2019

Final Design Report: Design and Development of an Ackermann Steering Geometry for a Formula SAE Car

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Design and Development of an Ackermann Steering Geometry for a Formula SAE Car

Trinity University Motorsports

Formula SAE 2018-19

A. Anderson, G. Bentz, D. Grusak, J. Hudson, J. Marques, L. Wilson

Executive Summary

The steering system was designed to be implemented in Trinity's Formula SAE racecar. All design choices were made first with respect to the FSAE rules and then to the team's production capabilities (manufacturing skill level and the limitations of Trinity's machine shop equipment). The system was first evaluated by its compliance with FSAE rules: limited degrees of free play, quick release safety compliance, and the clearance of the cockpit template, front uprights, and wheel rims. The next feature that was evaluated was the car's ability to navigate a hairpin turn. The steering system was evaluated by its toe in/out, steering ratio, and Ackermann percentage. At low speeds, Ackerman geometries improve the cornering ability in fast, technical tracks.

Test 1 evaluated the free play present in the steering system. FSAE mandates that there be no greater than 7 degrees of free play. The car successfully passed Test 1 revealing that on average there are only 5 degrees of free play in the steering system. Test 2 assessed the car's ability to navigate both clockwise and counterclockwise hairpin turns by comparing the actual operating range with previously computed minimum inner and outer toe angles. The operating angles exceeded the minimum steering angles; therefore, the car should be able to navigate all turns in the Autocross and Skidpad events. Test 3 was designed to mimic the track at the annual FSAE competition. The powertrain subsystem remains incomplete, so the car is to be pushed by design team members while another member steers the vehicle. Due to a recent unexpected break in the left front A-arm of the suspension, Test 3 has not been performed. Test 4 assessed the function of the quick release, cockpit ergonomics, and the ability of a driver to safely exit the vehicle in 9 seconds. Thirty trials by three different drivers demonstrate the success of the quick release feature and the ability to exit the vehicle in far less than 9 seconds.

A primary objective of this senior design project was to meet FSAE guidelines and create a robust system that can be optimized by future senior design teams. Given that the steering system passed the 3 tests that were performed, it is clear that we have produced a working steering system that will provide a strong basis for the next team that continues to prepare the car for competition. Another objective was to produce the car while cognizant of the different FSAE events that the TUMS car will eventually compete in. Two other objectives were to follow a thorough design process for the steering system and to maintain records of design decisions, engineering drawings, and inventory for future students who will work on the car. Throughout the process the team kept organized notes on materials, vendors, purchases, and decisions. Two more objectives were to fabricate and assemble the steering system and implement a placeholder for the incomplete suspension system. Both objectives were met: the steering system is complete and two wooden blocks were placed next to the uprights to support the car in lieu of a function suspension system for testing.. A final primary objective was to dynamically test the steering system (Test 3), but this was not met. Several welds must be repaired before Test 3 can be safely performed. All welds on the suspension and powertrain should be evaluated and strengthened if needed before dynamic testing should proceed.

A secondary objective (not formally evaluated) was to manage the implementation of a braking system to be completed by the current TUMS members. All components of the braking system have been ordered and received. There is a plan for the assembly, but there were not as many active and available TUMS members as anticipated so it has not been completed. To achieve a fully-implemented braking system, all parts should be assembled and plumbing lines purchased and strategically attached. The final goal was to integrate and complete as much of the previously designed subsystems as possible (powertrain, suspension, electronics, etc.). Much research and many steps have been taken towards this objective, but there is a significant future work necessary to achieve a running powertrain and integrated, functional car.

Acknowledgments

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Introduction

Formula SAE is a collegiate competition in which teams of students design, build, and race small open-seat, formula-style race cars in static and dynamic events. Previous senior design teams implemented a complete chassis and a mostly complete powertrain and suspension. The 2018-19 senior design team was tasked with the design, manufacturing, and integration of a steering system for the car which includes some refinement of the existing subsystems and the addition of others like the cockpit. This report outlines the empirical techniques used to validate the features of the steering system, the results of those tests, and recommendations for future work on the car.

This year's objectives are centered around the goal of taking an iteration of this car to compete in the near future and making choices that will benefit the Trinity University Motorsport (TUMS) club over the span of several years. This team aims to design, fabricate, assemble, and test its a steering system. To dynamically test the steering system, it will be necessary to design and implement a device that serves as a placeholder for the suspension system, as well as a braking system. The team will strive to meet FSAE 2018-2019 Guidelines where appropriate and produce a system that can be modified, optimized and further tested by future Trinity Engineering students working on the project. A final goal is to integrate and complete previously designed subsystems as much as possible.

In addition to compliance with the standards listed in the FSAE 2018-2019 Guidelines, the car will be evaluated by its cockpit template clearance, ergonomic positioning, driver effort, and the quick release capability. The steering system will be evaluated based on several parameters: kinematic point location, toe angle, steering ratio, and an ideal Ackermann steering percentage. Keeping in mind the static and dynamic tests that the car will eventually compete in, the steering system should provide the stability to navigate hairpin turns and the skidpad event turns.

Overview of the Design as Tested

Steering System

The steering system uses a rack-and-pinion style actuator to convert rotational driver input to axial displacement to rotate the wheel assemblies about their steering axes. The steering column assembly connects the steering wheel and quick release to the rack.

Design Parameters

The Ackermann steering model describes the difference in the steering angle of the outside and inside tires. Ackermann developed the model to account for differences in angle required on each tire to navigate a defined curve smoothly without bump steer. This model accounts for the inner wheel needing to turn more than the outer wheel. If both front wheels are free to follow their own path, they would converge and cross each other. With the vehicle moving in a single path, wheel tracks conflict, causing detrimental tire slip and tread scrub (Fig. 1). This principle optimizes the handling of the car on corner entry and mid-corner.

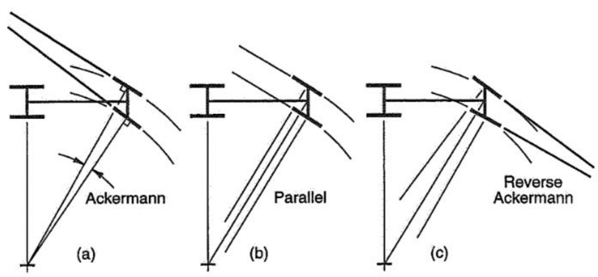


Figure 1. Steering geometry variations: (a) Ackerman, (b) parallel, and (c) reverse Ackerman Steering

Ideal geometric steering angles. An initial condition from which the appropriate linkage geometries and the geometric steering angles was derived using geometric relationships calculated from dimensional vehicle parameters and the desired turning radius. The ideal angles that we care about the most are the inner and outer hairpin turn angles which were found to be 23.40 degrees for the inner angle and 33.26 degrees for the outer angle.

Design Overview

- Geometric Constraints and Packaging.
- Optimization for Minimal Steering Effort.
- System Goals & Final Specifications.

Steering Rack and Rack Mounts

Steering rack selection. Manufacturing a steering rack during the 2018-19 timeline was not feasible for the scope of our project due to time constraints and limited fabrication facilities. To meet the design objectives, a steering rack manufactured by North American Racing Co. (NARRco) was selected for its performance characteristics, specific geometry, and minimal weight.

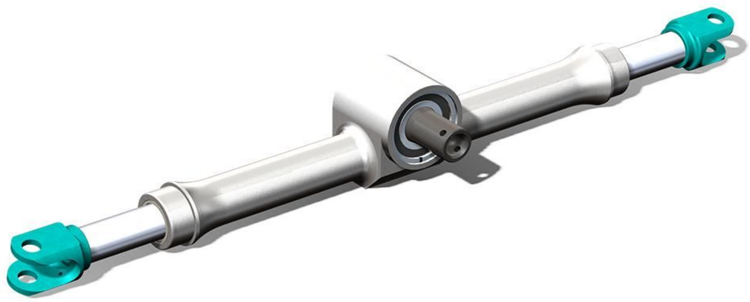


Figure 2. 3D Rendering of the NARRco Rack V1

Maximum Rated Load, Axial Loading & Off-Axis Loading. The maximum rated load is the full axis force in tension or compression which the rack is designed to withstand in service. Experiencing this load under normal driving conditions is uncommon except in the event of a collision involving the front wheels. The maximum rated load corresponds to the maximum force the driver of a small formula-style car could resist before the steering is torn from their grip. For the design of related steering components, the manufacturer stipulates a maximum rated load of 6670 N. The recommended maximum operating load for the NARRco is 1780 N. The installation strove to minimize off-axis loading. The steering arms were installed front of the wheelbase center line at an angle of 3.6 degrees which resulted in minimal off axis loading.

Steering rack specifications and installation recommendations. The NARRco V1 steering rack operates with a clockwise steering input (Fig. 3), has an eye-to-eye length of 11.4", a rack speed of 3.46 in/rev, and weighs 1.3 lbs. The rack-and-pinion assembly can withstand a maximum axial load of 6670 N, which is consistent with a vehicle that weighs less than 900 lbs and acceptable for the current vehicle weight estimate of 400 lbs.

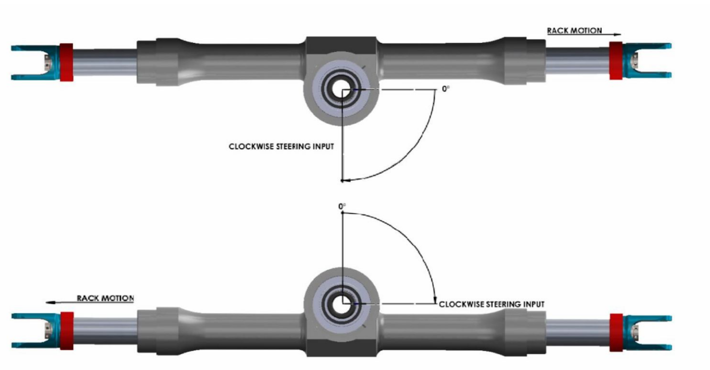


Figure 3. NARRco Rack-and-Pinion Travel Direction

NARRco. recommends a maximum tie rod installation angle of 10 [deg]. The linear travel of the rack is constant and proportional to the rotary steering input angle. The steering input angle has a range of -130º to 130º. The clevis connections to the tie-rods are compatible with standard spherical rod end connectors. The steering shaft is attached to the pinion shaft using a shaft coupling placed over the pinion shaft. The coupling is attached to the pinion shaft with a ⅛" steel coiled spring pin and then coupled to the steering shaft.

Design and fabrication of rack mounts. The steering rack is affixed to the chassis of the car with two machines 6061 aluminum mounts positioned at both ends of the rack and then attached to the ⅛" steel floor of the car. The mounted are positioned at the far ends of the pinion housing for maximum deflection resistance resolving the torque transfer from the beveled gear of pinion and the rack. To ensure the fasteners could resist the shear stress, SAE Grade 8 fasteners were used to fix the assembly. The rack mount design (Fig. 4-5) is fixed to the floor closeout of the car with a ¼"-28 x 3" partially threaded hex cap bolt with accompanying washers and nuts, torqued to their recommended settings. The bolts fasten the two pieces tightly around the rack restricting its range of motion.

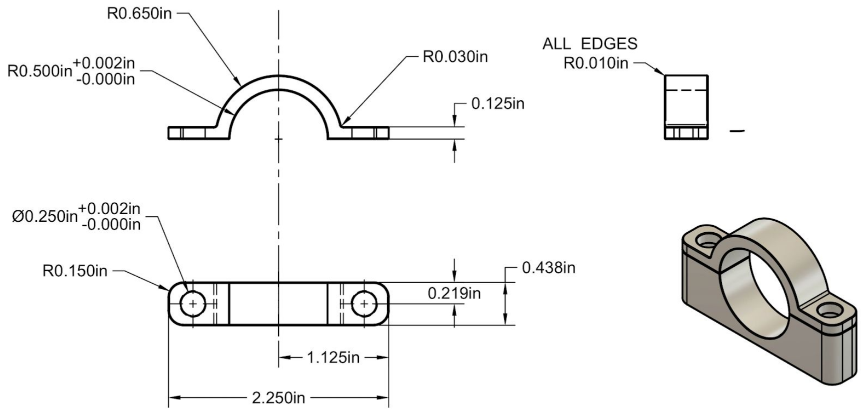


Figure 4. Engineering drawing of the rack support

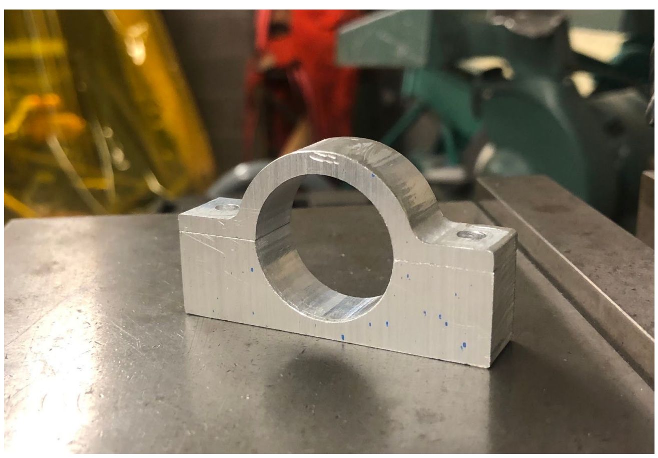


Figure 5. Fabricated rack support made from 6061 aluminum

Steering Shaft assembly and Pinion Shaft Connection

Background and Design. A *shaft* is a rotating member, usually of circular cross section, used to transmit power of motion. In the steering system, it provides the axis of rotation, or oscillation, of the elements in the assembly and controls the geometry of their motion. The shaft assembly consists of a cold-drawn steel shafts, a u-joint, and collars to dictate the axial positioning of components under axial loading. The steering shaft assembly consists of the steering shaft, a u-joint and shaft-mounted devices, coupling connections. A ¾" cold drawn steel shaft was coupled to the pinion of the rack and pinned with ⅛" steel spring pins. The other end of the shaft was splined and mated to a needle bearing u-joint with a maximum operating angle of 35 degrees. The splined u-joint uses a ⅜-16 set screw to fasten the shaft in place. The other end of the u-joint has a ¾" bore and was fastened to another length of ¾" steel rod with a ⅜-24 set screw (Fig. 6). This length of steel rod was connected to the splined input of the quick release via a fabricated steel coupling fixed with ⅜-24 set screws.

