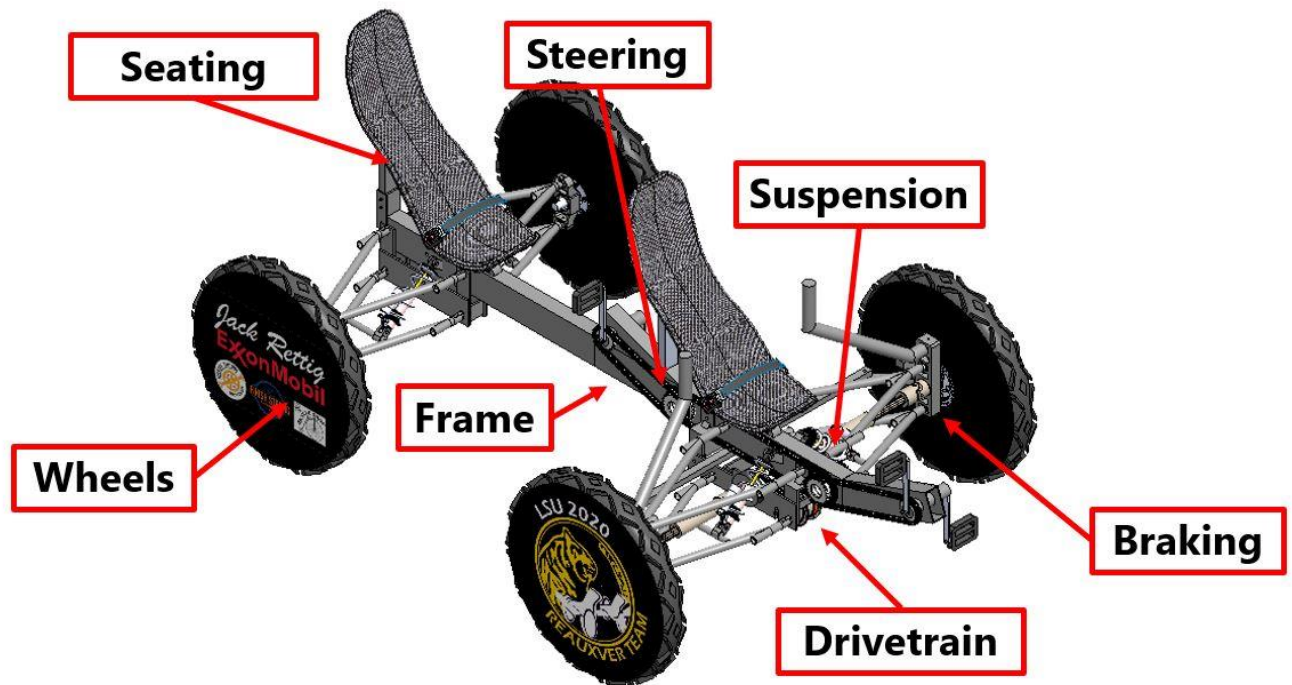


Final Design Proposal Report

Team #24 – NASA Human Exploration Rover Challenge

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I. Abstract

Team 24 of the LSU Engineering Capstone Design class has been tasked with designing, manufacturing, and testing a human-powered rover capable of completing a half-mile course with rigorous extraterrestrial terrain in under eight minutes. The key functions of the rover design include the following: traversing a demanding obstacle course; supporting the load of two riders plus vehicle weight; storing collected samples/task items; converting rider input to useful output; maintaining durability, rigidity, and reliability throughout the competition; and easy and quick assembly. Safety, simplicity, maneuverability, human-powered, and comfortable/ergonomic were the qualitative constraints identified. Additionally, twelve measurable engineering specifications were identified including: a collapsed volume of no greater than 5 ft x 5 ft x 5 ft; a width of no greater than 5 ft; a weight that is less than or equal to 130 lbs.; and a rover assembly time of less than two minutes. The 2020 rover will include the following concepts: a front tandem drive; four solid disk sandwich structure composite wheels; simple single beam frame; double wishbone suspension; push-pull steering; hydraulic disc brakes; and recumbent seating. Significant deliverables of the project include the following: providing a bill of materials by November 15, 2019; completing manufacturing by March 16, 2020; and competing in the NASA Rover Competition April 17-18, 2020.

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II. Executive Summary

Objectives and Problem Definition:

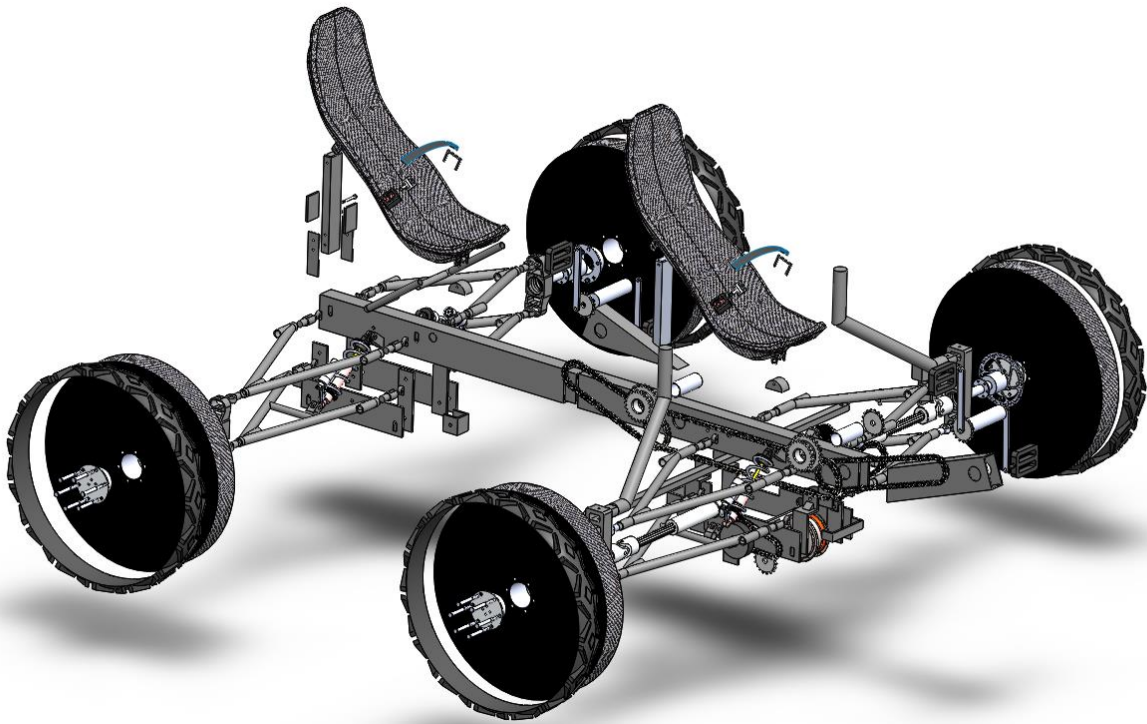
Team 24 has been tasked with engineering a human-powered rover to win the 2020 NASA Human Exploration Rover Challenge by improving the 2019 rover's drivetrain, designing and manufacturing four new wheels, and reducing overall weight. The NASA Human Exploration Rover Challenge is an annual competition held at the U.S. Space and Rocket Center in Huntsville, Alabama. The NASA Rover Challenge will be held on April 17-18, 2020. The competition challenges high school and college teams to design and build a human-powered rover. The rover must be capable of traversing simulated extraterrestrial terrain including craters, boulders, ridges, inclines, crevasses and ruts. The extraterrestrial terrain is split into various obstacles and tasks that, when completed, allows the team to accrue points. The course is 0.5 miles long and consists of 5 tasks and 14 obstacles. The teams must complete as many obstacles and tasks as possible in at most eight minutes to receive maximum points. Each team will have two chances to navigate the course, and their best score of these two attempts will be used for the rankings. To be eligible to compete, all teams must pass a Mission Readiness Review (MRR) and an Excursion Readiness Review (ERR) [1].

