results and Adjustments

The 2015 Zips SAE Baja car raced in the first of three national competitions in Auburn, Alabama in mid-April. None of the subsystems experienced catastrophic failure during the competition as the team earned a 37th place finish overall out of 105 teams with strong 7th place and 13th place finishes in the suspension and hill climb events, respectively. Both the static and dynamic brake tests were passed on the first attempt. By the end of the competition, however, it was clear that several changes needed to be made to the brake and throttle system.

First, both the brake and throttle pedals had been deformed due to torsion. This was expected to an extent because of the L-shaped design, but the cross-sectional dimensions of both pedals were increased for the next competition.

The biggest problem of the entire event was the throttle cable. The cable pulled out of its rear connection multiple times during competition which resulted in time spent racing at less than full-throttle. The tab in the front box that holds the end of the cable sleeve was also severely bent, which caused the cable to not return properly into the sleeve. To fix these issues, a new tab was made for the sleeve that spans between two tubes in the front box instead of being cantilevered from one tube like the original tab. This adjustment will significantly decrease the risk of bending during competition, especially if the tab is inadvertently kicked by the driver. It was discovered that the cable was pulling out of its rear attachment because the set-screw that is used to hold the cable in place was not making sufficient contact with the cable. To fix this, a small piece of metal was placed in the set-screw hole before the screw was put in and tightened. This resulted in a greater contact area with the cable and less slippage.

The other major problem was the wear of the front rotors and front brake pads. By the end of the competition, there was no brake pad material left at all, and the rotors had worn by several thousandths. It is believed that the cutout pattern of the rotors possessed too many edges on which the brake pad could scrape. For the next competition, the cutout pattern has been changed to feature less edges, and the thickness of the rotors was increased slightly to ensure that the rotor will still be above the minimum thickness specified by the caliper manufacturer even if they are worn during competition. The original front rotor design used at the first competition and the modified design that will be used at the second competition are shown in Figure (12) on the next page.
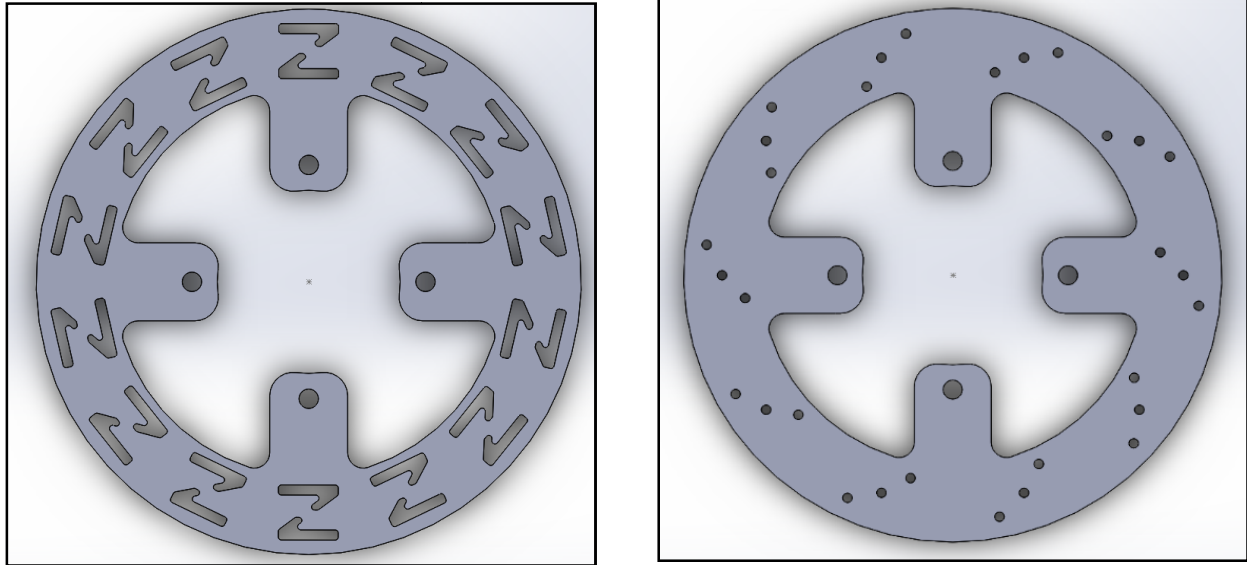
Figure 12: (LEFT) Original front rotor design. (RIGHT) Modified front rotor design.

All of these changes will be monitored very carefully during testing leading up to the second competition in order to verify that sufficient adjustments have been made.

Conclusion

The SAE Baja student design team allows BSME candidates to execute all aspects of a design project from start to finish. Beginning with the full design process, the team set design goals that reflected the desired performance characteristics of the car. A full CAD model was generated and stress analysis was performed on all components. During the manufacturing process, the team machined and fabricated many elements of the car. Besides the physical work, students also had to deal with business and logistical issues such as acquiring raw materials and commercial parts, communicating with businesses that provided services such as gear machining and metal cutting, and coordinating with other university groups to get time in the machine shop. In competition, the team found success, but also identified modifications that needed to be made to optimize the performance of the car. The performance of the vehicle with these modifications will be closely monitored during testing and competition and optimization will continue this year and in years to come.