Figure 6. U-joint and shaft coupling assembly

Steering Column Support

Steering column support. To support the steering column assembly and to allow rotation of the shaft, a supporting component was fabricated out of 6061 aluminum using the CNC and fastened to the 1" chromoly tubing of the chassis (Figs. 7-8). Using The CAM workspace in Fusion 360, the part was cut as a single piece on the CNC, holes were then milled and threaded to accept #10-32 hex head machine screws.

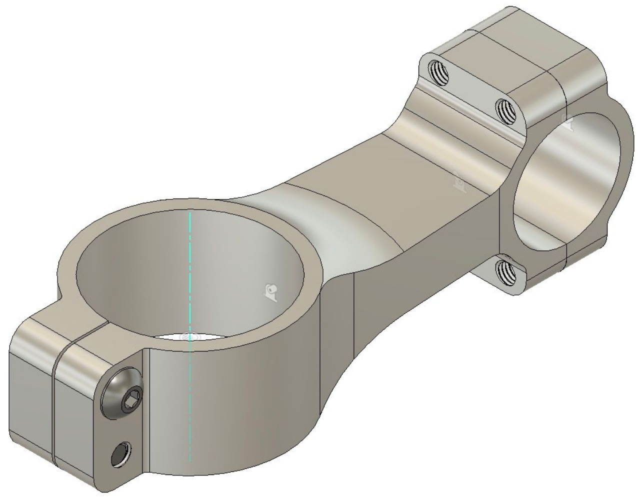


Figure 7. Fusion 360 model of the STEERING-RACK-SUPPORT

The body was cut into components, then thru holes were milled through top (i.e. that which clamp onto the chassis).

Figure 8. CNC manufactured steering column support fixed to the vehicle chassis

The other side of the part has a 1-½" hole to accept a needle roller bearing to support the radial load of the assembly and allow smooth rotation of the steering shaft. The needle-roller bearing uses a shaft liner to seal the bearing from outside contaminants and to reduce the interior diameter of the to ¾". The needle bearing is press fit into the steering column support and fixed axially with external retaining rings. A HSS grooving tool was fabricated to cut the retaining ring groove required for installation.

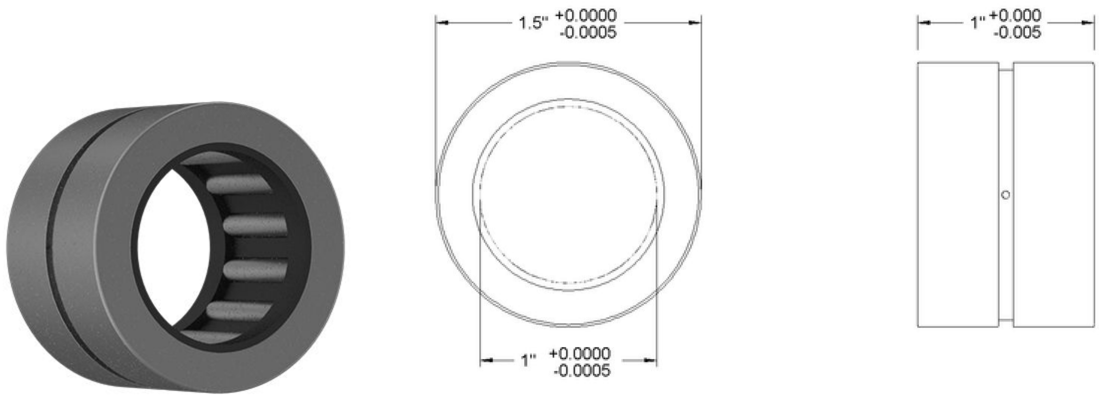


Figure 9. Needle roller bearing

Tie Rods

Background, design, and fabrication. The tie rods connect the clevis ends of the steering rack to the wheel uprights and given an input from the user, rotate the wheel assembly about the steering axis. Spherical rod ends (¼"-28 LH) are attached to each clevis end of the rack with an AN-3 fastener. The tie rod linkages are 13" steel hex turnbuckles with ⅜"-24 threaded ends. To reduce the size of the threaded rod and make the connections to the rod end, ¼"-28 LH bungs were TIG (GTAW) welded to into place (Fig. 10) finalizing the heim joint. SAE jam nuts (¼-28 LH) are tightened onto the rod ends to lock the rod end in place, but are not show in Figure 11.

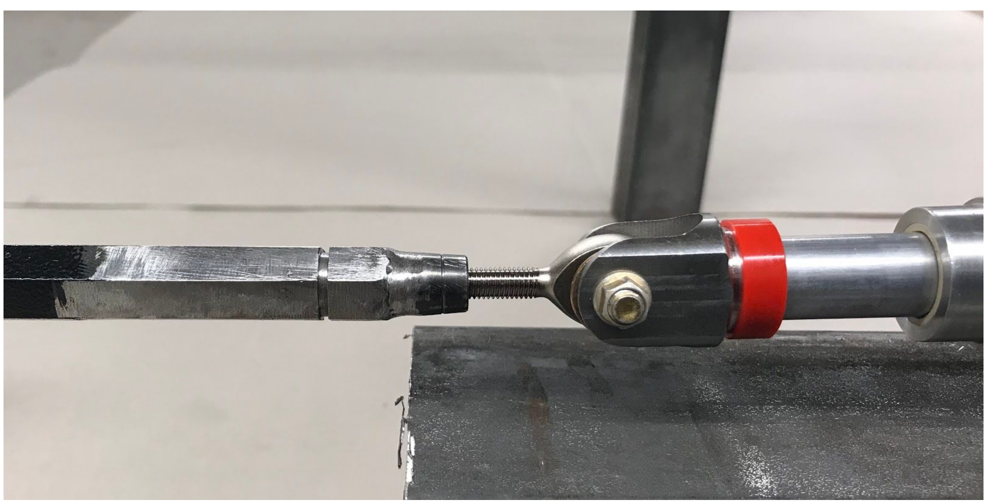


Figure 10. Steel hex tie rods showing the clevis rack connection via a ¼-28 spherical rod ends creating a heim joint.

Figure 11. Steering end links making the tie rod connection to the uprights. The tie rod end links are connected to the upright at an angle with the horizontal plane of less than 5 degrees.

Other Relevant Subsystems

Brake System

The car currently employs an outboard disc brake system, which is considered reliable with acceptable performance figures in terms of braking force and thermal dissipation capabilities. Larger components were chosen for the front brake assembly because when turning, more lateral forces are applied to the front tires. More braking force in the front allows for increased handling and driver response/feel.

Rotors:

- Front: 10”
- Back: 9”

Calipers:

All brake calipers were sourced from Wilwood.

- Front brakes are fitted with Wilwood Dynapro single brake calipers, the surface area is $3in^2$
- Rear brakes are fitted with Wilwood PS-1 brake calipers, the surface is $2in^2$

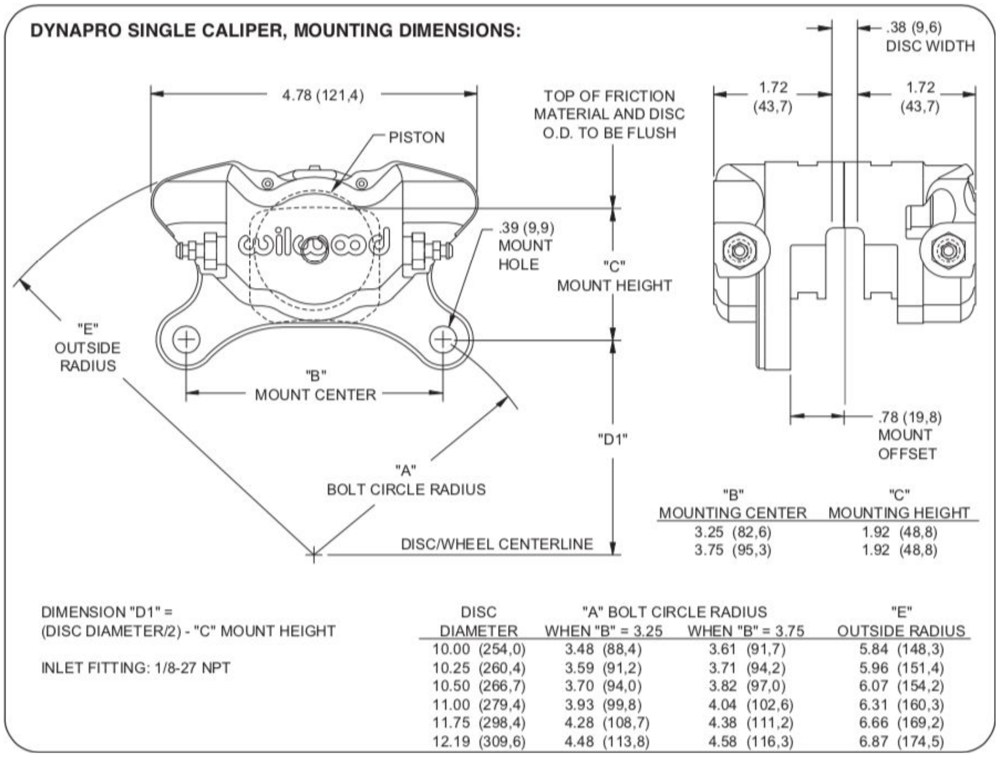


Figure 12. Wilwood Dynapro brake calipers. Surface area of 3in^2.

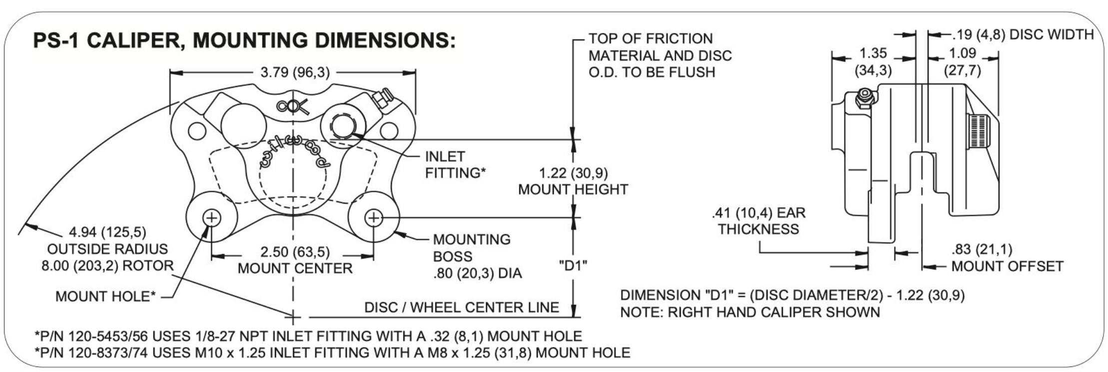


Figure 13. Wilwood PS-1 brake calipers. Surface area of $2in^2$.

Brake Pads:

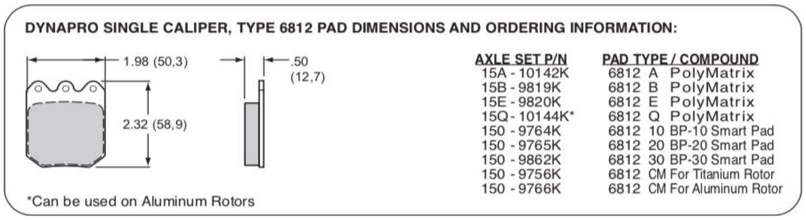


Figure 14. Front brake pads with a 3in^2 surface area

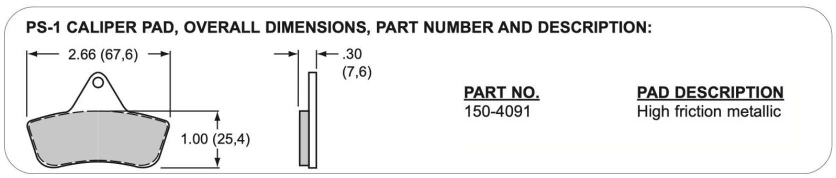


Figure 15. Rear brake pads with a 3in^2 surface area

Master Cylinders:

The Tilton 76-series master cylinder are an aluminum bodied master cylinder with a AN-4 (7/16”-20) inlet port adapter, which accepts AN-4 fittings to allow for remote mounted reservoirs for straightforward packaging in the cockpit. We decided on a ¾” bore size as a midway point for brake biasing when tuning the brakes.

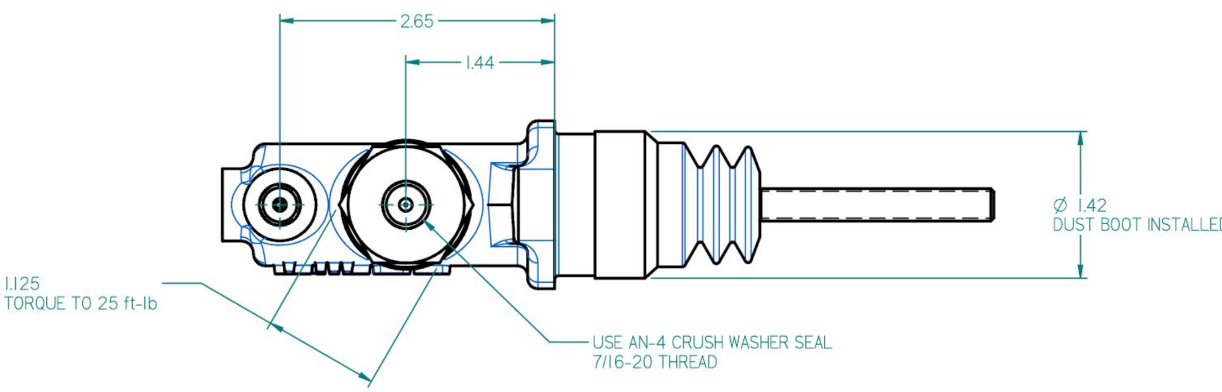


Figure 16. Master cylinder AN-4 port shown

Dual AN-3 outlet ports give plumbing flexibility and allow brake pressure sensors or brake light switches to be installed.