Key Engineering Specifications:

The key engineering specifications are the following: rover width of ≤ 5 ft; rover clearance of ≥ 1 ft from the ground to the lowest part of rover with human contact; rover average speed of 4 mph to complete the half mile course in under eight minutes; rover turning radius of ≤ 15 ft. Failure to meet these would result in immediate disqualification from the competition. Other significant engineering specifications include the following: rover collapsed volume of ≤ 5 ft x 5 ft x 5 ft; rover weight: ≤ 130 lbs. (5 points); 131 - 170lbs (3 points); 170 - 210 lbs. (1 point); rover assembly time from collapsed volume completed by drivers: ≤ 2 min (2 pts); 2 - 4 min (1 pt.). These specifications are important for placing high in the competition. However, they are not constraints that would get Team 24 disqualified from the competition if they are not met.

Product/System Description:

In its most simple form, the 2020 rover design is a four-wheeled, tandem, recumbent, off-road quadricycle. See Exploded View Solid Model below for the current prototype design.



Exploded View Solid Model

Engineering Analysis Result Summary:

Engineering analysis conducted for the 2020 Rover consisted of stress analysis, performance capabilities, and material selection for the various components that compose the rover's subsystems. The broad range of analysis conducted yielded safety factors greater than that of industry standards, proving adequacy and feasibility of concepts for integration into the design. Performance analysis resulted in output rover metrics which adhered in their entirety to both team-imposed and competition-imposed quantitative constraints such as rover output velocity, rover total weight, clearance, turning radius, etc. Materials were chosen based on how their mechanical properties compared against design requirements and were either selected or altered in order to adhere to these minimum design requirements based on the expected loading conditions during competition.

Manufacturing/Assembly Summary:

Manufacturing the 2020 rover will consist of both manufacturing of individual parts and assembly of collections of parts to form the seven subsystems that compose the rover. The manufacturing techniques that will be used in realization of the prototype include CNC milling, lathing, sanding, welding and heat treating, water-jetting, grinding, 3D printing, and vacuum-bagging. The actual techniques required to manufacture each subsystem vary amongst those stated above and will be addressed in the following text during subsystem breakdown of the manufacturing processes.

Statement of Expected Performance:

Team 24 has very high expectations for the 2020 Rover's performance in the competition. Every design decision made during the conceptual design, concept selection, and engineering analysis sections of the Fall semester were evaluated based on how these decisions improve the rover's overall performance in 2020. Team 24 has a relentless goal for first place and hopes to set a maximum standard for the future LSU NASA Rover Challenge capstone teams.

Safety Considerations:

All appropriate safety precautions will be taken by the team while performing any activity related directly or indirectly to the LSU 2020 NASA Rover Challenge project. All safety guidelines imposed by NASA will be strictly adhered to, which includes wearing long-sleeves (shirt/pants), helmets, gloves, and safety glasses during competition. All NASA-imposed safety requirements of the physical design, which includes, but is not limited to, mitigating protrusions and including safety restraints (seat belts), will be designed for. Similarly, all safety requirements of the LSU Advanced Manufacturing Facility will be adhered to when manufacturing the rover within the facility, which includes, but is not limited to, wearing long pants and safety glasses. Specific safety considerations can be found in Section VII. – Safety Considerations.

Design-Phase Testing Results:

Many significant design decisions made by Team 24 were based on testing of previous rover models during the Fall design phase. Some key takeaways from design-phase testing include drivetrain optimization and a large budget allocation for the drivetrain. Testing of LSU's 2019-model rover showed that better concepts could be implemented involving the 2020 rover's drivetrain, including internally geared hub integration optimization, tandem drivetrain integration due to overall weakness in individually-powered axles, and chain tensioner integration in order to prevent chain slippage/derailment.

Design-phase testing was also conducted heavily on the wheels. Again, decisions regarding wheel design for the 2020 rover were heavily influenced by the results of testing on previous rover models. Some of the main takeaways were the 2019 LSU rover's wheel failure during impact loading and the shifting of the wheel hub relative to the wheel center. Proper design of the 2020 rover model's wheels will inhibit these problems from occurring in the future.

Validation Test Plan Summary:

Team 24 hopes to complete manufacturing by mid-March in order to allow for a month's worth of testing prior to competition. The validation test plan will consist of creating a mock rover course based on some of the most demanding obstacles/tasks that the team will have to face during competition. The team plans to execute the course many times over the testing month in order to strengthen rider dynamic, improve rider endurance and strength, and modify any components that either fail or illustrate potential for future failure at competition.

III. Engineering Specification

III.A. Objective Statement

To engineer a human-powered rover to win the 2020 NASA Rover Challenge by: improving the drivetrain, manufacturing four new wheels, and reducing overall weight.

III.B. Introduction

III.B.1. Background Information

The 2020 NASA Human Exploration Rover Challenge will be held at the United States Space and Rocket Center in Huntsville, Alabama from April 16th to April 18th. In this annual competition, high school and college students are required to design and manufacture a human-powered vehicle capable of navigating an obstacle course that simulates extraterrestrial terrain. The highest an LSU team has placed in this competition is 6th in 2015. From 2016-2019, LSU placed 20th, 15th, 9th (unofficially)¹, and 16th respectively. The 2020 NASA Human Exploration Rover Challenge consists of several possible awards. Team 24 has set a goal to win 1st place.

III.B.2. Problem Description and Motivation for the Project

The competition requires the half-mile course to be completed in under eight minutes. The obstacle course consists of 5 optional tasks and 14 obstacles, 11 of which can be bypassed. Although a bypassed obstacle earns no points, many teams strategically skip difficult obstacles to save time. Before being eligible for placement awards, the rover must pass a Mission Readiness Review (MRR). The MRR will verify that the rover abides by the competitions size and weight constraints [1]. Competing rovers will also need to pass an Excursion Readiness Review (ERR). The judges conducting the ERR verify that the rover has a United States or Louisiana State University flag, contains all the tools necessary to complete each task, and that the vehicle adheres to the safety rules specified in the guidebook. The safety requirements include seat restraints, commercially manufactured protective headgear, and padding on all protrusions sticking out of the rover [1]. Once through the MRR and ERR, the teams will be ready to compete. Each team will have two opportunities to complete the course, once on Friday and again on Saturday. The best score from these two runs will be used for placement awards.

III.B.3. Existing/Competing Technologies

Recognizing the success and failure of existing designs is an important step in making future design decisions. The major components of previous rovers included wheels, drivetrain, chassis, suspension, steering, and braking.

¹ The 2018 LSU Team did not complete the course within the time limit, so no official points were awarded to them. The 9th place position was determined from the unofficial points earned.