The design and manufacturing processes of the braking and throttle system in particular has been a similarly iterative process. Modifications were made during the design process as concessions had to be made with other subsystems according to the best interests of the car and the overall design goals. The initial design process and the modification process after analyzing the system's performance have brought the system closer to optimal, but this iterative optimization process will theoretically continue indefinitely. The knowledge has been passed to the next group of system designers and the process will continue, as The University of Akron Zips SAE Baja Team grows closer to the perennially elite status.

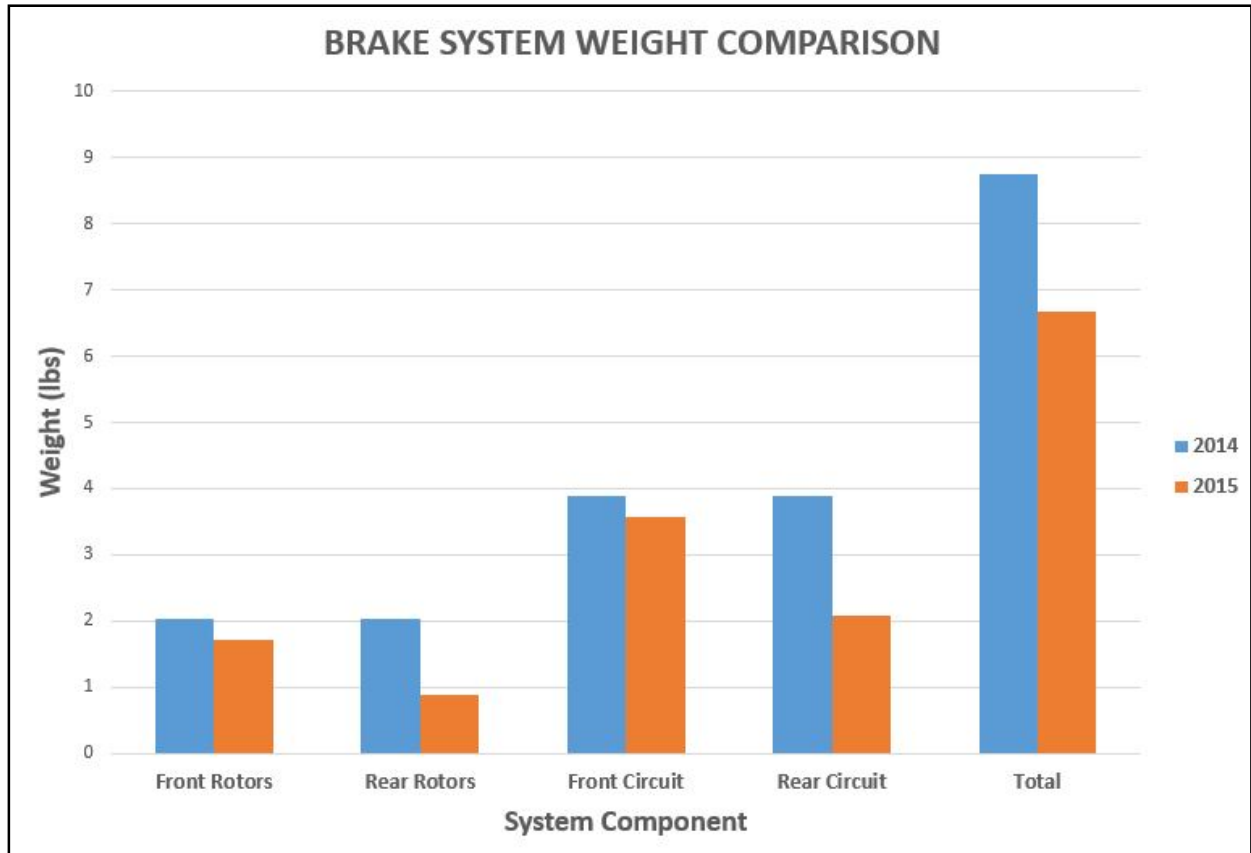# APPENDIX A

ADDITIONAL GRAPHS AND FIGURES

Figure 13: Weight comparisons between 2014 and 2015 Zips Baja vehicles.
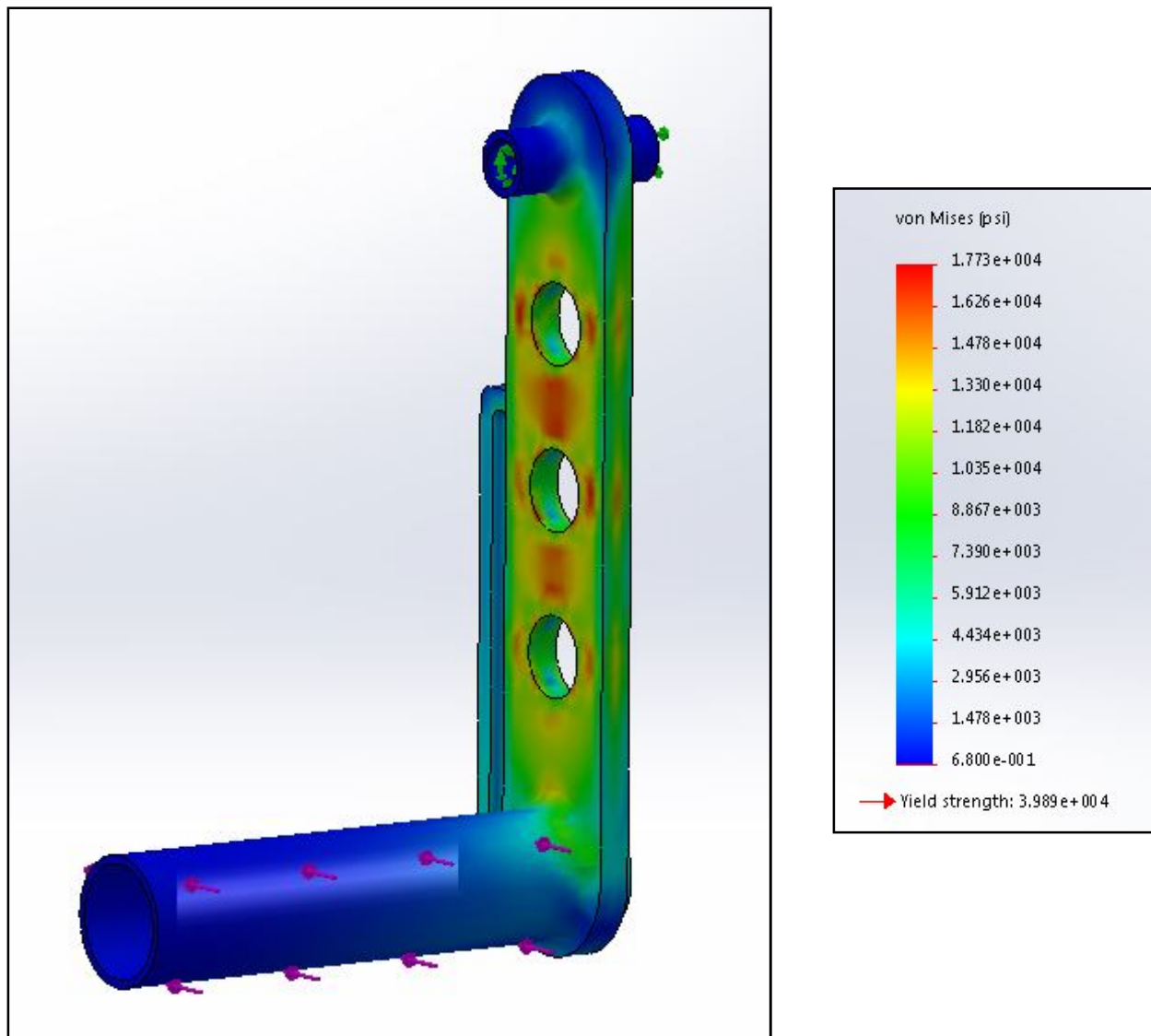
APPENDIX B

FEA ANALYSIS AND RESULTS

BRAKE PEDAL:

The brake pedal FEA analysis was conducted with a hinge connection at the pivot point, a fixed geometry connection at the master cylinder connection location, and an external force at the foot bar. A force of 300 lbs. was used to simulate an emergency stop situation. A final safety factor of 1.8 was achieved with respect to yield in the worst-case scenario.
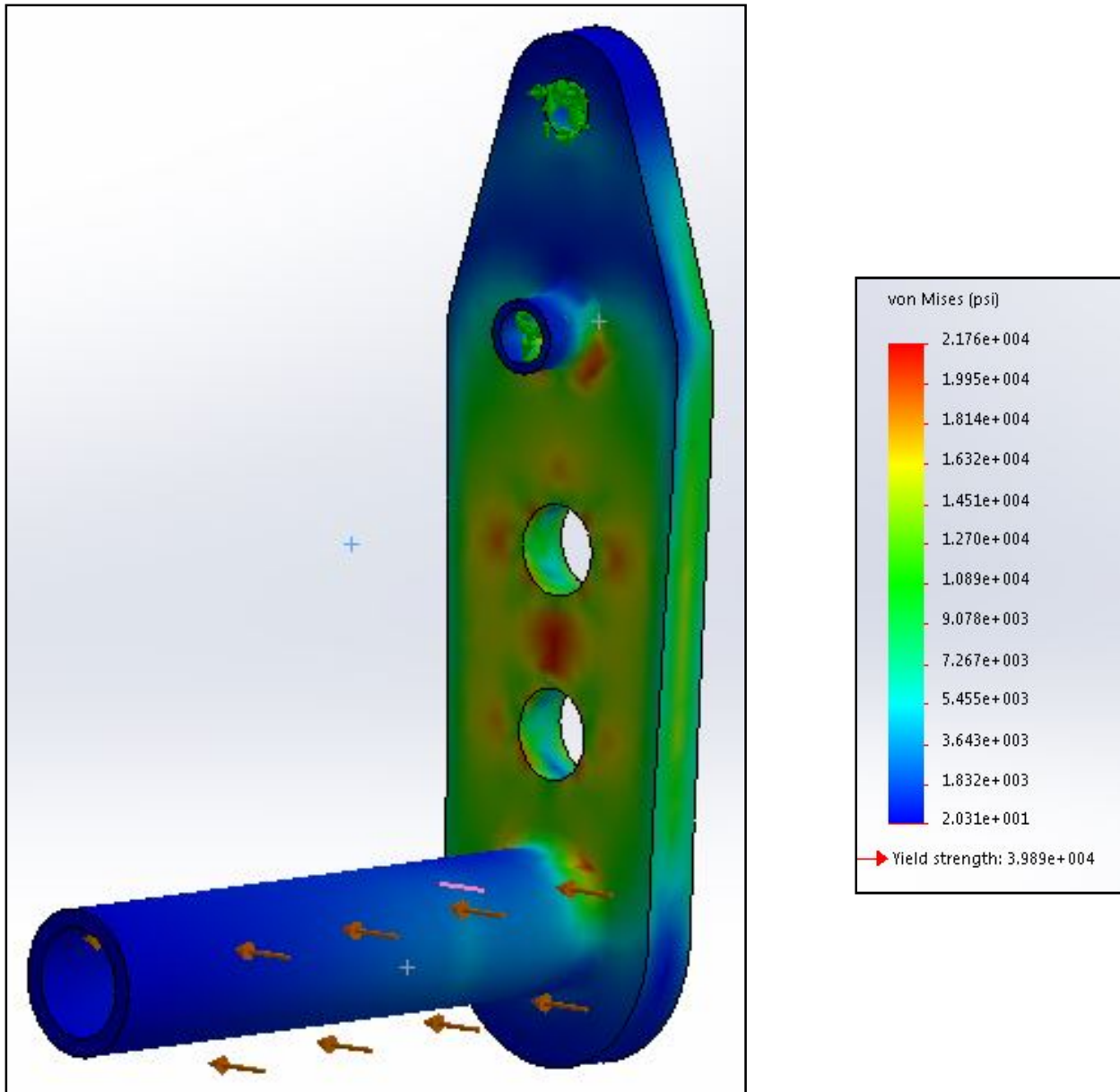
THROTTLE PEDAL:

The throttle pedal FEA analysis was conducted with a hinge connection at the pivot point, a fixed geometry connection at the location of the mechanical stop, and an external force at the foot bar. A force of 65 lbs. was used to simulate three-times as much force as needed to fully engage the throttle. A final safety factor of 2.2 as achieved with respect to yielding.

BRAKE ROTORS:

The brake rotor FEA analysis was conducted using a fixed connection over the area of contact with the brake pads and external forces in the tangential direction at all of the bolt locations.
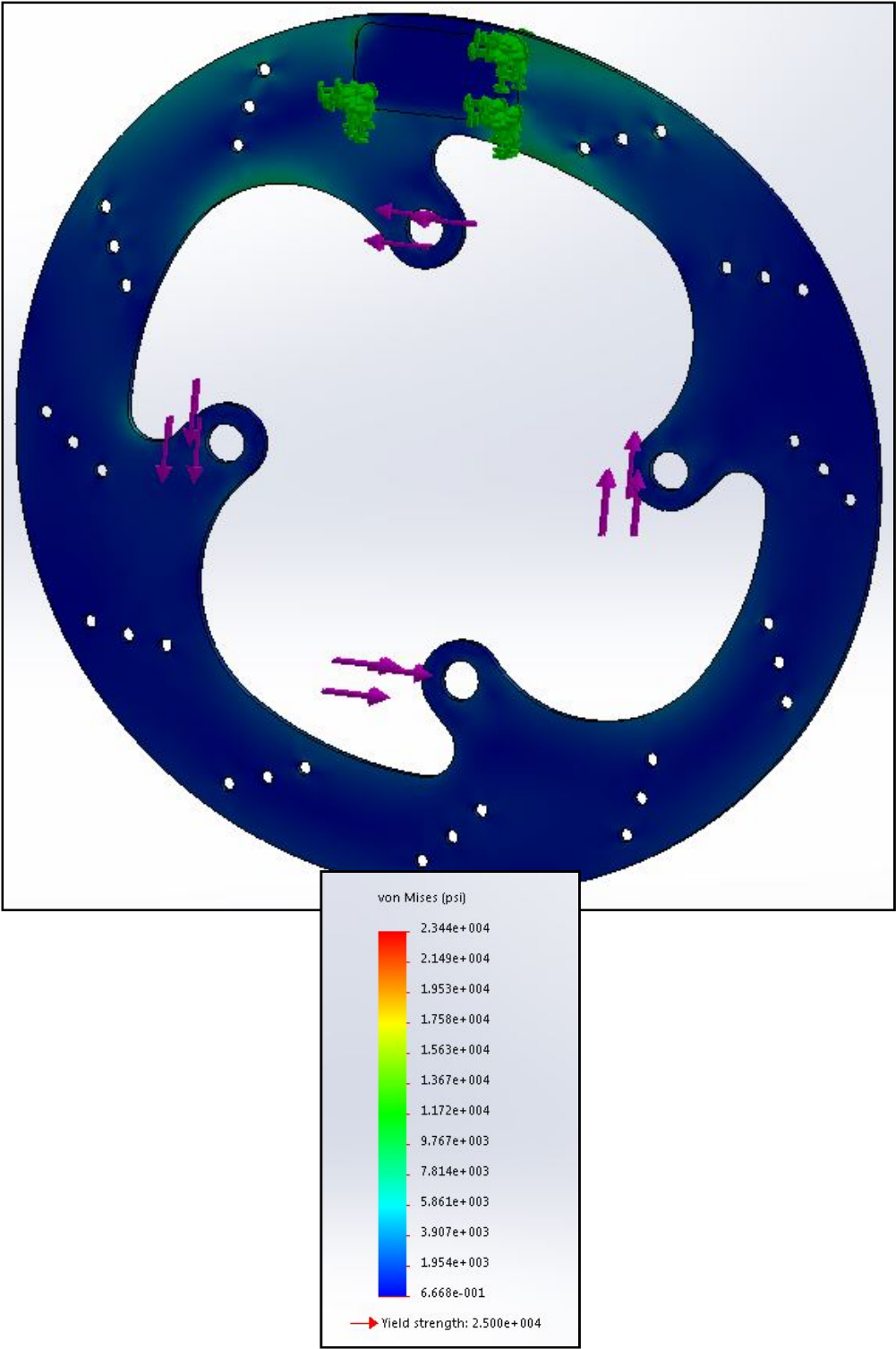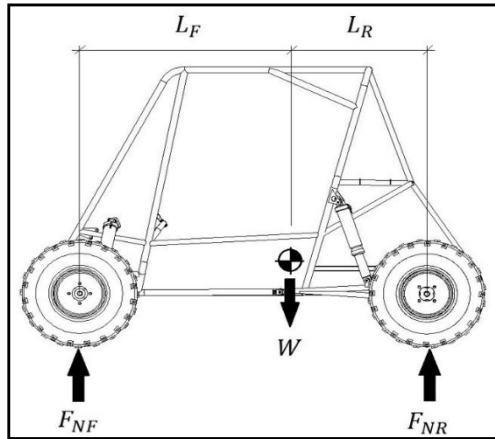
APPENDIX C

THEORY & DERIVATIONS

CALCULATING NORMAL FORCES WITH WEIGHT TRANSFER

First calculate normal forces at steady-state (static or constant-velocity):
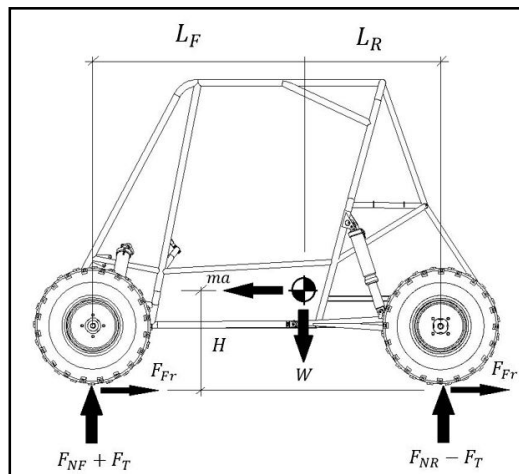

Figure 14: Free body diagram of vehicle at steady state.

Taking moments about center of gravity yields:

$$\Sigma M_G = -F_{NF}(L_F) + F_{NR}(L_R) = 0$$

$$F_{NF}(L_F) = F_{NR}(L_R)$$

To calculate the normal forces during maximum acceleration, the force due to deceleration and friction forces between the tires and the road are considered:

Figure 15: Free body diagram of vehicle during deceleration.

Taking moments about the center of gravity yields:

$$\Sigma M_G = (W\mu_T)H - (F_{NF} + F_T)L_F + (F_{NR} - F_T)L_R = 0$$

$$\boxed{F_T = \frac{\mu_T WH}{L}} \qquad (1)$$

Where F_T is the weight transferred from the front wheels to the rear wheels during deceleration and L is the total wheel base ($L = L_F + L_R$).