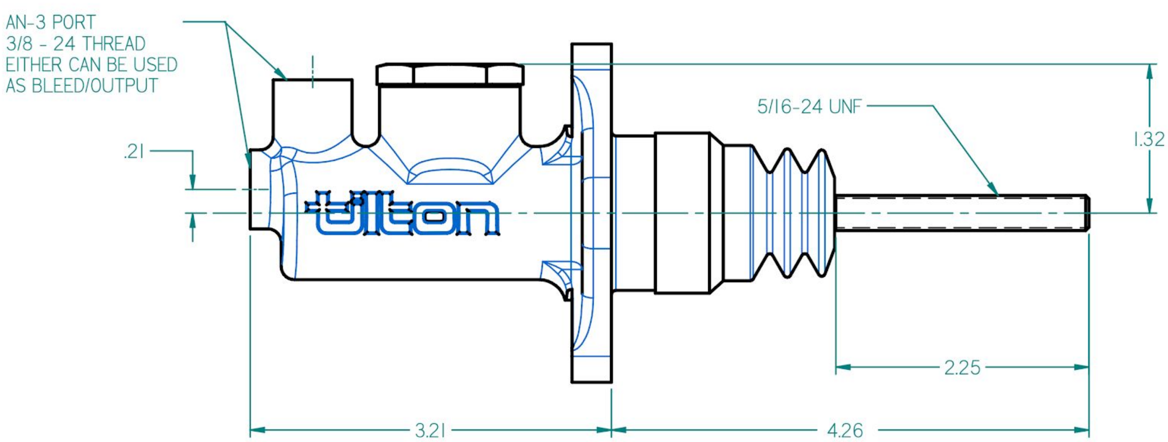


Figure 17. Master cylinder AN-3 port shown

The top outlet port is compatible with both AN-3 and banjo fittings to connect to the remote reservoir. *To increase stopping power or reduce the pedal effort required: A) decrease the master cylinder bore size, B) increase the pedal ratio.* [1]

Reservoir:

The 3-chamber remote reservoir allows remote mounting which is required for underfoot pedal assemblies. Remote mounting also provides ease of packaging in the cockpit and don't have to mount individual reservoirs per master cylinder. The rear brake chamber has a volume of 8.9 oz (263 mL) and the front brake chamber has a volume of 10.3 oz (313 mL). The reservoir connects to the brake master cylinder with AN-4 braided lines. Note: only PTFE, EPDM or SBR house can be used.

[1] https://www.onallcylinders.com/2014/08/07/pedal-pushers-figure-pedal-ratio-master-cylinder-bore-size/

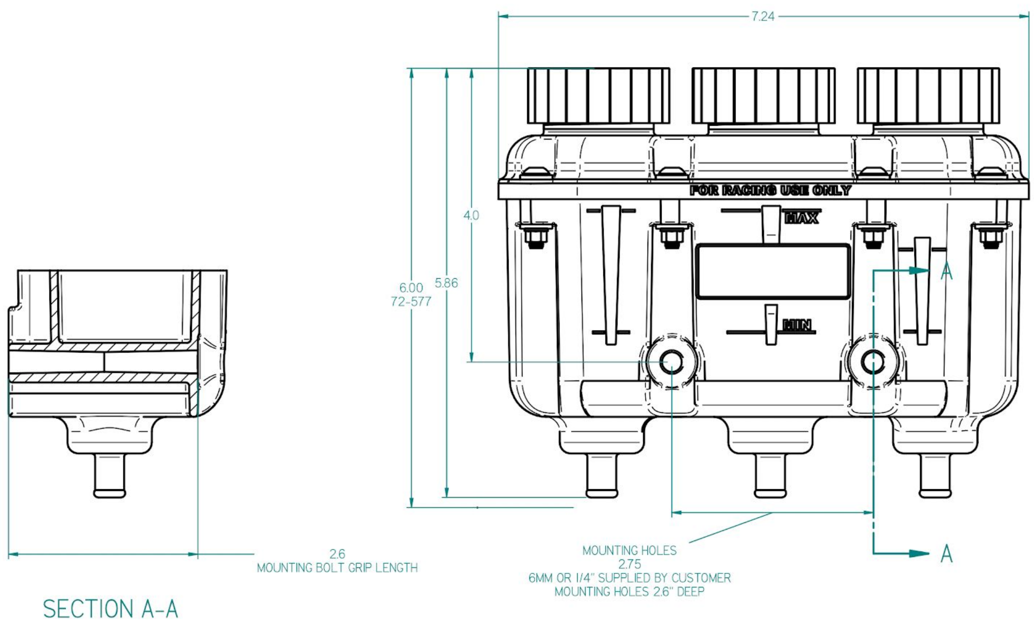


Figure 18. 3-chamber reservoir for brake fluid

Pedal Assembly:

Tilton 72-618 underfoot throttle and brake pedal assembly was chosen for its compact packaging, variable pedal ratio, and is compatible with the 76-series brake master cylinders (Fig. 20). The aluminum body has adjustable foot pads allowing pedal ratio adjustment from 5.4:1 to 6.9:1. A 7/16"-20 balance bar allows front and rear brake bias adjustment. The master cylinder is affixed to the pedal assembly via a 2.25" (center-to-center) front flange mount, which is an industry standard. Adjustable throttle pedal stops limit pedal movement in both directions which is an FSAE requirement. A mechanical linkage is connected to the throttle pedal and regulates airflow to the combustion chamber of the engine via the the position of the butterfly valve on the throttle body.

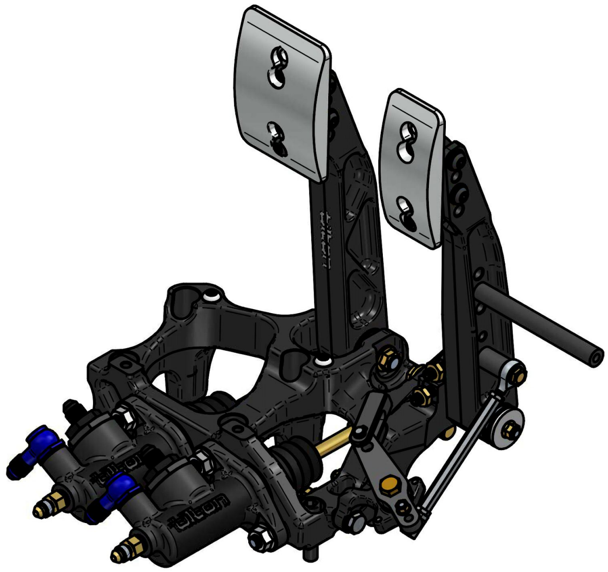


Figure 19. Tilton 72-618 Underfoot Throttle/Brake Pedal Assembly

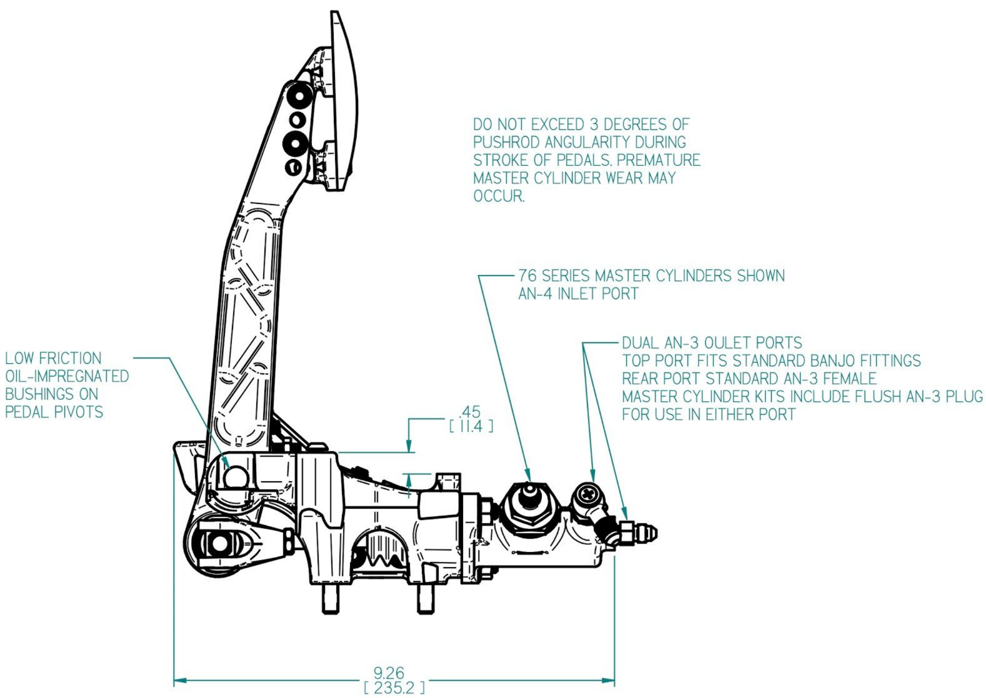


Figure 20. Brake plumbing instructions for the pedal assembly: AN-3 inlet ports, dual AN-3 outlet ports (top uses banjo fitting, rear uses standard AN-3 female fitting)

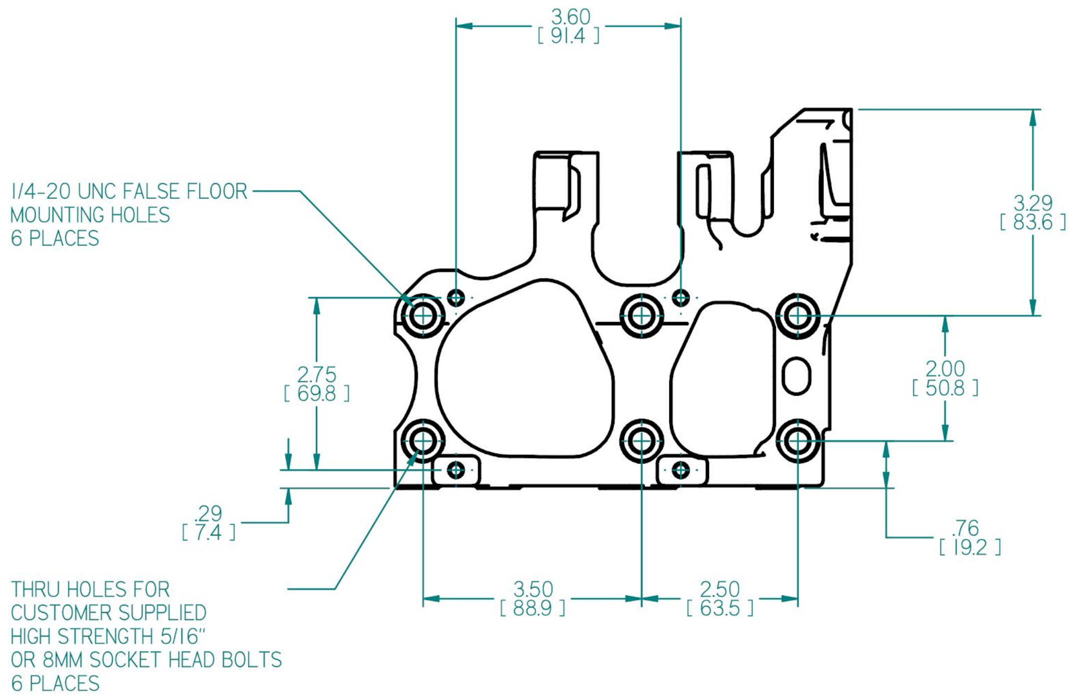


Figure 21. Mounting fixture for the Tilton 72-618 underfoot throttle/brake pedal assembly: false floor mounted via (6) ¼-20 UNC, mounting plate fixed with (6) 5/16 socket head bolts.

Wheels and Tires

The choice of the tire size was between 10" and 13" diameter as those were the most commonly available sizes that could be found. The decision was made to go with $24.50" \times 8.00" - 13"$ Hoosier tires. This provided ample space for packaging components in the assembly, which for a new team was thought to serve future iterations better than focusing on reducing the rolling moment of inertia or the mass of the assembly.

Cockpit

For the floor close-out, FSAE requires four rules to be met. The material chosen was a hot rolled steel sheet S112 (0.105 thick), which is a solid/non-brittle material, meeting rule T.3.4.1. The choice was based on the material's ease to weld, form, drill and low price. The group designed and fabricated three plates to close the floor, and gaps do not exceed the maximum (T.3.4.4). The closeout extends from the foot area to to the firewall (T.3.4.2), and does not leave room for track debris to enter the car (T.3.4.1).

For the seat, the group chose a standard go-kart seat for testing purposes, attached to the floor. Even though not fabricated, the permanent solution was designed according to FSAE guidelines. The design consists of two metal plates angled in 110°, covered by a creafoam to accommodate the driver. The design meets meets FSAE rule T.3.3.1 by having the lowest point of the driver's seat no lower than the bottom surface of the lower frame rails, and two longitudinal tubes passing underneath the seat. The following three rules concern insulation, which can only be accomplished with the completion of the Powertrain Subsystem. As far as visibility is concerned, with both temporary and designed solutions, when seated in normal driving position, the driver has a field of vision of 100° to either side. In addition, both solutions have the proper spacing to fit safety harness. The seatbelt chosen, was a 5-point system, which consists of two lap belts, two shoulder straps, and one anti-submarine strap, and meets rule T.4.1.1.

As previously mentioned, there is work to be done by future Senior Design Teams. With the completion of other subsystems, new groups will be able to pay closer attention to the heat transfer happening between the driver's compartment and the powertrain. Proper insulation needs to be installed to meet FSAE guidelines. Once controls are installed, a tractive system firewall (EV only), will also need to be installed to separate the driver compartment and all tractive system components, including any HV wiring. Other safety instruments must also be incorporated to the car, once the cockpit is completed.

Prototype Testing

Test 1: System Free Play

Test Overview and Objectives

This test measures a parameter called the steering system free play, which is defined as the maximum angular displacement of the steering wheel in response to driver input with the two front tires locked in position. In an ideal scenario, the two front tires of the car would turn instantly in response to angular movement of the steering wheel from driver input, resulting in zero degrees of free play in the system. In reality, however, due to imperfections in the assembly of the system members, the steering wheel will experience some free movement before the front tires begin to turn in response to driver input. In our project charter, one of our key objectives was to strive to meet all Formula SAE guidelines stated in the FSAE rulebook, which requires a steering system free play of no more than seven degrees for a competition-ready steering system [1]. The objective of this test is to assess our steering system free play and ensure that it does not exceed the seven-degree maximum required by the FSAE rulebook. System free play is a vital parameter in assessing our car's ability to safely navigate the hairpin turn of the formula SAE Autocross and Skidpad events. The less system free play there is, the more control the driver has when navigating the turn, and the higher the chance our car will navigate the turn safely without issues.

Test Scope and Test Plan

To ensure the validity of this test, the front wheels of the car were completely constrained from all motion. Since the neutral position of the joints and connections of the system are unclear, it was necessary to define the initial measurement of the angular position of the steering wheel to be its leftmost extreme position with front tires fixed, and the final angular position measurement at its rightmost extreme angular position. The total degrees of system free play is defined in this test as the difference between initial and final steering wheel angular position measurements. Steering wheel angular position was measured using a Tacklife PRO laser measure with electronic angle sensor for increased precision relative to analog methods.