The design and fabrication of the rover's wheels are a crucial aspect of the competition. According to the guidebook, teams must redesign at least 50% of a past team's design [1]. The wheels are worth 40%. Therefore, a primary focus of Team 24 will be to create a new wheel design. Most wheels from previous rovers came in two forms: spoked wheels and disk wheels. The materials used for spoked wheels included metal, polymer, and wood. The 2019 LSU Rover team used a hollow disk wheel design made of carbon fiber. To reduce weight, they opted for a 3-wheel design. Unfortunately, this 3-wheel design also reduced the rover's overall stability, forcing the team to skip obstacles that were deemed too dangerous to traverse. In previous years, the winning teams all had 4-wheel designs, which allowed for better stability and traction. The design of the rover's drivetrain is extremely important because the course will require that the riders transfer as much power as possible through the wheels. Almost all rover designs in past competitions used a pedal driven gear system. The higher performing rovers employed multiple gears to traverse inclines and difficult terrain. Many designs, including the 2019 LSU rover, also used bike chains, which have proven to be susceptible to breaking and slipping during competition. The 2020 NASA guidebook specifically suggests not to use bike chains due to their poor performance. There are a wide range of chassis used in competition. These range from simple one-bar chassis to complex truss designs. Last year's sturdy truss design added stability, but also added weight. Efficient suspension is required when trying to traverse rough terrain. Some teams opted to go without suspension, however they weren't very successful due to the bumpy nature of the course. Suspension can be implemented simply through shocks, as was done in last year's design. The 2019 team's most successful components were their steering and braking system. The steering employed a relatively simple u-joint that provided for very efficient control of the rover. The hydraulic disc braking system mimics that of a motorcycle.

III.B.4. Potential Customers

Primary:

As a competition-based project, the primary customers are the NASA Rover Challenge judges. The rover's sole purpose is to perform well in a competition that will be scored by NASA judges. NASA judges define the vehicle's constraints and provide the course layout, objectives, tasks, and competition rules. Team 24 will aim to score the most points while satisfying all competition requirements.

Secondary:

Secondary customers include the LSU Capstone Design Review Panel and the project sponsors, Jack Rettig and Dr. Dimitris Nikolopoulos. They expect the 2020 LSU Rover Team to place first and to justify all design choices made. The riders for Team 24 will also be secondary customers. Since they will be traversing the obstacle course using the rover, their personal requirements for safety, rover efficiency, and comfort are to be accounted for.

III.C. Functional Requirements

Table III-1: Required Functions

#	Function	Explanation
F1	Traverse a demanding obstacle course	The NASA Mars Rover competition consists of 14 challenging obstacles and 5 tasks that the rover has the option to complete in order to score well & place.
F2	Support the load of two riders plus vehicle weight	The rover must be able to support its own weight as well as the loads induced by the two riders.
F3	Store collected samples/task items	The rover is required to have designated storage areas for collected samples and task items acquired by the team while traversing the course.
F4	Convert rider input (pedaling) to useful output	The rover must be man-powered and efficient in terms of converting human energy into useful output work.
F5	Maintain durability throughout the competition	The rover must maintain its structural integrity while completing the competition. The team will not be awarded points if the rover cannot complete the course.
F6	Easy & quick assembly	The team is scored based on how quickly the rover is assembled into the 'ready' position. The rover must adhere to a volume constraint in the disassembled position prior to competition start.

III.D. Qualitative Constraints

Table III-2: Required Qualitative Constraints

#	Qualitative Constraint	Explanation
Q1	Safe (Competition)	Safety is the number one priority of the team. The competition requires seat restraints, no protrusions, and proper riding attire.
Q2	Human-Powered (Competition)	The competition rules & regulations require the rover to be human-powered. No energy storage devices may be used.
Q3	Maneuverable (Team)	The rover must be easily maneuvered while completing different challenges in the competition.
Q4	Simple (Team)	Maintaining a simple design reduces risk of analytical error because analytical methods on simple designs are generally less strenuous, eases technical communication.
Q5	Comfortable/Ergonomic (Team)	The rover should be comfortable for the riders and should possess physical appeal. The riders will not be able to perform adequately if the rover is not comfortable to operate.

III.E. Measurable Engineering Specifications

Table III-3: Required Measurable Engineering Specifications

#	Name	Symbol	Units	Value(s)	Explanation
M1	Rover Width (Competition)	W_R	ft	$W_R \leq 5$	The rover cannot exceed a width of 5 ft
M2	Rover Clearance (Competition)	C_R	ft	$C_R \geq 1$	Clearance between the ground and the lowest part of the rover with rider contact
M3	Rover Average Speed (Competition)	V_R	mph	$V_R \geq 4$	The rover has 8:00 minutes to complete the 0.5-mile course
M4	Rover Turning Radius (Competition)	R_T	ft	$R_T \leq 15$	Turning radius compliance required to compete
M5	Rover Collapsed Volume (Competition)	V_R	ft ³	$V_R \leq 125$ (5x5x5) (4 pts)	Competition-imposed volume constraint
M6	Rover Assembly Time (Competition)	t_A	min	$t_A \leq 2$ (2 pts) $2 < t_A \leq 4$ (1 pt) $t_A > 4$ (0 pts)	Riders will assemble the rover while being timed. Points given accordingly
M7	Rover Weight (Competition)	W_R	lbs	$W_R \leq 130$ (5 pts) $130 < W_R \leq 170$ (3 pts) $170 < W_R \leq 210$ (1 pt) $W_R > 210$ (0 pts)	The rover will be weighed. Points given accordingly
M8	Budget (Sponsor)	B	\$	8,000	Must stay within allotted budget
M9	Wheel Qty. (Team)	W_Q	#	3-4	For stability and simplification
M10	Steering (Team)	D.O.F.	#	1	Simplify steering system
M11	Drive Input (Team)	D.O.F.	#	1	Simplify the driving system

III.F. Deliverables

Fall 2019 Capstone Design Imposed Deliverables	
Date	Description
09/23/19	Engineering Specification Interim Report
10/02/19	Midterm Concept Group Presentation
10/14/19	Embodiment Interim Report
10/28/19	Engineering Analysis Technical Briefing
12/01/19	Final Project Presentation Slides & Poster
12/02/19 - 12/06/2019	Final Presentation
12/11/19	Final Design Proposal Report

Spring 2020 Competition Imposed Deliverables	
Date	Description
01/16/20	Final Date for Team Registration
02/14/20	Student Data Feedback Media Release Form Vehicle Reuse Report
03/13/20	Team Photo
04/03/20	Completed Rover Photo
04/16/20	Team Check-In Course Walk Safety Meeting
04/17/20	HERC Competition Day 1
04/18/20	HERC Competition Day 2

By the end of the Fall 2019 semester, Team 24 will have signed up for the 2020 Human Exploration Rover Competition and completed all design and analysis deliverables required for Capstone Design. Along with the design deliverables, a completed bill of materials and engineering drawings will have been produced to ensure that parts can be ordered as quickly as possible. Required Capstone Design deliverables are listed in Table III.F.1: Required Capstone Design Deliverables.