The normal forces in the front and rear during deceleration are:

$$F_{NF} = F_{NFS} + F_T \qquad\qquad F_{NR} = F_{NRS} - F_T$$

Where F_{NFS} and F_{NRS} are the front normal forces and rear normal forces at steady state, respectively.

To get expressions for F_{NFS} and F_{NRS}, the moments are taken about the front wheels and rear wheels separately in the free body diagram shown in Figure (14):

$$\Sigma M_R = W(L_R) - F_{NFS}(L) = 0$$

$$F_{NFS} = \frac{WL_R}{L}$$

$$\Sigma M_F = W(L_F) - F_{NRS}(L) = 0$$

$$F_{NRS} = \frac{WL_F}{L}$$

Using the expressions derived for steady-state normal forces and transferred weight, the normal forces during deceleration are given as:

$$\boxed{\begin{aligned} F_{NF} &= F_{NFS} + F_T = \frac{WL_R}{L} + \frac{\mu_T WH}{L} = \frac{W}{L}(L_R - \mu_T H) \qquad (2a) \\[2mm] F_{NR} &= F_{NRS} + F_T = \frac{WL_F}{L} + \frac{\mu_T WH}{L} = \frac{W}{L}(L_F - \mu_T H) \qquad (2b) \end{aligned}}$$

Note that these quantities represent the total normal force on both front wheels and the total normal force on both rear wheels, respectively. The normal force on each front wheel would theoretically be half of the value given by Equation (2a) and the normal force on each rear wheel would theoretically be half of the value given by Equation (2b).

CALCULATING FRONT CALIPER FORCES AND FRONT CIRCUIT PRESSURE

To calculate the necessary force applied by the caliper squeezing onto the rotor, consider the free body diagram of the front wheel assemblies during deceleration:
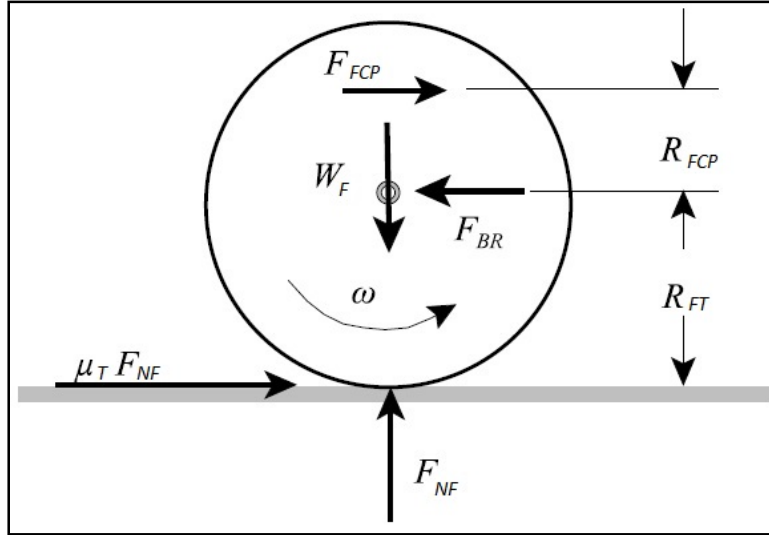
Figure 16: Free body diagram of front wheel assemblies during deceleration. (Courtesy of Dr. R. Gross)

Taking moments about the center of the wheel gives:

$$\Sigma M_O = (F_{NF}\mu_T)R_{FT} - F_{FCP}(R_{FCP})$$

$$F_{FCP} = \frac{(F_{NF}\mu_T)R_{FT}}{R_{FCP}} \qquad (3a)$$

This force represents the force that is applied in the tangential direction when the caliper clamps onto the rotor. This force acts at a distance from the center of the wheel that is equal to the average diameter of the rotor, where the brake pads make contact with the rotor. Note that this force represents the total force on both front assemblies combined (due to all four brake pads).

This force can also be written in terms of the friction force between the brake pads and the rotor using the normal force of the caliper pistons and the friction coefficient between the pads and rotor:

$$F_{FCP} = \mu_R * F_{NFCP} \qquad (3b)$$

Where F_{NFCP} is the force applied by all four front caliper pistons perpendicular to the face of the brake rotor. This force applied by the pistons can also be found in terms of the piston area and the pressure in the front brake circuit:

$$F_{NFCP} = p_F(4 * A_{FCP}) \qquad (3c)$$

Where A_{FCP} is the bore of the front calipers and p_F is the pressure in the front brake circuit.