Acceptance Criteria

In order to meet the requirements of this test, after completing 10 trials, the system must exhibit no more than the maximum acceptable seven degrees of free play as specified in the FSAE rulebook with a 95% confidence interval from the mean.

Test Results and Evaluation

As stated in the preceding sections, 10 trials were performed of this test. Full test results are shown in Table T1a, along with a summary of statistics of the results in Table T1b, both located in Appendix F. The steering system degrees of play ranged from 3.2° to 7.9°, with mean and standard deviation of 5.0° and 1.3°, respectively, and 95% confidence interval of 4.2°-5.8°. This test can be considered a pass, since with a double-sided confidence interval of 95%, the degrees of play does not exceed the Formula SAE requirement of 7°. The maximum recorded value of 7.9°, however, exceeds the maximum allowable value. However, since this is the only value that is more than two standard deviations from the mean, it can be considered an outlier that is uncharacteristic of the overall behavior of our system and is likely due to measurement error. Additional testing trials are necessary to confirm the that this value is in fact an outlier and can be disregarded.

Test 2: Static Steering Test

Test Overview and Objectives

The purpose of this test is to ensure that the operating range of the steering system will allow the car to compete in the Autocross and Skidpad events in the Formula SAE competition. In our previous design work, taking into account our car's wheelbase and track width and the geometry of the FSAE hairpin turn, we determined the minimum geometric inner and outer toe angles required for our car to successfully navigate tightest hairpin turn of the Formula SAE circuit. These angles were found to be 33.26° for the outer toe angle ϕ_{out} , and 23.40° for the inner toe angle ϕ_{in} [2]. In this test we measured inner and outer toe angles of the front tires across the entire range of motion of the steering wheel in order to assess whether or not there exists a steering wheel position in which the toe angles exceed the established minimum requirements for the car to navigate the hairpin turn. The existence of this critical steering wheel position serves as preliminary confirmation that our steering design will allow the car to compete in the Formula SAE Autocross and Skidpad events.

Test Scope and Test Plan

Test Setup

The setup of this test is shown in Figure 24, with θ representing the angular position of the steering wheel with respect to the vertical axis, and ϕ_{in} and ϕ_{out} representing the toe angles of the inner and outer front tires with respect to the horizontal axis defined by the car's wheelbase. The input of this test is the angular position of the steering wheel, θ which begins at the neutral position of θ_0, equal to zero degrees, and increases by interval $d\theta$ for each successive test measurement. The steering wheel test position n is defined as the number of intervals of $d\theta$ from the neutral position θ_0. Each test position n defines a steering wheel angular position of $\theta_0 + nd\theta$. Corresponding to each test position is left and right toe angle of $\phi_{in,n}$ and $\phi_{out,n}$.

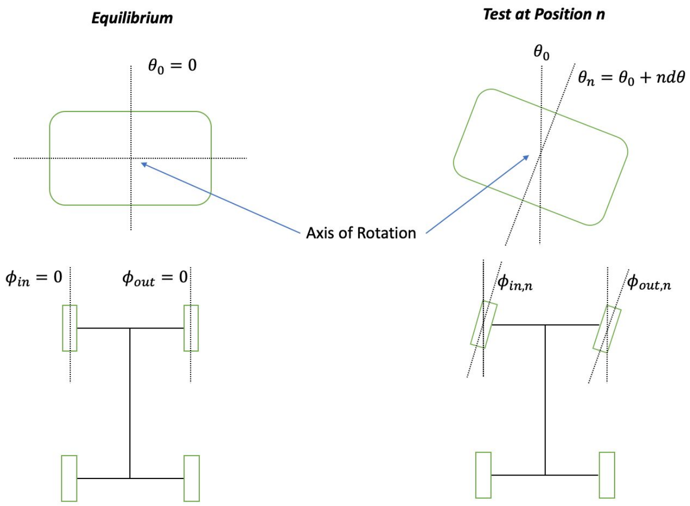


Figure 24. Test 1 setup with defined variables of steering wheel angular position and inner and outer toe angles

Defined in below are the governing equations of this test, along with interval size and range in test positions, with negative θ corresponding to counterclockwise rotation of the steering wheel and positive θ corresponding to clockwise rotation of the steering wheel.

$$\theta_n = \theta_0 + nd\theta$$

$$-7 \leq n \leq +7$$

$$\theta_0 = 0°$$

$$d\theta = 15°$$

As shown above, maximum and minimum test positions were chosen to be n = -7, and n = +7, corresponding to maximum and minimum steering wheel angular positions of ndθ = -105° and ndθ = +105°.

Ideal Test Conditions:

In order to ensure valid results, the following conditions were observed and met for each trial of this test:

- Vehicle completely stationary on a level surface
- Changes in toe angle/rotational movement of the front tires caused only by rotation of the steering wheel with no additional forces applied to the tires or to any other portion of the vehicle
- Testing begins at neutral steering position θ_0 of zero degrees, corresponding to toe angles ϕ_{in} and ϕ_{out} of zero degrees
- Consistent interval $d\theta$ of 15° with respect to vertical axis observed for each successive jump n in test position.

Instrumentation and Engineering Tools

In this test, angular position for both the steering wheel and the front tires was measured using a Tacklife PRO laser measure with electronic angle sensor. An electronic method of angular position measurement was chosen in favor of an analog method for increased accuracy and precision. In terms of engineering tools, this test requires a top-level understanding of the intended functionality of our steering system to transfer the angular motion of the steering wheel to linear motion of the rack's clevis ends, and ultimately to transformed angular motion of the front tires. It also requires the skill of defining angular position of individual components in a three-dimensional coordinate system, and an understanding of how the components of our system interact with each other in three-dimensional space.

Data Collected

As stated in the preceding sections, for each steering wheel test position n from n = -7 to n = +7, corresponding left and right toe angles $\phi_{in,n}$ and $\phi_{out,n}$ were recorded. Positive and negative (clockwise and counterclockwise) critical test positions were then identified, in which corresponding inner and outer toe angles exceed the required toe angles for the car to successfully navigate the hairpin turn of the Formula SAE circuit. It is important to note that, for positive values of n, the wheel is turned clockwise, and inner toe angle corresponds to the right wheel and outer toe angle corresponds to the left wheel. For negative values of n, however, the wheel is turned counterclockwise and inner toe angle corresponds to the right wheel and outer toe angle corresponds to the left wheel.

Acceptance Criteria

In order to successfully pass this test, there must exist clockwise and counterclockwise critical steering wheel test positions n and -n in which inner and outer toe angles $\phi_{in,n}$ and $\phi_{out,n}$ exceed the required toe angles for the car to successfully navigate the hairpin turn of the Formula SAE circuit, which are 23.40° and 33.26°, respectively. These required toe angles were determined in our previous design work based on the dimensions of our particular vehicle and the radius of curvature of the hairpin turn [1].

Test Results and Evaluation

Full test results are shown in Table T2a, along with critical steering test positions assessed against the requirements of the FSAE circuit in Table 2b, both located in Appendix F. Test positions n = +6 and n = -7 were identified to be the test positions in which inner and outer toe angles exceed those required to successfully navigate the hairpin turn of the FSAE circuit. This test can be considered a pass, since for both clockwise and counterclockwise rotation of the steering wheel, inner and outer toe angles exceeded the minimum required angles to navigate the circuit by more than 4%. As a result, we predict that our steering system has the capability to successfully navigate the car around the tightest hairpin turn of the FSAE circuit. However, the general spread of toe angle data across all steering wheel test positions was found to be asymmetrical, and further tuning of the system is required to ensure that it behaves identically when the steering wheel is rotated clockwise and when it is rotated counterclockwise. Nonetheless, test results confirm that our system is sound overall and we predict no additional fabrication required to achieve a symmetrical distribution.

Test 3: Dynamic Steering Test

Test Overview and Objectives

This test will require the car's steering system to guide it through a course including turns to mimic the hairpin turns it will have to steer through in an FSAE competition event.

Test Scope and Test Plan

Because previous tests suggest this test will be passed by providing information about the angular deflection, they do not account for the normal forces and alignment capabilities that are also needed for the car to steer itself. Therefore, any and all requirements for the steering system to function in a driving situation will be tested, since the final requirement is that the car steer itself through this course. This course itself will consist of a short straight track of 10 meters which develops into a 9 meter-diameter circle indicated by markers. The markers will be laid so that a track width of 4 meters is maintained throughout the course (2 meters on either side of the intended centerline). The car may be pushed by testers from the tail end of the car, but force may only be applied in the forward direction so as not to assist in the steering, and one tester must sit in the driver's seat in order to operate the steering wheel. The full test will consist of six runs, with three in each starting direction of the course.

Acceptance Criteria

If the driver is able to steer the car only using the steering wheel and the strictly forward directed force of the team members outside the car, and for all runs, the test will be considered a success.

Test Results and Evaluation

At the time that this report is being written, this test has not been completed. Upon moving the car for the test, there were two complete failures, one in both of the rear right side A-arms. The failures in both cases occurred where two members were welded together. Both the welds were done by the previous senior design group and that is one of the reasons why these failures were so devastating and unexpected. However, even though the results of this test will not be displayed in this version of the Final Design Report, these results will be obtained by the Final Presentation, and there will be an addendum added to the report after the test is completed. The damage done to the A-arms have already been repaired and our group is still planning on completing Test 3, but were simply unable to perform the test before the due date of our Final Design Report.

Test 4

Test Overview and Objectives

This test verifies the effectiveness of the quick release system, along with cockpit ergonomics to ensure that the driver is safely able to exit the car within a certain amount of time, defined by the International Automobile Federation (FIA).

Test Scope and Test Plan

This test has been designed to ensure that the driver is able to exit the car within minimum time. This test consists in a sample population of 30, simulating an emergency situation. Each person involved will be wearing all safety harness (i.e. seatbelt, head restraint, etc) required by FSAE 2018-2019 rules and will have to safely unbuckle, release the steering wheel and exit the car. Several features of the cockpit were evaluated, such as ergonomics, seatbelt, head restraint, seat, steering system and all the necessary safety equipment. Mean, maximum, and minimum times were then obtained.

Acceptance Criteria

According to the International Automobile Federation (FIA), the maximum allowed time to exit the cockpit of is 9 seconds.

Test Results and Evaluation

Three group members divided 30 samples between themselves. The results are shown in the table below:

Table T4a. The table shows 30 instances recorded divided evenly by three group members

Test No.	Subject 1 (seconds)	Subject 2 (seconds)	Subject 3 (seconds)
1	5.63	4.60	2.59
2	4.02	3.98	2.83
3	3.86	5.14	2.66
4	3.65	3.62	2.35
5	3.49	3.38	5.52
6	3.16	3.75	2.23
7	2.92	5.26	2.33
8	3.98	2.96	2.25
9	3.23	2.66	2.20
10	2.92	3.61	2.02

As seen above, 100% of the times a group member tried to safely exit the car, in simulation of an emergency, the times taken to release the steering wheel, unbuckle the seatbelt and safely exit the car, were below the 9 seconds required by FIA. As seen in the summary below, the maximum time among all samples was 5.63s for Subject 1, whereas the minimum was 2.02 for Subject 3. The mean found between all samples was 3.43s and the standard deviation was 1.01s.

Table T4b. Summary of results

	Average (s)	Minimum Time (s)	Maximum Time (s)
Subject 1	3.69	2.92	5.63
Subject 2	3.90	2.66	5.26
Subject 3	2.70	2.02	5.52
Net	3.43	2.02	5.63

Conclusion

By the standard FSAE guidelines, and by the objectives laid out in the Project Charter, the steering system design that we implemented in our car is a success. Due to our smart design choices, we were able to implement a design that partially required very basic shop tools and equipment to fabricate in house, and party relied on store bought parts which were acceptable to use by FSAE standards. In addition to following all the guidelines, the design was also compact enough to fit into the tight frame of the chassis, fulfilling our geometric constraints. The NARRco Rack V1 offers 3.46 in of linear movement for every revolution of the steering wheel which allows sharp turns to be made with minimal effort by the driver. The fact that the car is also relatively light, reduces the amount of resistance felt by the driver, which is necessary due to the absence of a power steering system.

In addition to meeting our design goals, the implemented steering system also passed the majority of the tests that we created to measure its validity and accuracy to our theoretical model. In Test 1 we was found that the degrees of play our steering system displayed were typically less than 5 degrees, but consistently less than 7 degrees within the 95% confidence interval that we were aiming to get in this test. Test 2 results show that in both clockwise and counterclockwise directions inner and outer toe angles exceed the minimum acceptable criteria of 23.40 degrees for the inner toe angle and 33.26 degrees for the outer toe angle. This means that our car can be can be expected to make it around every curve in a FSAE regulation course. This also helps partially make up for the fact that we were unable to complete Test 3 which was a dynamic test of our car's ability to navigate the same hairpin turn on which Test 2 was based. From the measurements taken in Test 2, our group expects the car to be able to complete Test 3, which is based off of the hairpin turn of the FSAE competition circuit. These inner and outer toe angles are the theoretical angles required for the car to make it around the turn. Finally, Test 4 helped us achieve some safety standards which are required by the FIA. In order to pass this test, the occupant of the vehicle needed to be able to remove their seatbelt, remove the steering wheel, and remove themselves from the vehicle in no longer than nine seconds. The results of the test are well within the criteria with no one exiting the vehicle in more than 5.63 seconds and people exiting in an average of 3.43 seconds.

It has not been established if the current steering system can successfully navigate a course mimicking the 10m-wide FSAE competition track because the dynamic testing (Test 3) could not be performed due to issues outside of the steering system. The future work necessary to complete this objective is to assess all previous welds connecting components of the powertrain and suspension (A-arms in particular). Once the weak welds have been strengthened, it will be safe to perform Test 3 and evaluate dynamic steering performance. At some point, it will become relevant to perform Test 3 again with a working powertrain. The scope of Test 3 is limited by the slow speeds that the car moves at when being pushed by students rather than propelled by the gas engine.

While the brake system was successfully planned out and all components have been purchased, the system has yet to be assembled. Future students should consider the configuration of all major car components before attaching plumbing and fully integrating the brake system into the car. Finally, the ECU, powertrain, and suspension must be completed, tuned, and integrated in order to prepare the car for eventual entry into the annual FSAE competition. This year's team has performed research, brainstormed, and written plans for the completion of each of these subsystems.