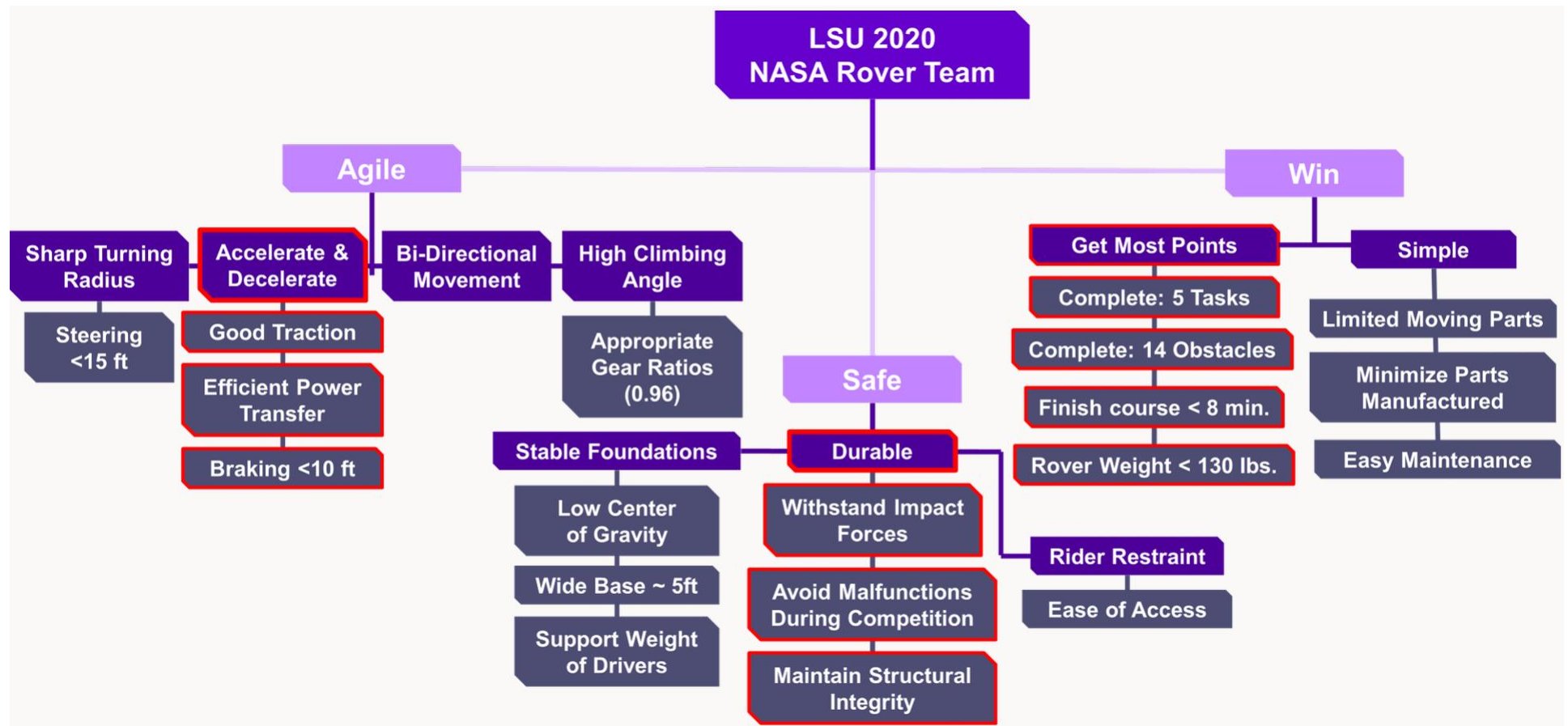
By early March 2020, Team 24 will have designed and manufactured a human powered rover which will be used in the Human Exploration Rover Challenge. Having a fully functional rover completed by March 2020 will enable Team 24 to have adequate time for testing before the April competition. All other Capstone Design deliverables are listed in the above table: Spring 2020 Competition Imposed Deliverables.

IV. Embodiment

IV.A. Functional Breakdown - Objective Tree(s)

The functional requirements that were key design components were: the rover's ability to accelerate and decelerate, having a sharp turning radius, supporting the weight of the riders, withstanding impact forces, avoiding malfunctions during competition, and minimize parts to manufacture. The project was broken in seven rover subsystems: drivetrain, wheels, frame, suspension, braking, seats, and steering. The subsystems will work together to execute the functions required. The wheels, drivetrain, and steering are responsible for acceleration while braking is responsible for deceleration. Steering is responsible for having a sharp turning radius. The drivetrain is responsible for having appropriate gear ratios dependent on rider torque input. The frame, wheels, and suspension are responsible for withstanding impact forces and supporting the weight of the riders. All subsystems will be responsible for avoiding malfunctions during the competition and minimizing parts to manufacture; however, drivetrain has more responsibility due to it being the root cause of malfunction in the 2019 rover. The main goal of the team is to get first place, which requires the rover to complete all obstacles and complete all tasks. A complete objective tree can be found in Figure IV.A.1- LSU 2020 NASA Rover Objective Tree.

Figure IV.A.1: LSU 2020 NASA Rover Objective Tree



IV.B. Concept/Solution Generation, Evaluation and Selection

IV.B.1. Concept/Solution Generation Method(s) Used

To begin the concept generation phase of the project, the team studied previous rover designs (previous LSU, high school, and college teams) to articulate conceptual ideas. The primary form of concept generation used by the team was brainstorming. The team held a brainstorming session, where each member came up with as many ideas as possible for the rover design. After a complete list was formed during the brainstorming session, team members were assigned to sub-system teams (drivetrain, wheels, frame, suspension, braking, steering). Each subsystem team utilized the list created during the brainstorming session to begin refining the concepts to their respective subsystem. Once a complete list was formulated for each subsystem, the team began concept evaluation.

IV.B.2. Concept/Solution Evaluation Method(s) Used

The primary concept/solution evaluation method used by the team was decision matrices with weighted metrics. The team used traditional decision matrix evaluation methods, which required multiplying a team-defined rating quantity by the weight of a metric for each different metric and concept. After completing this for all concepts for the subsystem, the sum of each rating per concept was calculated, and the highest rating corresponded to the best performing concept. The team remained objective when defining metric weights and ratings per concept. Otherwise, the results of the decision matrices would be of little value.

A secondary concept/solution evaluation method used was done by comparing the concepts to higher-performing teams of the previous NASA rover competitions. Many of the team's primary concepts, such as the front-wheel drivetrain concept or the I-beam wheel concept, were evaluated based on how other teams performed with these concepts as part of their final competitive design. Their performances were taken into consideration when determining the ratings of the respective concepts.

One tertiary concept evaluation method used by the team was reflecting on previous team's feedback regarding certain concepts. The team consulted with multiple members of previous LSU Rover teams thus far. One concept that was directly influenced by the feedback of previous rover teams was the 3- or 4-wheel concept. It was highly suggested that this year's rover team go with a 4-wheel design due to factors such as rover stability and design symmetry.

Lastly, the team used results from testing the previous LSU rover in order to evaluate concepts. Some concepts considered by the team prior to testing were a four-bar steering mechanism controlled by lever-action (utilized in zero-turn lawn mowers) and the implementation of an internally-gear hub. Both concepts were evaluated during the 2019 LSU rover testing stage because the 2019 LSU rover consisted of these components.

IV.B.3. Concept/Solution for Function F1 - Traverse a Demanding Obstacle Course

IV.B.3.a. Concept/Solution Selected for Function F1

The concepts/solutions selected to traverse a demanding obstacle course were a four-wheel design with curved rubber treads and the double wishbone suspension.

Four solid disk wheels were selected based off on the decision matrix seen in XI.B.1.a. Stability and traction were the highest-weighted metrics due to issue encountered by the 2019 rover as well being able to accelerate and decelerate on different terrains.

Curved rubber treads attached onto solid disk wheels was selected due to elasticity provided to absorb impact forces while also allowing the rover to accelerate faster than a flat rubber tread. Team 24 consulted with a professional machinist to discuss the advantages of the curved rubber tread and was informed that by reducing contact area between the wheel and surface, the rover would be much easier to move and accelerate than that of a flat tread. To manufacture a curve surface, it was recommended to utilize additive manufacturing to create a negative mold that can be attached onto the wheel and allow the liquid rubber to adhere directly onto the surface of the wheel.

The Push-Pull Steering system was selected because it facilitated steering with a convenient rider input that fit well with the rover frame. The tie rod linkage allowed the rover to easily meet NASA's turning radius requirement 15 ft or less based off the rover's wheelbase and track length.