Combining Equations (3a), (3b), and (3c) gives the pressure needed in the front circuit:

$$p_F = \frac{F_{NFCP}}{4A_{FCP}} = \frac{F_{FCP}}{4A_{FCP}\mu_R} = \frac{F_{NF}\mu_T R_{FT}}{R_{FCP}\mu_R(4*A_{FCP})}$$

$$\boxed{p_F = \frac{F_{NF}\mu_T R_{FT}}{R_{FCP}\mu_R(4*A_{FCP})}} \qquad (3)$$
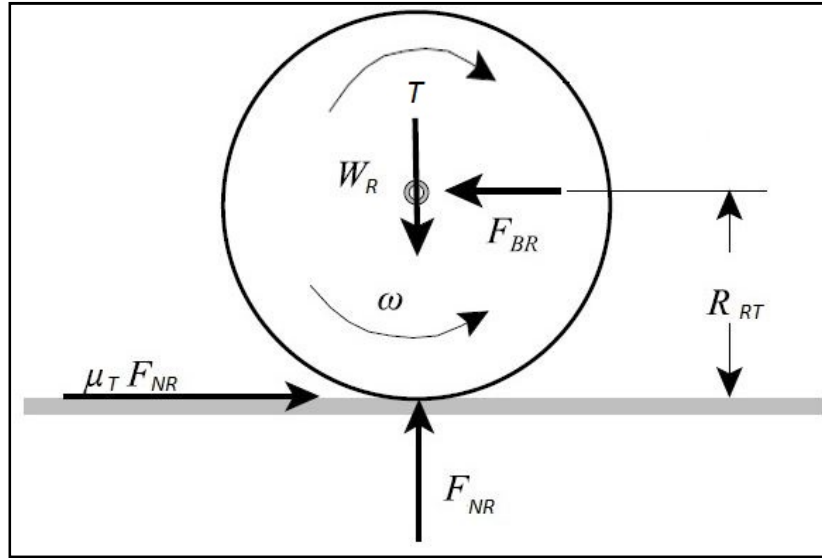
Once the master cylinder is selected, the force that needs to be applied to the master cylinder piston in order to create the necessary pressure can be calculated:

$$\boxed{F_{MCF} = p_F A_{MC}} \qquad (4)$$

CALCULATING REAR CALIPER FORCES AND REAR CIRCUIT PRESSURE

First consider the torque applied to the rear wheel assemblies due to the friction between the tire and the road:

Figure 17: Free body diagram of rear wheel assemblies during deceleration. (Courtesy of Dr. R. Gross)

In Figure (17), the torque, T, applied to the wheel is applied by the axle that is experiencing angular deceleration due to the inboard rear brake. The rear brake is applied to the final output shaft of the gear box. This torque is transmitted to the wheels as it is shown in the figure above through the CV axles.

Taking moments about the center of the wheel gives:

$$\Sigma M_O = (F_{NR}\mu_T)R_{RT} - T = 0$$

$$T = (F_{NR}\mu_T)R_{RT} \qquad (5a)$$

This torque represents the total torque that needs to be applied by the axles on the rear wheel assemblies. Figure (18) shows the free body diagram of the rear brake assembly that is attached to the final output axle of the gear box.
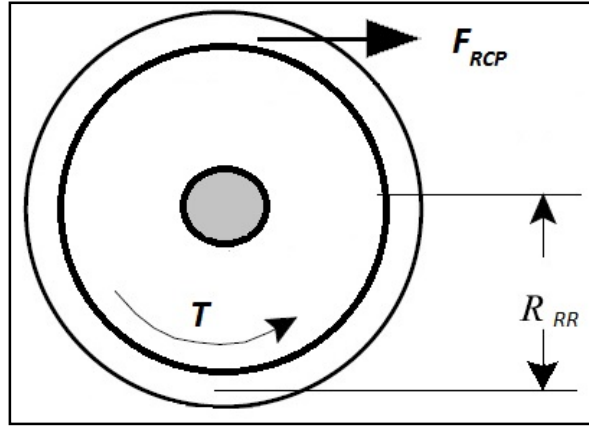
Figure 18: Free body diagram of rear brake assembly.

The torque shown in in Figure (18) is the torque applied on the axle by the friction force acting between the tires and the ground. This is the same torque magnitude as the one described in Figure (17) on the axle. This torque is essentially generated on the axle by the opposing moments of the friction force between the wheels and the ground and the tangential force applied by the caliper on the rear rotor (which is rigidly attached to the axle).

Taking the moment about the center of the shaft gives:

$$\Sigma M_O = T - F_{RCP}R_{RCP} = 0$$

$$T = F_{RCP}R_{RCP} \qquad (5b)$$

Setting Equations (5a) and (5b) equal and rearranging gives:

$$\boxed{F_{RCP} = \frac{(F_{NR}\mu_T)R_{RT}}{R_{RCP}}} \qquad (5)$$

This force can also be written in terms of the friction force between the brake pads and the rotor using the normal force of the caliper pistons and the friction coefficient between the pads and rotor:

$$F_{RCP} = \mu_R * F_{NRCP} \qquad (6a)$$

Where F_{NRCP} is the force applied by both rear caliper pistons perpendicular to the face of the brake rotor. This force applied by the pistons can also be found in terms of the piston area and the pressure in the rear brake circuit:

$$F_{NRCP} = p_R(2 * A_{RCP}) \qquad (6b)$$

Where A_{RCP} is the bore of the rear caliper and p_R is the pressure in the rear brake circuit.