Appendices

Appendix A: Setup, operation and safety instructions

Instructions for Quick Release Use:

Attaching:

1. Line up the grooves of the quick release so that the largest notch matches up with the largest slot
2. Push two pieces together until there is an audible "click"
3. Quick release is now in place and connection will not separate until the user chooses to separate them

Detaching:

1. Locate the yellow section of the release which fits directly around the shaft
2. Pull this yellow piece in the opposite direction of the shaft and at the same time pull the two pieces apart
3. The shaft and release mechanism should detach from each other easily

Appendix B: Figures, design schematics and drawings

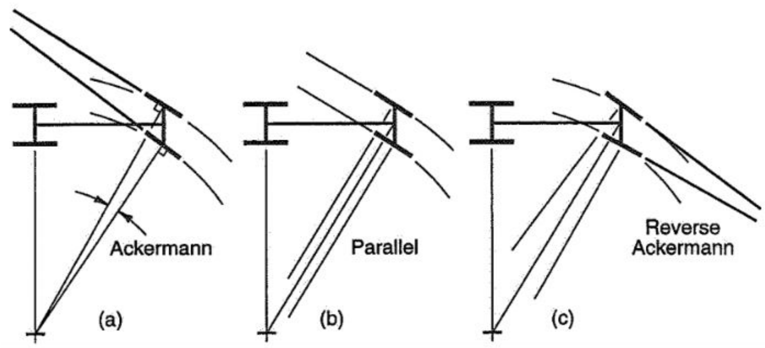


Figure 1. Steering geometry variations: (a) Ackerman, (b) parallel, and (c) reverse Ackerman Steering

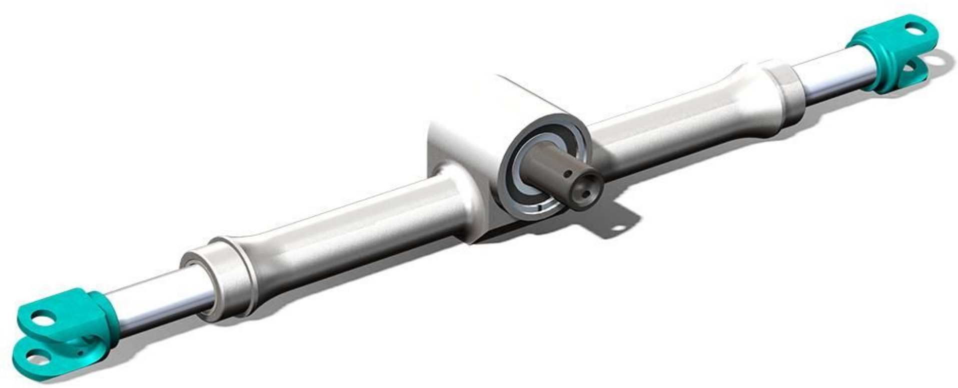


Figure 2. 3D Rendering of the [NARRco Rack V1]

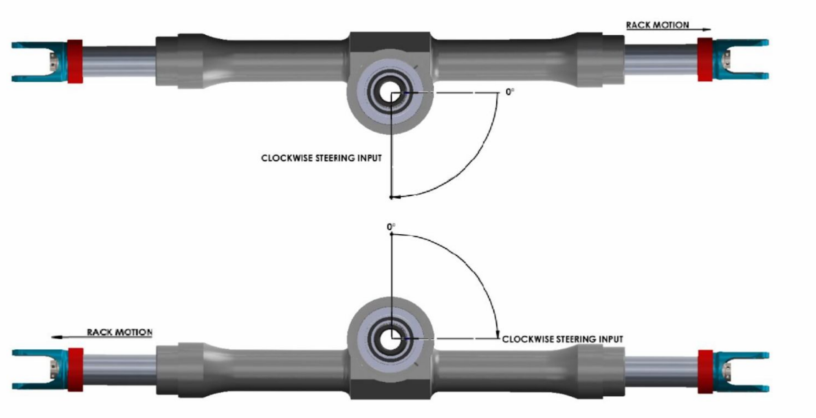


Figure 3. NARRco Rack-and-Pinion Travel Direction

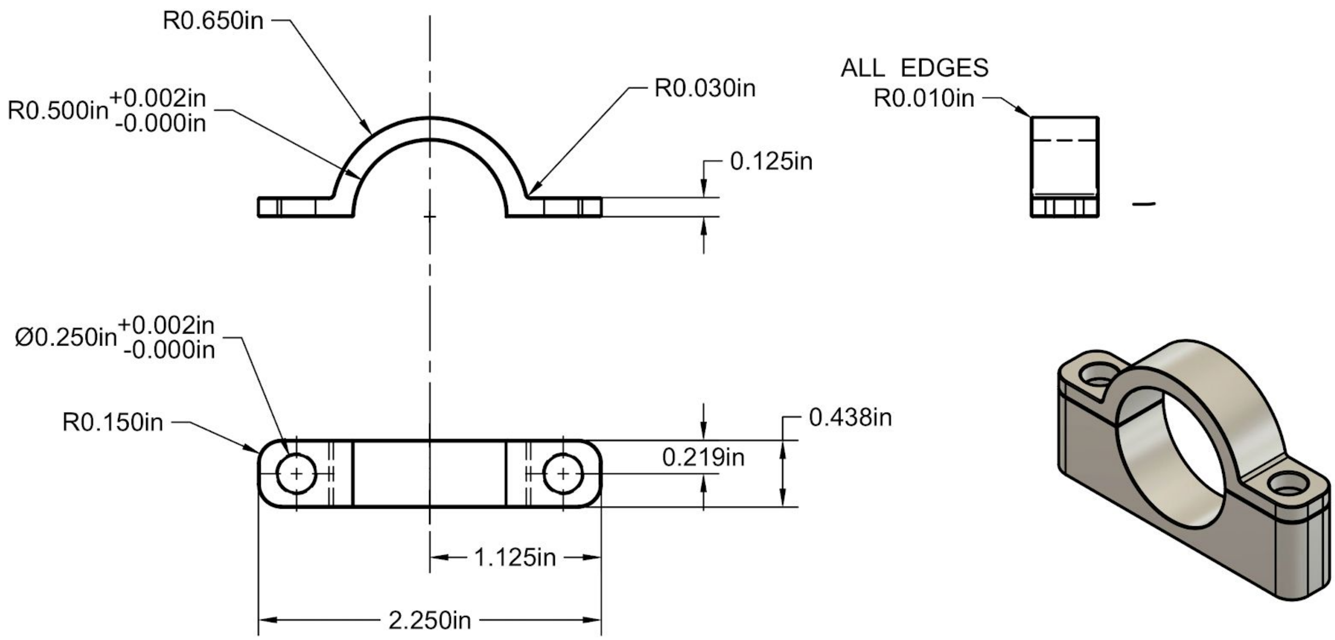


Figure 4. Engineering drawing of the rack support

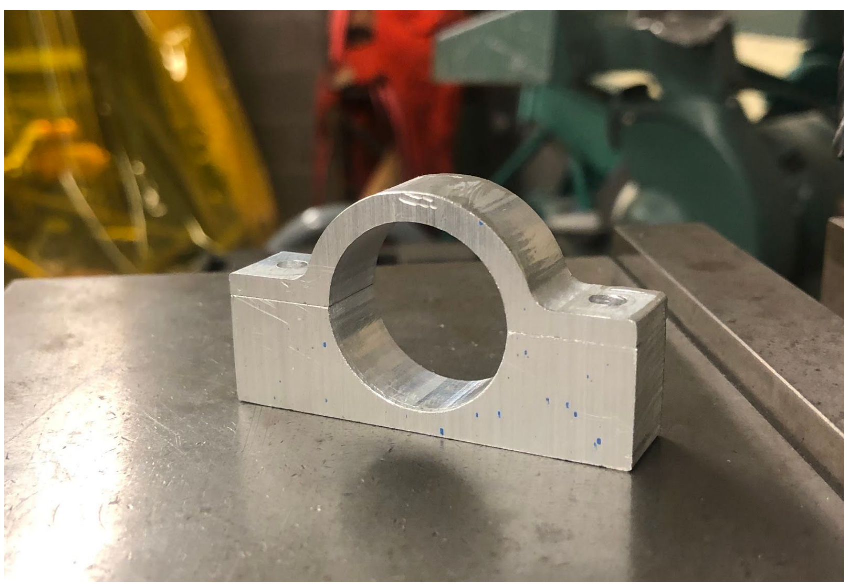


Figure 5. Fabricated rack support made from 6061 aluminum

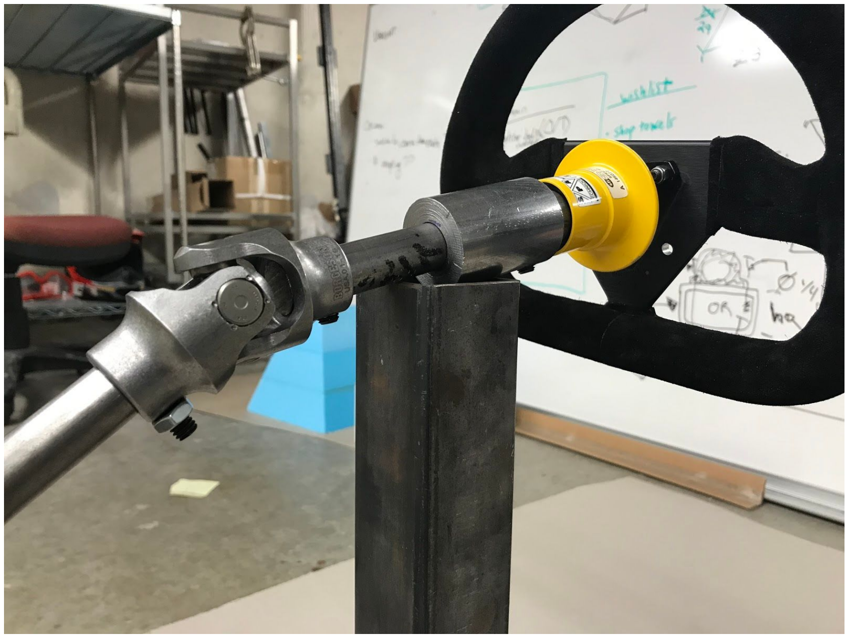


Figure 6. U-joint and shaft coupling assembly

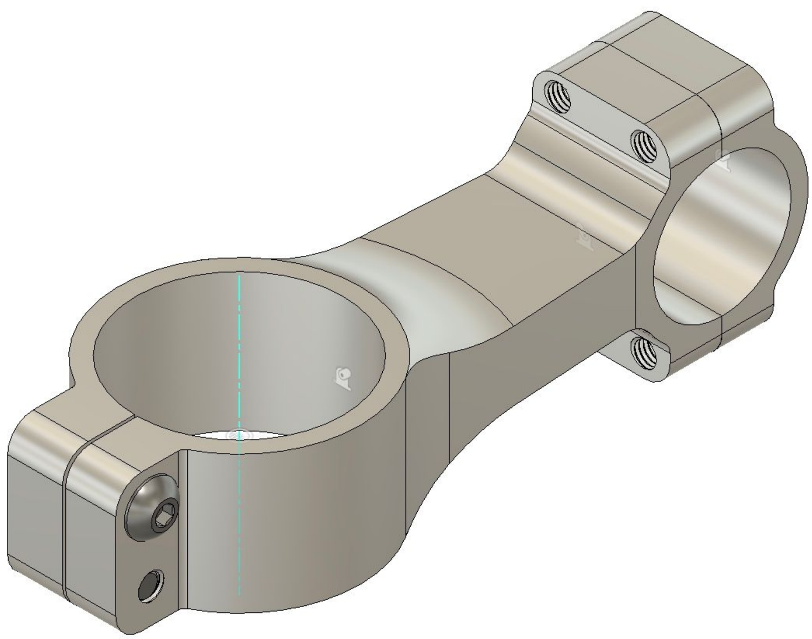


Figure 7. Fusion 360 model of the STEERING-RACK-SUPPORT

Figure 8. CNC manufactured steering column support fixed to the vehicle chassis

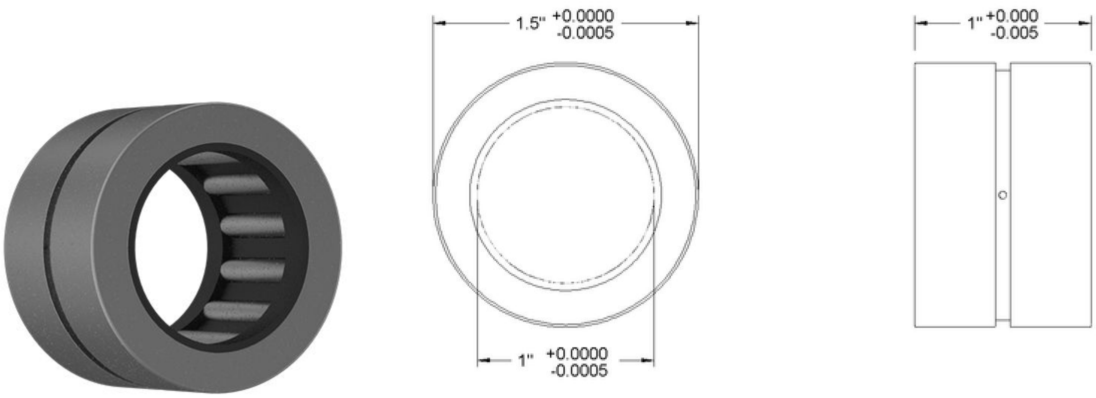


Figure 9. Needle roller bearing

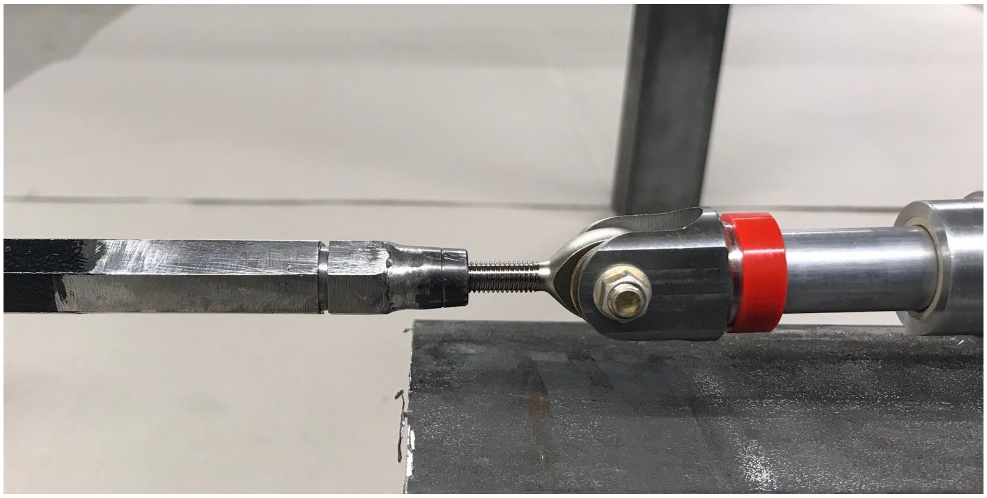


Figure 10. Steel hex tie rods showing the clevis rack connection via a ¼-28 spherical rod ends creating a heim joint.