IV.B.3.b. Concepts/Solutions Considered for Function F1

The 2019 rover used a three-wheel designed and after consulting with their team members, one recommendation was to use a four-wheel design because of the instability of a three-wheel rover. Team 24 looked into reusing two of the 2019 wheels and manufacture two new ones, but the dimensions required for the 2020 rover changed the design and therefore would not work.

Initially, a spring tread was considered; however, manufacturability of a spring tread is time-consuming and would conflict with the team-imposed constraint of having the rover manufacturing complete a month before the competition. A flat tread was utilized by the 2019 rover and performed well to create traction between the wheel and different surfaces, however during impact, the tread offered no elasticity to mitigate initial impact forces from the composite wheel.

A standard rack and pinion design was also considered, however, the team ultimately decided against it because of the difficulty presented by the prospect of incorporating a steering column into the rover's design. Placing a straight steering column down the middle of the rover would conflict with the rover's single beam frame. This is where the Push-Pull design succeeded in its ability to fit the rover's frame. A full decision matrix can be found in I.B.1.a.

IV.B.4. Concept/Solution for Function F2 – Support the Load of Two Riders Plus Vehicle Weight

IV.B.4.a. Concept/Solution Selected for Function F2

The concepts/solutions selected to support the load of two riders plus the vehicle weight were the single beam frame, the double wishbone suspension and the solid disk wheels.

The single beam frame was selected based on the results of the decision matrix seen in XI.B.2.a. Manufacturability and weight were the highest-weighted metrics due to the team-imposed constraint of having the rover manufacturing complete by March 16, 2020 and reducing the overall weight from last year's rover design. Additionally, Team 24 consulted with past LSU rover teams and one of the main recommendations was to simplify the frame design to allow for more time of the testing and improvement stage.

The decision to select the double wishbone suspension came down to three main factors, as seen in the decision matrix in XI.B.2.a. Stability was one of the most important deciding factors. The double wishbone suspension's stability not only allows for the rover to be easily maneuvered, but also reduces the amount of stress seen on other subsystems in the design.

Solid disks wheels were selected based on the results of the decision matrix seen in XI.B.2.a. Compressive strength and manufacturability were the highest-weighted metrics due to the wheels yielding during impact of the 2019 rover as well as the team-imposed constraint of having the rover manufacturing complete by March 16, 2020. Additionally, the technology and experience to manufacture the wheels were secured by the 2019 rover team and can be reproduced. The wheels from the 2019 rover were one of the features that Team 24 found to be good but adjusted accordingly using a different core material.

IV.B.4.b. Concepts/Solutions Considered for Function F2

There were three concepts considered for the Frame to support the weight of two riders plus the vehicle weight. The concepts considered were the 4-Bar Trapezoidal, 3-Bar Triangular, and Single Beam Frame as seen in XI.B.2.a. Team 24 considered these three frame designs based on past rovers from LSU and successful rovers from past competitions.

Three concepts were considered for the Suspension to support the weight of the two riders plus the vehicle. These concepts were the MacPherson Strut, Leaf Spring, and Double Wishbone as seen in XI.B.2.a. Team 24 considered these three suspension designs based on rovers seen in past competitions.

There were four concepts considered for the Wheels to support the weight of two riders plus the vehicle weight. The concepts considered were the Spring, Spokes, I-Beam cross-section, and the Solid Disk Wheels as seen in XI.B.2.a. Team 24 considered these four-wheel designs based on past rovers from LSU and successful rovers from past competitions. The wheels have to be made by the team; they cannot be bought and used off-the-shelf.

IV.B.5. Concept/Solution for Function F3 – Store Collected Samples/Task Items

IV.B.5.a. Concept/Solution Selected for Function F3

The concepts/solutions selected to store and collect sample and task items were the single beam frame, 3D-printed liquid/soil retrieval scoop, and PVC cross-beam core sample retrieval tool.

The single beam frame was selected based on the results of the decision matrix seen in XI.B.3.a. Manufacturability was the highest weighted metrics to ensure that the rover would be completed before the team-imposed constraint date of March 16, 2020. Manufacturability also included simplicity of the frame to ensure easy storage for the tasks.

The 3D Printed Scoop Design was selected for the Solid/Liquid Sample Retrieval task as seen in decision matrix XI.B.3.a. Collection effectiveness, no cross-contamination, and weight were the highest weighted metrics because these were the most crucial features to ensure successful completion of this task. The 3D printed scoop design was selected because of its high effectiveness in collecting and storing the required amount of liquid and soil samples while eliminating cross contamination and being lightweight.

The Cross-Beam retrieval tool was selected for the core sample retrieval task as seen in decision matrix XI.B.3.a. Weight, Fast Extraction, and Fast Containment were the highest weighted metrics due to the 8-minute total time constraint imposed by the competition rules. The cross-beam design was the most effective in all three criteria, which was why it was selected.

IV.B.5.b. Concepts/Solutions Considered for Function F3

Other concepts considered can be found in all three decision matrices shown in XI.B.3.a. For frame, other concepts considered included the 4-Bar Trapezoidal and 3-Bar Triangular Beam. For the Liquid/Solid retrieval task, circular, cap shovel, and vacuum pump design were also considered. For the Core Sample retrieval task, Plunger, Digger and Scoop designs were considered.

IV.B.6. Concept/Solution for Function F4 – Convert rider input (pedaling) to useful output

IV.B.6.a. Concept/Solution Selected for Function F4

The concept/solution selected to convert rider input into useful output was the tandem front drivetrain. The team chose the tandem drive concept as the best concept for the drivetrain based on the results of the decision matrix seen in XI.B.4.a: Concept Selected – Details. Available output power and simplicity were the two highest-weighted metrics in the drivetrain decision matrix. These were specifically weighted in this way because the primary goal of a drivetrain is to transmit power to a system as efficiently as possible. Also, one of Team 24's most important qualitative constraints is to maintain simplicity of the entire design. Manufacturability was weighted second because the team will likely purchase most, if not all, of the parts required by the drivetrain tandem concept. This is why the manufacturability metric was ranked equally amongst the three concepts being evaluated. Cost and weight were the final two metrics which were weighted the lowest. Cost was weighted lowest because a large portion of our current budget (~18.5%) will be allocated to the drivetrain subsystem. Also, a member of Team 24 has access to many free parts that can be used for manufacturing the drivetrain. Finally, weight was the lowest weighted metric because the team will be using primarily standard bicycle parts; thus, only very small deviations from standard part weights will be possible.

IV.B.6.b. Concepts/Solutions Considered for Function F4

There were three concepts considered for the drivetrain subsystem: All-wheel drive, classical tandem drive, and front-wheel drive – see XI.B.4.a: Concept Selected – Details. The team chose these three primary concepts because of their feasibility and usage by high-

performing teams of the previous years' competition. The feasibility of each of the above subsystems was evaluated with the aid of a professional bicycle mechanic.

IV.B.7. Concept/Solution for Function F5 – Maintain durability throughout the competition

IV.B.7.a. Concept/Solution Selected for Function F5

The concepts/solutions selected to maintain durability throughout the competition were the Double Wishbone Suspension, Aluminum 6061-T6 for a majority of load bearing components and maintaining a factor of safety with respect to yield of ≥ 2.5 .

The decision to design the double wishbone suspension was based on the results of the decision matrix seen in XI.B.2.a. The double wishbone suspension allows for impact forces to be absorbed through the shocks. An outcome of this absorption is that lower stresses will be seen on other subsystems throughout the rover's design.