Combining Equations (5), (6a), and (6b) gives the pressure needed in the rear circuit:

$$p_R = \frac{F_{NRCP}}{2A_{FCP}} = \frac{F_{RCP}}{2A_{RCP}\mu_R} = \frac{F_{NR}\mu_T R_{RT}}{R_{RCP}\mu_R(2 * A_{RCP})}$$

$$\boxed{p_R = \frac{F_{NR}\mu_T R_{RT}}{R_{RCP}\mu_R(2 * A_{RCP})}} \qquad (6)$$

Since a tandem master cylinder is being used, the force of the master cylinder piston is applied to the front and rear circuits. The theoretical pressure created by this force in the rear circuit is given as:

$$p_R = \frac{F_{MCF}}{A_{MC}}$$

Note that this pressure is not necessarily (and not likely) the same as the necessary pressure given by Equation (6). The weight transfer of the vehicle from rear to front during deceleration leads to a greater necessary braking force in the front and, most likely, more pressure. For this reason, the proportioning valve is built into the rear circuit to reduce the pressure in the rear circuit.

CALCULATING PEDAL DIMENSIONS

A brake pedal such that the location of the driver's applied force and the attachment to the master cylinder are on opposite sides of the pivot point is shown in Figure (19).
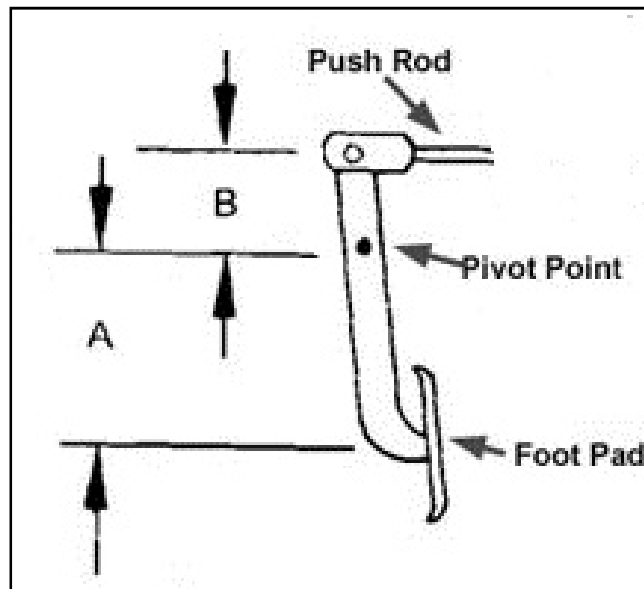
Figure 19: Schematic of generic brake pedal.

Taking moments about the pivot point gives the following:

$$\Sigma M_O = F_{MC}(b) - F_{foot}(a) = 0$$

$$\boxed{\frac{A}{B} = \frac{F_{MC}}{F_{foot}}} \qquad (7)$$

The ratio given in Equation (7) is the mechanical advantage of the pedal, or pedal ratio. The force on the master cylinder is calculated by Equation (4). After a desired driver force is chosen by the designer, the ideal ratio will be known. With this number, any combination of A and B dimensions can be chosen to satisfy Equation (7) based on the available space in the vehicle.

APPENDIX D

2015 SAE BAJA COMPETITION REGULATIONS

ARTICLE 11: BRAKING SYSTEM

ARTICLE 11: BRAKING SYSTEM

B11.1 Foot Brake
The vehicle must have hydraulic braking system that acts on all wheels and is operated by a single foot pedal. The pedal must directly actuate the master cylinder through a rigid link (i.e., cables are not allowed). The brake system must be capable of locking ALL FOUR wheels, both in a static condition as well as from speed on pavement AND on unpaved surfaces.

B11.2 Independent Brake Circuits
The braking system must be segregated into at least two (2) independent hydraulic circuits such that in case of a leak or failure at any point in the system, effective braking power shall be maintained on at least two wheels. Each hydraulic circuit must have its own fluid reserve either through separate reservoirs or by the use of a dammed, OEM-style reservoir.

B11.3 Brake(s) Location
The brake(s) on the driven axle must operate through the final drive. Inboard braking through universal joints is permitted. **Braking on a jackshaft through an intermediate reduction stage is prohibited**

B11.4 Cutting Brakes
Hand or feet operated "cutting brakes" are permitted provided the section (B11.1) on "foot brakes" is also satisfied. A primary brake must be able to lock all four wheels with a single foot. If using two separate pedals to lock 2 wheels apiece; the pedals must be close enough to use one foot to lock all four wheels. No brake, including cutting brakes, may operate without lighting the brake light.

B11.5 Brake Lines
All brake lines must be securely mounted and not fall below any portion of the vehicle (frame, swing arm, A-arms, etc.) Ensure they do not rub on any sharp edges. Plastic brake lines are prohibited