Figure 11. Steering end links making the tie rod connection to the uprights. The tie rod end links are connected to the upright at an angle with the horizontal plane of less than 5 degrees.

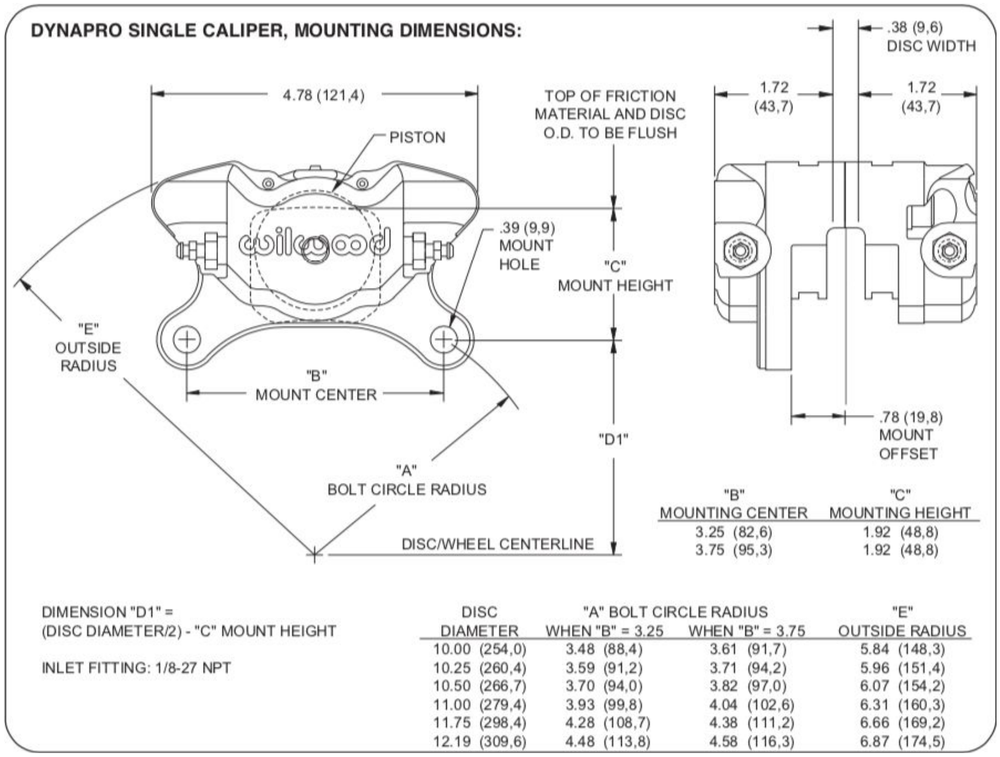


Figure 12. Wilwood Dynapro brake calipers. Surface area of 3in^2.

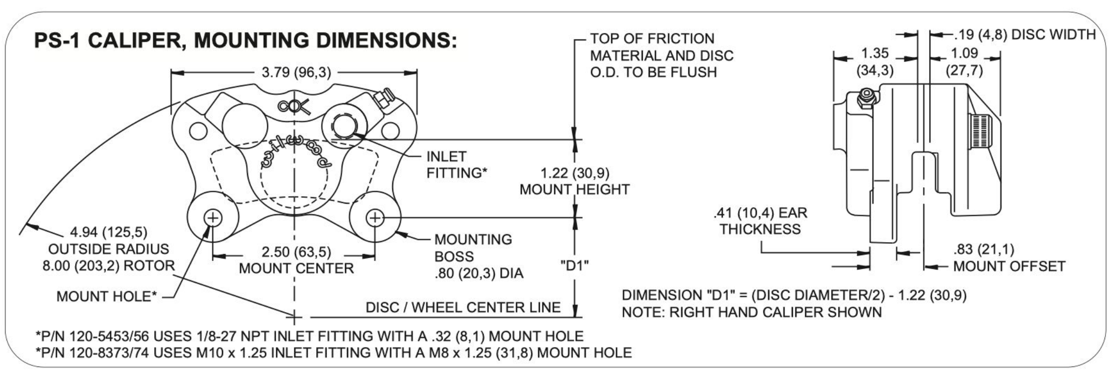


Figure 13. Wilwood PS-1 brake calipers. Surface area of $2in^2$.

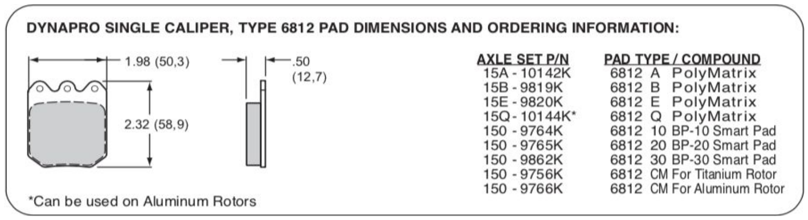


Figure 14. Front brake pads with a $3in^2$ surface area

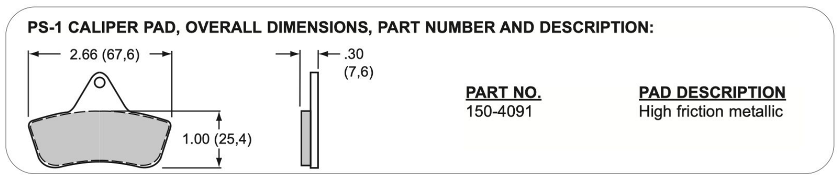


Figure 15. Rear brake pads with a $3in^2$ surface area

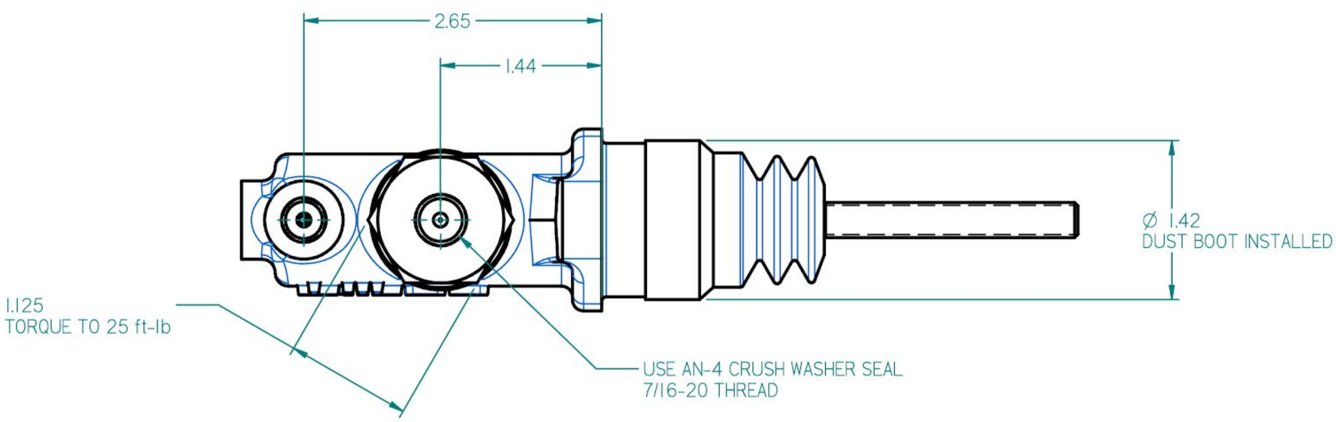


Figure 16. Master cylinder AN-4 port shown

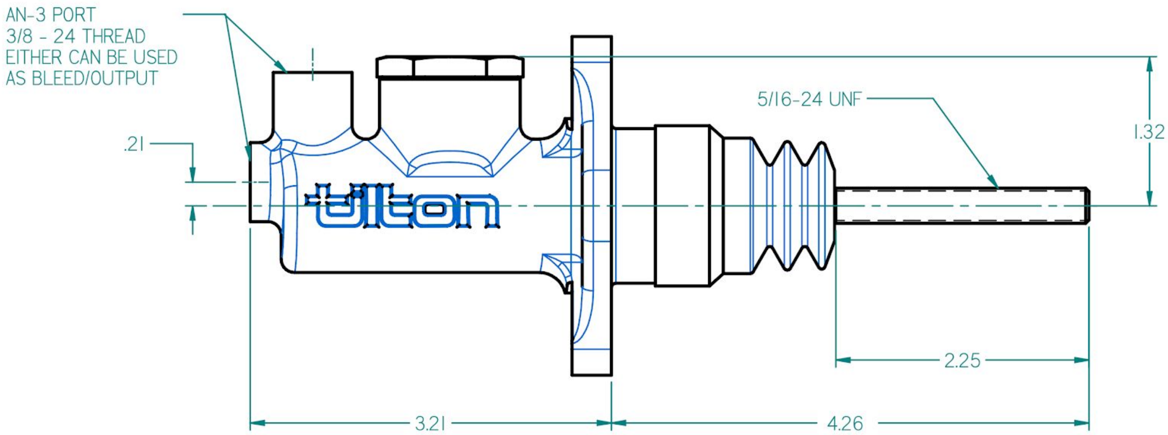


Figure 17. Master cylinder AN-3 port shown

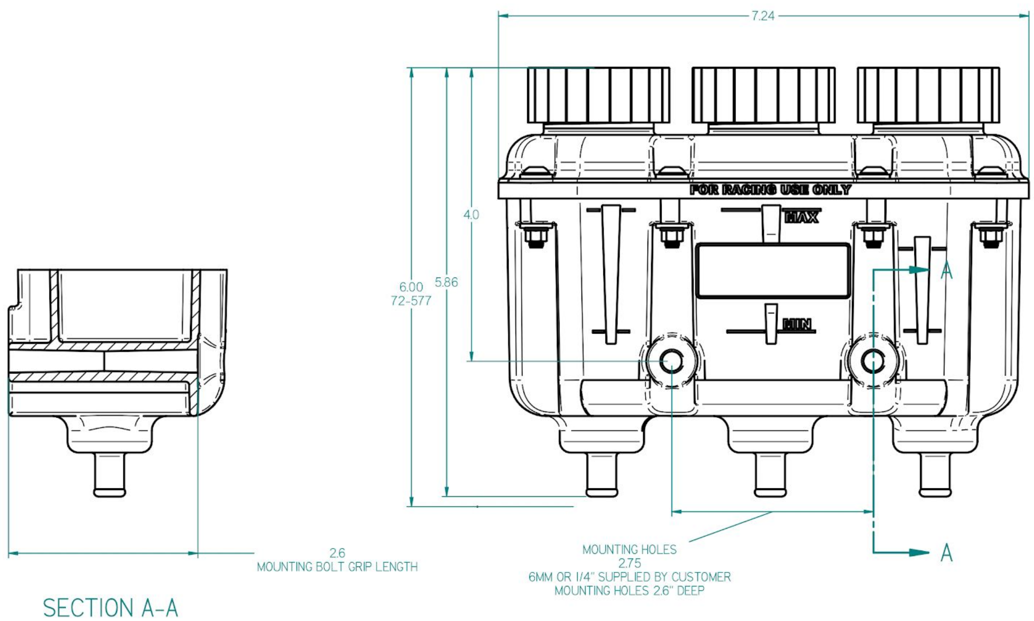


Figure 18. 3-chamber reservoir for brake fluid

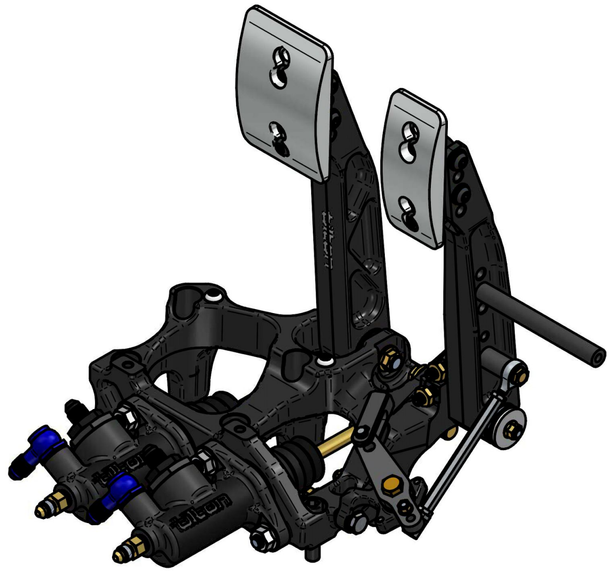


Figure 19. Tilton 72-618 Underfoot Throttle/Brake Pedal Assembly

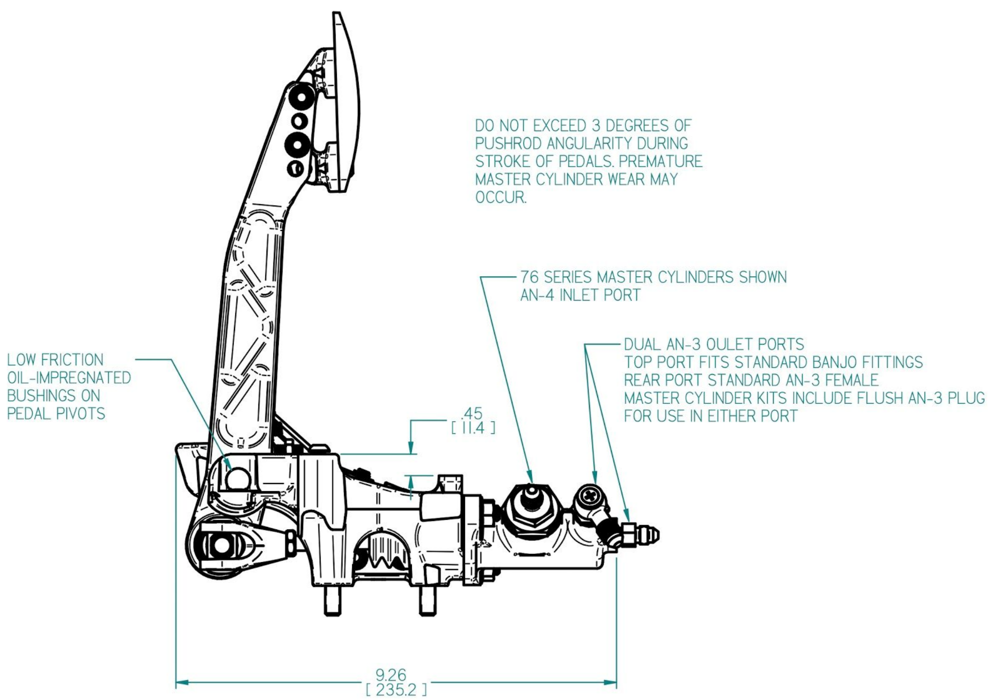


Figure 20. Brake plumbing instructions for the pedal assembly: AN-3 inlet ports, dual AN-3 outlet ports (top uses banjo fitting, rear uses standard AN-3 female fitting)