Team 24 selected Aluminum 6061-T6 based on the results of the decision matrix seen in XI.B.5.b. Manufacturability and weight were the highest-weighted metrics due to the team-imposed constraint of having the rover manufacturing complete by March 16, 2020 and reducing the overall weight from last year's rover design. Aluminum 6061-T6 was found to have high strength, low weight, and be more easily welded than other types of materials.

To maintain durability throughout the competition Team 24 selected a safety factor with respect to yield of ≥ 2.5 . This factor of safety was selected in accordance with Recognized and Generally Acceptable Good Engineering Practice from research. A safety factor with respect to yield of ≥ 2.5 is generally used with ordinary materials where loading and environmental conditions are not severe such as that of the 2020 Rover.

IV.B.7.b. Concepts/Solutions Considered for Function F5

Three concepts were considered for the Suspension to support the weight of the two riders plus the vehicle. These concepts were the MacPherson Strut, Leaf Spring, and Double Wishbone as seen in XI.B.2.a. Team 24 considered these three suspension designs based on rovers seen in past competitions.

Three concepts were considered for material to maintain durability throughout the competition. These materials included carbon fiber, aluminum and steel as seen in XI.B.5.b. These three materials were selected due to their availability as single beam frames, their use in past rover designs and from the analysis for material selection of the frame. Carbon fiber was found to be lightweight and strong but provided many uncertainties in manufacturing that were concerning to the team. Steel was shown to be a very versatile material but was found to be generally heavier than aluminum.

Factors of safety based on yield are often taken between 1.5 and 4.0. The goal of Team 24 was to reduce the weight of the 2020 Rover compared to last year's design while maintaining durability throughout the competition. A safety factor of 1.5 was determined to be too low to account for any uncertainty in the loading conditions and 4.0 was determined to be too high of a minimum safety factor for Team 24 to reduce the weight of the rover.

IV.B.8. Concept/Solution for Function F6 – Easy & quick assembly

IV.B.8.a. Concept/Solution Selected for Function F6

The concept/solution for an easy and quick assembly consisted of two key subassemblies: seats and the frame. The solution for the seats easy and quick assembly was to have the seats bolted on to the frame and have the rear seat post be removable. This will allow the two riders to quickly slide the bolts in and secure the components.

The solution for the frame easy and quick assembly is to have a bolted frame connection piece in the center of the front and rear frame sections. When the frame is in the storage position only the rear section of the frame will be connected to the connection piece while the rear of the rover will be slid past the front of the rover. This ensures the rover will fit within the collapsed volumetric constraint of 125 ft³. Upon assembly the exposed end of the connection piece will be slid into the front section of the frame and bolted secure.

IV.B.8.b. Concepts/Solutions Considered for Function F6

The other concept considered for the seats easy and quick assembly was to use seat stays as seen in XI.B.6.b. These seat stays did provide an easier manufacturing process for the solution, but the decision was made to not to use them because the cost for the components would have quadrupled the total cost of the seats.

IV.C. System Description/Product Architecture

The rover components were broken down into seven separate subsystems: Wheels, Frame, Drivetrain, Steering, Braking, Suspension, and Seats as illustrated in Figure IV-2: Assembled Prototype Drawing. The design is composed of four wheels, recycling two wheels from the 2019 rover and manufacturing two additional solid sandwich-style wheels for the back of the rover. A simple axle connects the back two wheels to the single bar frame.

The front portion of the frame houses the tandem drivetrain. Both riders' input torque from the pedals will go to a common chain on the right side of the rover. The power is then transferred on the left side to an internally geared hub, which will allow the riders to modify gear ratios to overcome different obstacles. A final chain attaches the output of the internally geared hub to a fixed gear on the front axle.

A push-pull steering system is connected to both wheel hubs, which allows for easy maneuverability. The two front wheels also have hydraulic brakes attached to the inner hubs to allow for quick stopping. A double wishbone suspension system is used to connect the wheel hubs to the single bar frame and provide additional stability to the overall design. Finally, two seats are attached to the single bar frame to provide a comfortable spot for the riders to sit.

Figure IV.C.1: Assembled Prototype Drawing

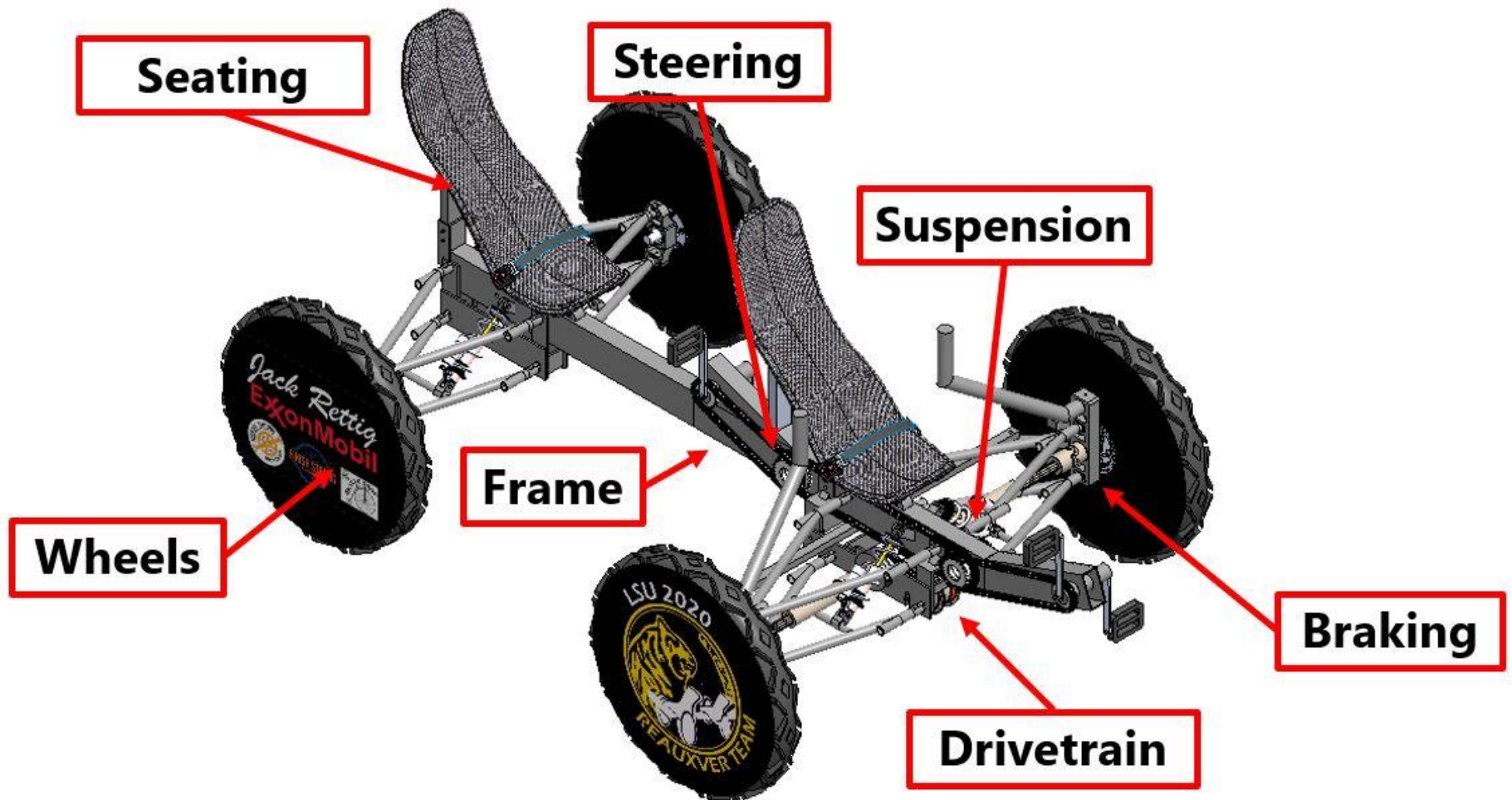


Table IV-1: List of Sub-Systems

#	Name	Brief Description of Sub-System Functionality
SS#1	Solid Disk Wheels	Allows the rover to move quickly, provides stability
SS#2	Single Bar Frame	Supports the rover and 2 riders
SS#3	Front Tandem Drivetrain	Convert rider input torque into output power
SS#4	Push-Pull Steering	Maneuvers rover
SS#5	Hydraulic Brakes	Stops the rover
SS#6	Double Wishbone Suspension	Absorbs impact from obstacles
SS#7	Seats	Provides place for riders to sit

IV.C.1. Description of Sub-System SS#1 – Solid Disk Wheels

The solid disk wheels consist of 4 sandwich structure composite wheels, round rubber treads, and aluminum 6061-T6 hubs. The wheels are responsible for converting rotational movement from the axle to lateral movement on a surface. The sandwich structure composite wheels will have a balsa wood core and a carbon fiber composite. The balsa wood core will allow the wheel to withstand the compressive stress of impact without yielding; this was a problem with the 2019 rover. The carbon fiber is responsible for the bending and torsional stress while keeping the wheels as light as possible. The round rubber tread will allow the wheel to absorb some impact forces, allow ease of acceleration and maintain traction on different surfaces such as: concrete, gravel, sand, dirt, grass, etc. The aluminum 6061 T6 hub will connect the wheels to the drive shaft and allow the wheels to rotate as the front drive shaft rotates.

IV.C.2. Description of Sub-System SS#2 – Single Bar Frame

The Single Bar Frame consists of a front frame section, a rear frame section, a frame connection piece, a front crankset support, a rear crankset support, a drivetrain bracket assembly, and a rear bracket assembly. All frame components consist of aluminum 6061-T6. The Frame is responsible for supporting the load of both riders plus the vehicle weight and being the central tie in point of all other subsystems. The front and rear frame sections allow connection of the drivetrain and suspension components while allowing the frame to be separated into two sections to meet the volumetric constraint. The frame connection piece allows the front and rear frame sections to be bolted in place when the rover is in use while still supporting the loads seen during competition. The front and rear crankset supports allow the drivetrain cranksets to be attached to the frame in a suitable position for maximum rider input. The drivetrain and rear bracketing assemblies allow tie-ins for the drivetrain and suspension components.

IV.C.3. Description of Sub-System SS#3 – Front Tandem Drivetrain

The front tandem drivetrain consists of five chains, seven 36-tooth cogs, three 18-tooth cogs, five cranksets which include square-tapered bottom brackets and 68-millimeter bottom bracket shells, two pairs of pedals, an aluminum 6061 T6 driveshaft, two steel telescopic u-joints, a NuVinci N380 Internally-Geared Hub CVP and controller, and three roller-style chain tensioners. As the drivers pedal the cranksets, the different cogs rotate and carry speed and torque ratios relative to their individual size. The chains are responsible for providing the external contact between the cogs. The chain tensions are regulated by the spring-loaded roller-style chain tensioners. The N380 internally-geared hub receives the input from the upstream drivetrain and outputs a variable speed based on the cable tension of the hub controller. This hub has a downstream cog chain system that connects directly to the aluminum driveshaft, which turns the wheels forward as the output cog rotates.

IV.C.4. Description of Sub-System SS#4 – Push-Pull Steering

The push-pull steering design consists of a tie rod linkage and steering handles. Both connect directly to the steering knuckle. The steering knuckle is the point of contact between the u-joint and wheel hub. It is also where the suspension control arms will connect to the wheel. The

tie rod linkage ensures that the necessary wheel angles are achieved during turns. It also ensures the rover will meet NASA's turning radius requirement. The rider will be able to directly influence the motion of the wheels by pushing and pulling the steering handles. A range of motion limiter will be welded onto the steering knuckle. It will contact the suspension control arm when the wheel reaches its maximum turning angle, as prescribed by the team.

IV.C.5. Description of Sub-System SS#5 – Hydraulic Brakes

From an engineering analysis and cost savings standpoint, the team decided to reuse the Shimano Deore M6000 braking system that the 2018 and 2019 team also used. The disc brake system consists of a metal rotor attached to the hub of the wheel. The location of the braking system allows for minimum wear as well as an overall lighter rim. Hydraulic brakes provide a significant advantage in overall control of braking, stopping power, and resistance to weather. The hydraulic disc brake system utilizes a lever connected to a piston by a hose containing an incompressible fluid. The braking system automatically compensated for pad wear, allowing the team to reduce maintenance.

IV.C.6. Description of Sub-System SS#6 – Double Wishbone Suspension

The double wishbone suspension will be implemented on all four of the rover's wheels. The suspension system's footprint was given significant consideration when deciding between concepts. The double wishbone is more appropriate for a rover's frame because it takes up less horizontal space between the wheels and rover frame, when compared to the other concepts evaluated. For the sake of stability, the rover will be designed with a maximum allowed width of 5 ft. The space between the frame and wheels can be utilized as space for a double wishbone suspension. A double wishbone suspension also induces a natural negative camber in the wheels which will aid in off-road maneuvering.

IV.C.7. Description of Sub-System SS#7 – Seats

The Volae Comfort Carbon Seat Shell offer a variety of benefits. The carbon fiber material used for these seats allow for each seat to weigh just under 2.5 lbs. The seats also provide a safer, more ergonomic rider position.

V. Engineering Analysis and Materials Selection

V.A.1. Engineering Analysis for SS#1 – Solid Disk Wheels

Types of Engineering Analysis Conducted:

- Stress analysis of wheel as seen in worst case scenario
- Validation of composite components using classical lamination theory
- Stress analysis of impact on the core component
- Moment of inertia calculations of different materials
- Buckling analysis

V.A.1.a. Engineering Analysis & Materials Selection for SS#1-P1- Solid Disk Wheel

Pham

Type of Engineering Analysis Conducted: Stress analysis of wheel as seen in worst case scenario

Outcomes/Conclusions: The primary outcome of the stress analysis was that the highest stress seen in the wheel was a bending stress of **63.8 ksi**. The other two stresses caused in the wheel were a torsional stress of **1.19 ksi** and a compressive stress of **6.37 psi**. These values would then be used to calculate stress seen in different plies of the carbon fiber using classical lamination theory.

V.A.1.b. Engineering Analysis & Materials Selection for SS#1-P1- Solid Disk Wheel

Pham

Type of Engineering Analysis Conducted: Validation of composite components using classical lamination theory

Outcomes/Conclusions: The primary outcome of the validation is that the stresses seen in different plies were below the strength of the composite in the longitudinal and transverse plane by orders of magnitude. The highest stress seen was **-11.7 ksi** in the first ply and **7.2 ksi** in the third ply compared to a longitudinal stress of carbon fiber at **1500 Msi**.