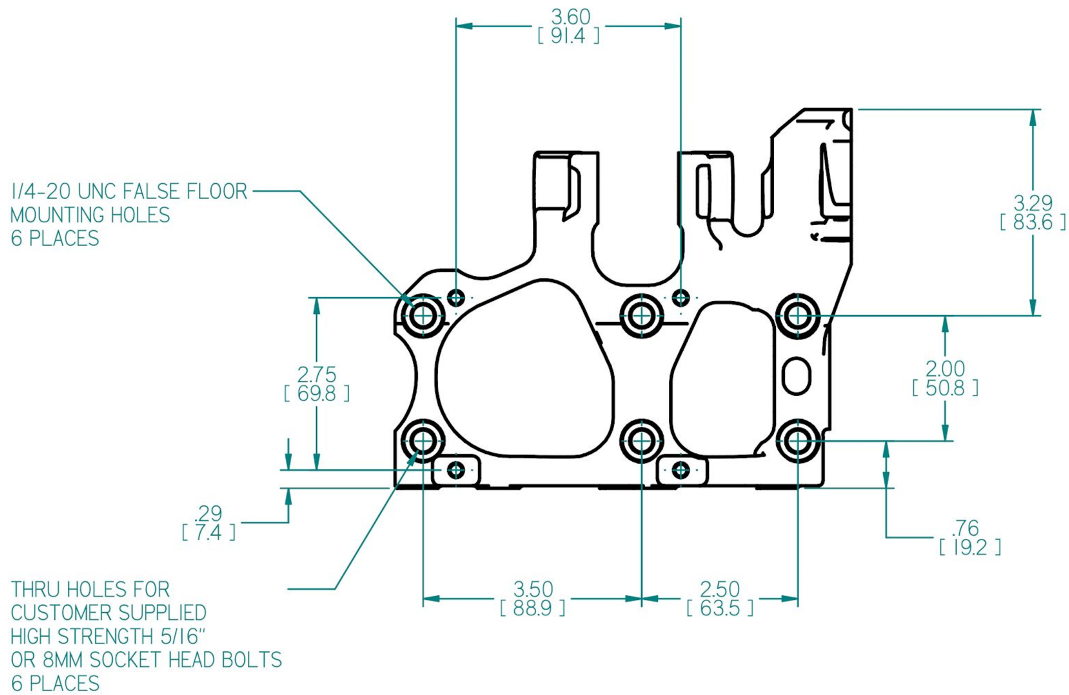
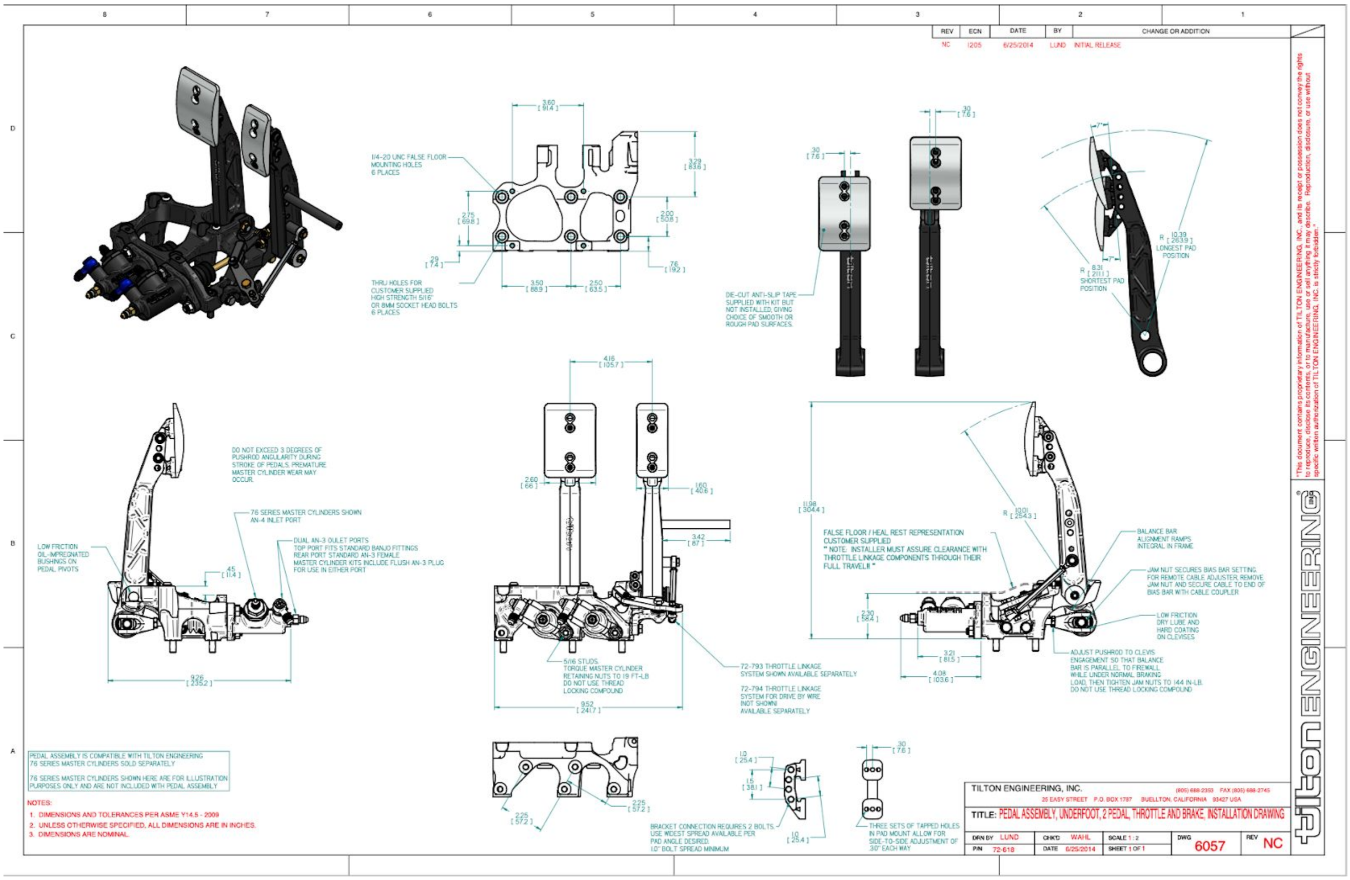


Figure 21. Mounting fixture for the Tilton 72-618 underfoot throttle/brake pedal assembly: false floor mounted via (6) ¼-20 UNC, mounting plate fixed with (6) 5/16 socket head bolts.


REV ECN DATE BY CHANGE OR ADDITION
NC 1205 6/25/2014 LUND INITIAL RELEASE
1/4-20 UNC FALSE FLOOR MOUNTING HOLES 6 PLACES
THRU HOLES FOR CUSTOMER SUPPLIED HIGH STRENGTH 5/16" OR 8MM SOCKET HEAD BOLTS 6 PLACES
DIE-CUT ANTI-SLIP TAPE SUPPLIED WITH KIT BUT NOT INSTALLED, GIVING CHOICE OF SMOOTH OR ROUGH PAD SURFACES.
R 10.39 [263.9] LONGEST PAD POSITION
R 8.31 [211.1] SHORTEST PAD POSITION
DO NOT EXCEED 3 DEGREES OF PUSHROD ANGULARITY DURING STROKE OF PEDALS. PREMATURE MASTER CYLINDER WEAR MAY OCCUR.
76 SERIES MASTER CYLINDERS SHOWN AN-4 INLET PORT
DUAL AN-3 OULET PORTS TOP PORT FITS STANDARD BANJO FITTINGS REAR PORT STANDARD AN-3 FEMALE MASTER CYLINDER KITS INCLUDE FLUSH AN-3 PLUG FOR USE IN EITHER PORT
LOW FRICTION OIL-IMPREGNATED BUSHINGS ON PEDAL PIVOTS
5/16 STUDS. TORQUE MASTER CYLINDER RETAINING NUTS TO 19 FT-LB DO NOT USE THREAD LOCKING COMPOUND
72-793 THROTTLE LINKAGE SYSTEM SHOWN AVAILABLE SEPARATELY
72-794 THROTTLE LINKAGE SYSTEM FOR DRIVE BY WIRE (NOT SHOWN) AVAILABLE SEPARATELY
FALSE FLOOR / HEAL REST REPRESENTATION CUSTOMER SUPPLIED
" NOTE: INSTALLER MUST ASSURE CLEARANCE WITH THROTTLE LINKAGE COMPONENTS THROUGH THEIR FULL TRAVEL!! "
BALANCE BAR ALIGNMENT RAMPS INTEGRAL IN FRAME
JAM NUT SECURES BIAS BAR SETTING. FOR REMOTE CABLE ADJUSTER, REMOVE JAM NUT AND SECURE CABLE TO END OF BIAS BAR WITH CABLE COUPLER
LOW FRICTION DRY LUBE AND HARD COATING ON CLEVISES
ADJUST PUSHROD TO CLEVIS ENGAGEMENT SO THAT BALANCE BAR IS PARALLEL TO FIREWALL WHILE UNDER NORMAL BRAKING LOAD, THEN TIGHTEN JAM NUTS TO 144 IN-LB. DO NOT USE THREAD LOCKING COMPOUND
PEDAL ASSEMBLY IS COMPATIBLE WITH TILTON ENGINEERING 76 SERIES MASTER CYLINDERS SOLD SEPARATELY
76 SERIES MASTER CYLINDERS SHOWN HERE ARE FOR ILLUSTRATION PURPOSES ONLY AND ARE NOT INCLUDED WITH PEDAL ASSEMBLY
NOTES:
1. DIMENSIONS AND TOLERANCES PER ASME Y14.5 - 2009
2. UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS ARE IN INCHES.
3. DIMENSIONS ARE NOMINAL.
BRACKET CONNECTION REQUIRES 2 BOLTS. USE WIDEST SPREAD AVAILABLE PER PAD ANGLE DESIRED. 1.0" BOLT SPREAD MINIMUM
THREE SETS OF TAPPED HOLES IN PAD MOUNT ALLOW FOR SIDE-TO-SIDE ADJUSTMENT OF .30" EACH WAY
TILTON ENGINEERING, INC.
(805) 688-2353 FAX (805) 688-2745
25 EASY STREET P.O. BOX 1787 BUELLTON, CALIFORNIA 93427 USA
TITLE: PEDAL ASSEMBLY, UNDERFOOT, 2 PEDAL, THROTTLE AND BRAKE, INSTALLATION DRAWING
DRN BY LUND CHK'D WAHL SCALE 1 : 2 DWG 6057 REV NC
P/N 72-618 DATE 6/25/2014 SHEET 1 OF 1
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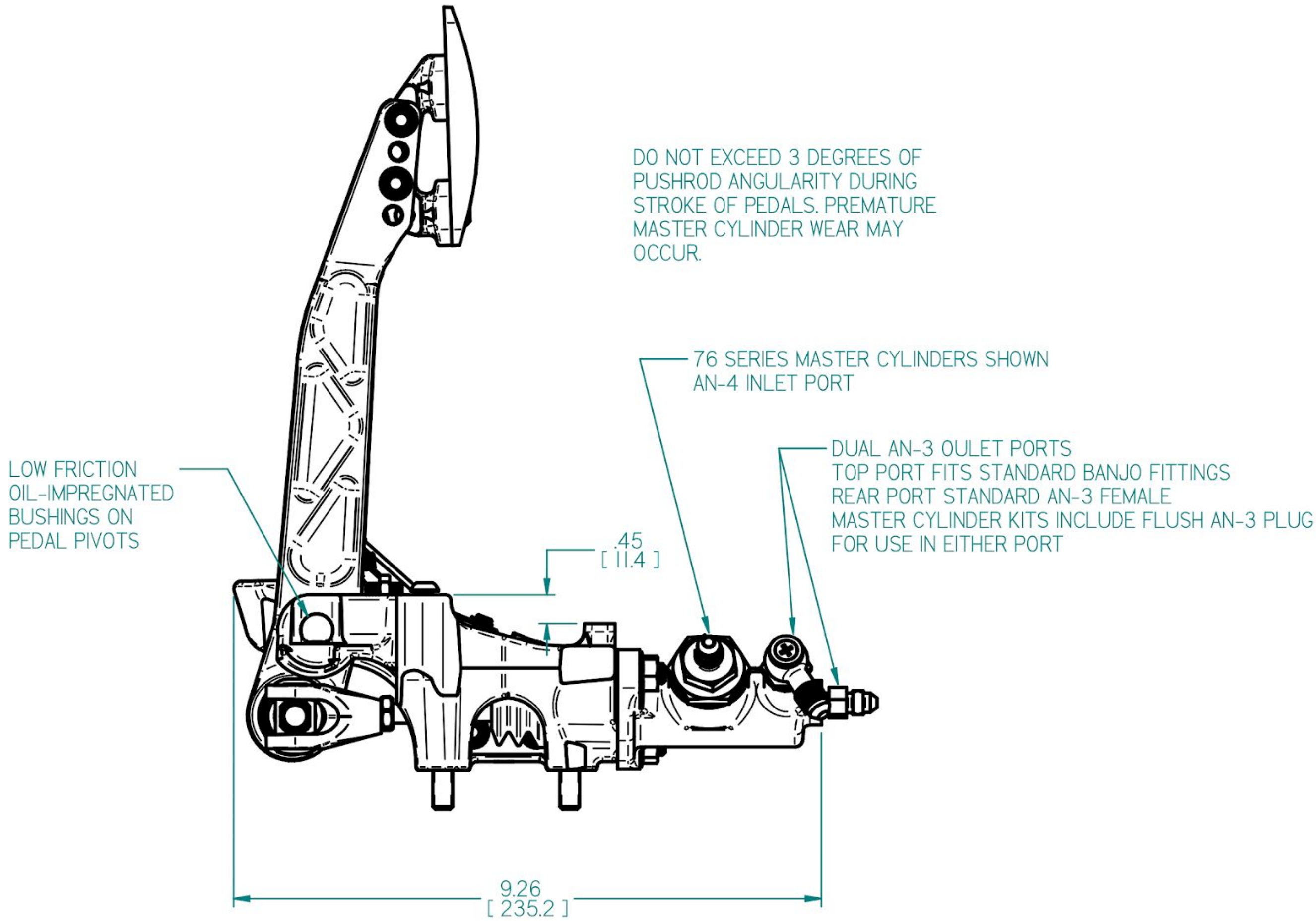
DO NOT EXCEED 3 DEGREES OF
PUSHROD ANGULARITY DURING
STROKE OF PEDALS. PREMATURE
MASTER CYLINDER WEAR MAY
OCCUR.
76 SERIES MASTER CYLINDERS SHOWN
AN-4 INLET PORT
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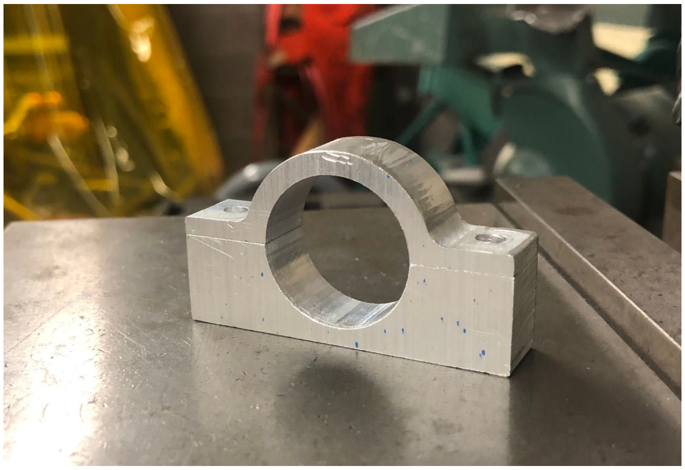
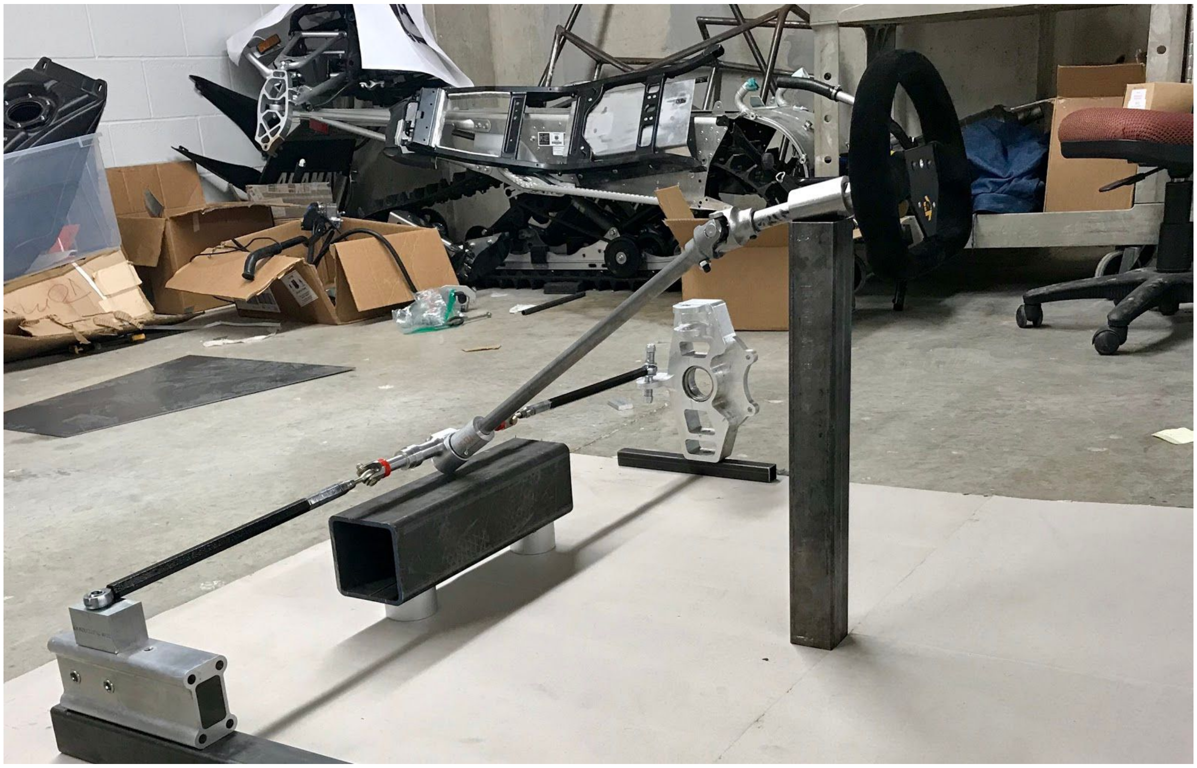
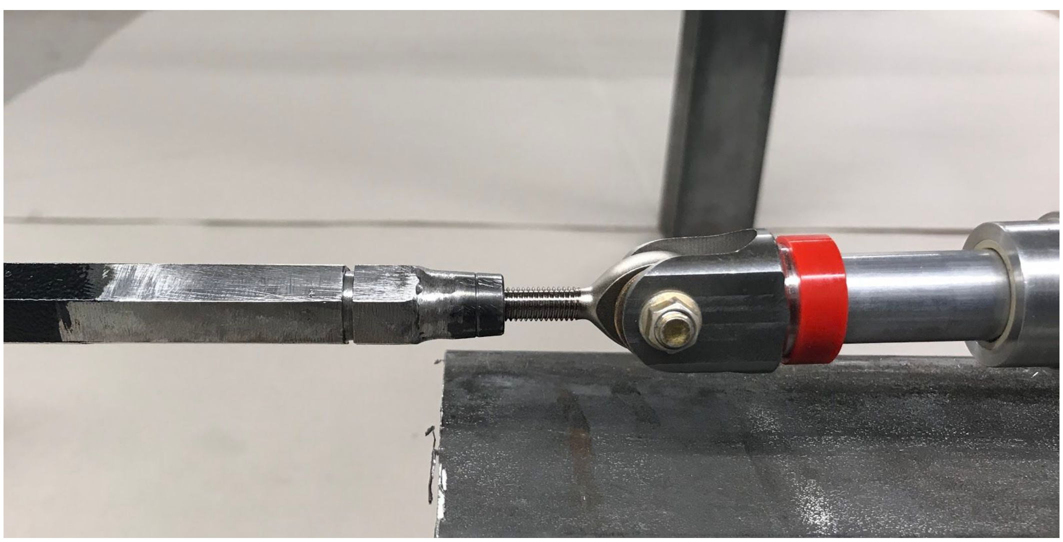
TOP PORT FITS STANDARD BANJO FITTINGS
REAR PORT STANDARD AN-3 FEMALE
MASTER CYLINDER KITS INCLUDE FLUSH AN-3 PLUG
FOR USE IN EITHER PORT
LOW FRICTION
OIL-IMPREGNATED
BUSHINGS ON
PEDAL PIVOTS
.45
[11.4]
9.26
[235.2]

Appendix C: Assembly drawings and Images






points of control
Tube
OD: 1in
t=1/4in



Appendix D: Bill of Materials

Description	Unit cost [$]	Quantity	Extended cost [$]
1/4"-28" thread bungs LH	12.99	1	12.99
3/8"-24" thread bungs RH	16.99	1	16.99
0.750" OD HREW tubing	12.99	2	25.98
3/8"-24" threaded tubes	30.99	2	61.98
3/8"-24" chrome rod ends	10.99	2	21.98
3/8"-24" straight style rod ends	12.99	2	25.98
3" ID bore spherics	8.99	1	8.99

Pedal assembly balance bar	55.25	1	55.25
Remore bias adjusters	165.75	1	165.75
Brake balance bar coupler fasteners	89.25	1	89.25
3-chamber plastic reservoire	135.15	1	135.15
600-series underfoot pedal assembly	531.25	1	531.25
Floor mount throttle linkage system	123.25	1	123.25
76-series master cylinders	106.25	2	121.50
EXP 600 racing brake fluid	18.75	1	18.75
RCV FSAE 10" brake rotor kit	330.00	2	660.00
1"-1.5" round steel bar	9.57	1	9.57
1.5'-0.75" round steel bar	9.00	1	9.00
metal cutting fee	10.00	1	10.00
Two aluminum blocks	30.00	1	30.00
Tire seating service	45.00	1	45.00
1'-3/4" aluminum round bar	6.50	1	6.50
Spiral 'O' tool	101.40	1	101.40
Laser measure	39.97	1	39.97
Hazardous material charge	1.50	1	1.50
TIG rod	24.14	1	24.14
TIG gloves	16.32	1	16.32
Drivers cowhide gloves	11.63	1	11.63
High back seat	38.47	1	38.47
Xsmall creafoam bead seat kit	189.00	1	189.00
3 ton steel jack stands	23.99	2	47.98

Cylindrical socket head	7.42	3	22.26
1/2" jam nut	5.25	3	15.75
1/4 jam nut	0.97	1	0.97
MOMO flat-bottom steering wheel	229.99	1	229.99
Quick release stearing hub	249.99	1	249.99
Steering collumn	24.99	1	24.99
Heavy duty U-joint	73.99	1	73.99
Alloy steel rod end	30.99	2	61.98
Airframe bolt	0.39	4	61.98
All metal locknut	0.64	4	2.56
Rod end retaining washer	1.79	4	7.16
Wire harness	82.95	1	82.95
23" Axle shaft	260.00	2	520.00
Drag racing slicks	197.99	4	791.96
Drag racing wheels	69.99	4	279.96
Easy protable P-ptouch	24.99	1	24.99
Genuine P-touch	24.95	1	24.95
P-touch hard case	11.99	1	11.99
Magnetic tap	24.97	2	49.94
Total			5193.88

Appendix E: Dynamic Parameters and Background

Stability and control of the vehicle depend on the setup geometries and the dynamic parameters defined by the steering axis and steering angle. The steering axis is the axis line from the upper and lower outboard A-arm pivots and the rotational axis of the wheel assembly.

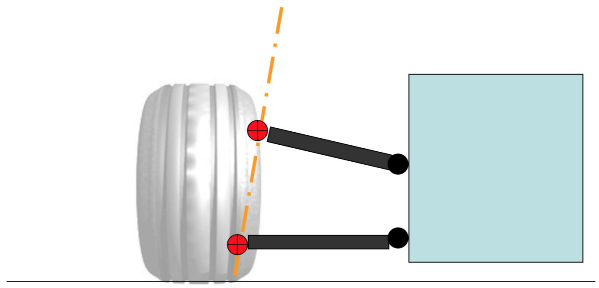


Figure 25. Steering Axis front view

The steering angle or camber angle is the angle of inclination (Θ) measured from the tire axis to the steering axis. Camber influences the ability of the tire to generate lateral forces. Increasing the angle will enlarge the slip angles. In formula suspension setups, the tires have minor positive camber, usually less than or equal to two degrees, which in combination with wheel size, tire size, tie-rod length, and component packaging, thus providing a self-centering force on the steering system to prevent tire slip and other variables which affect traction and control.

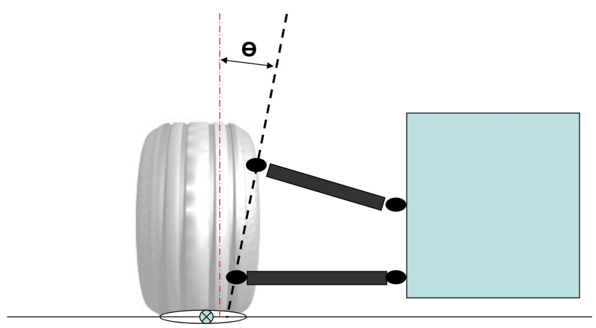


Figure 26. Steering Angle front view

Appendix F: Prototype Test Data

Table T1a. Test results for steering wheel degrees of play

Test No.	Degrees of play (deg)
1	5.0
2	5.3
3	4.7
4	4.5
5	7.9
6	3.4
7	3.2
8	4.9
9	5.4
10	5.3

Table T1b. Summary of statistics for steering wheel degrees of play

Mean	5.0 deg
Standard deviation	1.3 deg
95% Confidence Interval	4.2-5.8 deg
Minimum	3.2
Maximum	7.9

Table T2a. Complete steering system range of motion

Test Position (n) [-]	Steering Wheel Angular Position (θ) [deg]	Inner toe angle (ϕ_{in}) [deg]	Outer toe angle (ϕ_{out}) [deg]
-7	-105	33.3	34.8
-6	-90	30.7	33.2
-5	-75	24.3	30.1
-4	-60	20.5	25.6
-3	-45	17.5	18.5
-2	-30	14.5	16.7
-1	-15	8.8	7.3
0	0	0	0
1	15	4.2	6.1
2	30	10.1	11.5
3	45	18.2	22.3
4	60	25.6	27.8
5	75	27.0	30.7
6	90	28.2	34.8

Table T2b. Critical steering test positions compared with requirements for FSAE circuit

Test Position (n) [-]	Steering Wheel Angular Position (θ) [deg]	Inner toe angle (ϕ_{in}) [deg]	Inner toe angle required for FSAE Circuit [deg]	%Diff our system vs required	Outer toe angle (ϕ_{out}) [deg]	Outer toe angle required for FSAE Circuit [deg]	%Diff our system vs required
-7	-105	33.3	23.4	+42.3%	34.8	33.26	+4.63%
6	90	28.2	23.4	+20.5%	34.8	33.26	+4.63%

Table T4a. The table shows 30 instances recorded divided evenly by three group members

Test No.	Subject 1 (seconds)	Subject 2 (seconds)	Subject 3 (seconds)
1	5.63	4.60	2.59
2	4.02	3.98	2.83
3	3.86	5.14	2.66
4	3.65	3.62	2.35
5	3.49	3.38	5.52
6	3.16	3.75	2.23
7	2.92	5.26	2.33
8	3.98	2.96	2.25
9	3.23	2.66	2.20
10	2.92	3.61	2.02

Table T4b. Summary of results

	Average (s)	Minimum Time (s)	Maximum Time (s)
Subject 1	3.69	2.92	5.63
Subject 2	3.90	2.66	5.26
Subject 3	2.70	2.02	5.52
Net	3.43	2.02	5.63

Bibliography (IEEE)

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[2] A. Anderson, G. Bentz, J. Marques, L. Wilson, J. Hudson, and D. Grusak, "Trinity University Motorsports Preliminary Design Report."