V.A.1.c. Engineering Analysis & Materials Selection for SS#1-P1- Solid Disk Wheel

Pham

Type of Engineering Analysis Conducted: Stress Analysis at impact on the core component

Outcomes/Conclusions: The primary outcome of the stress analysis at impact on the core component was to see if the core component of the 2019 rover's wheel would yield on impact due to utilizing a low-density polyurethane foam. It was concluded that an impact force of 1400 lbf will produce a compressive stress in the core of **160 psi**; this is higher than the compressive strength of low-density polyurethane foam (5.6 lbf/ft³) of **130 psi** [2]. A balsa wood core would provide a compressive strength of **950 psi** but at a slightly higher density of 6.5 lbf/ft³.

V.A.1.d. Engineering Analysis & Materials Selection for SS#1-P1- Solid Disk Wheel

Pham

Type of Engineering Analysis Conducted: Moment of inertia calculations of different material

Outcomes/Conclusions: The primary outcome of the moment of inertia calculation is a moment of inertia of **282.2 in⁴**. This moment of inertia was used in bending analysis as well as buckling analysis for the wheel.

V.A.1.e. Engineering Analysis & Materials Selection for SS#1-P1- Solid Disk Wheel

Pham

Type of Engineering Analysis Conducted: Buckling analysis on a wheel

Outcomes/Conclusions: The primary outcome of the buckling analysis was that buckling force required to buckle the solid disk sandwich structure composite is **50 Msi** which is much higher than any expected force seen throughout the course. The main point of failure would come from compressive yield or transverse impact on the face of the wheel.

Engineering Analysis for SS#2- Single Bar Frame

Types of Engineering Analysis Conducted:

- Material Selection
- Static Load Analysis
- Impact Analysis
- Fatigue Analysis
- Tipping Analysis – side to side
- Tipping Analysis – front to back
- Drivetrain Bracketing Assembly Load Analysis

V.A.1.f. Engineering Analysis & Materials Selection for SS#2-P1-12 – Frame

Smith

Type of Engineering Analysis Conducted: Material Selection

Outcomes/Conclusions: The material index for the light and stiff beam frame is $\frac{\sqrt{E}}{\rho}$ and the materials considered for the frame are Steels, Aluminum Alloys, and Carbon Fiber. **Aluminum** was selected due to its manufacturability and weight.

V.A.1.g. Engineering Analysis & Materials Selection for SS#2-P1-5 – Main Frame

Smith

Type of Engineering Analysis Conducted: Static Load Analysis

Outcomes/Conclusions: The maximum moment acting on the frame during static loading is $M_{Max} = 3.752 \text{ in} - \text{kip}$ with a maximum deflection of $\delta_{Max} = 0.0311 \text{ in}$. The maximum transverse shear stress is $\tau_{Max} = .350 \text{ ksi}$ with a maximum bending stress of $\sigma_{Max} = 2.65 \text{ ksi}$. The results yielded values well below the yield stress for aluminum 6061-T6 therefore failure during static loading was not a concern.

V.A.1.h. Engineering Analysis & Materials Selection for SS#2-P1-5 – Main Frame

Smith

Type of Engineering Analysis Conducted: Impact Load Analysis

Outcomes/Conclusions: The maximum force seen by the frame during impact is $F_T = 406 \text{ lbf}$. The maximum moment acting on the frame during impact loading is $M_{Max} = 6.708 \text{ in} - \text{kip}$ with a maximum deflection of $\delta_{Max} = 0.207 \text{ in}$. The maximum transverse shear stress is $\tau_{Max} =$

. **509 ksi** with a maximum bending stress of $\sigma_{Max} = 4.74 \text{ ksi}$. The maximum torsional stress induced by impact on one wheel was found to be $\tau_T = 6.71 \text{ ksi}$. The results yielded a Von Mises Stress of $\sigma_{V.M} = 12.6 \text{ ksi}$ and a safety factor with respect to yield of 3.19. The safety factor ensured failure in the worst-case impact scenario was not a concern.

V.A.1.i. Engineering Analysis & Materials Selection for SS#2-P1-5 – Main Frame

Smith

Type of Engineering Analysis Conducted: Fatigue Analysis

Outcomes/Conclusions: Assuming a worst-case scenario of fully reversed bending and torsion during maximum impact loading. Using the maximum bending stress of $\sigma_{Max} = 4.74 \text{ ksi}$ and the maximum torsional stress of $\tau_T = 6.71 \text{ ksi}$. For bending the frame has **indefinite life**. For torsion the frame has a life of **211,228 cycles to failure**.

V.A.1.j. Engineering Analysis & Materials Selection for SS#1-7– Full Rover

Smith

Type of Engineering Analysis Conducted: Tipping Analysis Side to Side

Outcomes/Conclusions: The maximum allowable height of the center of gravity of the rover on a 30° incline is $H_{COG} = 4.33 \text{ ft}$ when the rover is in a side to side configuration. This exceeds the height of the rover from the ground of $H_{Rover} = 3.64 \text{ ft}$ therefore tipping in the side to side configuration is not a concern.

V.A.1.k. Engineering Analysis & Materials Selection for SS#1-7– Full Rover

Smith

Type of Engineering Analysis Conducted: Tipping Analysis Front to Back

Outcomes/Conclusions: The maximum allowable height of the center of gravity of the rover on a 30° incline is $H_{COG} = 5.20 \text{ ft}$ when the rover is in a front to back configuration. This exceeds the height of the rover from the ground of $H_{Rover} = 3.64 \text{ ft}$ therefore tipping in the front to back configuration is not a concern.

V.A.1.l. Engineering Analysis & Materials Selection for SS#2-P6-12 – Drivetrain and Rear Bracketing

Smith

Type of Engineering Analysis Conducted: Load Analysis for Drivetrain Bracketing

Outcomes/Conclusions: The worst-case scenario is found during impact with the force of the chain pulling up on the internally geared hub. The maximum force seen by the bracketing is in the front drivetrain area and was found to be $F_{Max} = 957 \text{ lbf}$ in the vertical direction. This creates a

maximum moment of $M_{Max} = 1.91 \text{ in} - \text{kip}$. The minimum allowable thickness of the bracketing is $t_{min} = 0.387 \text{ in}$ at the most concerning point of failure.

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V.A.2. Engineering Analysis for SS#3- Front Tandem Drivetrain

Types of Engineering Analysis Conducted:

- Mathematical model illustrating output rover speed range based on variable cog size (gear ratios), input rider cadence, and condition of internally-geared hub (underdrive/overdrive)
- Minimum torque required for the rover to traverse the High Butte obstacle. Size the low gear ratio according to this torque value.
- Stress analysis on driveshaft/driveshaft components
- Maximum chain tension assuming a worst-case scenario load on the chain/cog system.
- Bearing sizing based on worst-case scenario loading conditions
- Stress analysis incorporating local stress concentration due to hole cut into frame
- Drivetrain transmission diagram
- Hub Anchor: Proof of concept

V.A.2.a. Engineering Analysis & Materials Selection for SS#3-P1, P2, P9 - Surly 36T/18T Cog & NuVinci N380 Internally-Geared Hub

Prejean

Type of Engineering Analysis Conducted: Mathematical model illustrating output rover speed range based on variable cog size (gear ratios), input rider cadence, and condition of internally-geared hub (underdrive/overdrive)

Outcomes/Conclusions: The primary outcome of creating the model was that the rover's average output speed over the course of eight minutes will be between **5-17 mph**. This result is significant because, as stated in the quantitative constraints, the rover must travel at a minimum average velocity of 4 mph in order to complete the course in eight minutes. Being that the actual calculated average speed range is largely greater than the minimum speed constraint, the result is satisfactory.

V.A.2.b. Engineering Analysis & Materials Selection for SS#3- P1, P2 Surly 36T/18T Cogs

Prejean

Type of Engineering Analysis Conducted: Minimum torque required for the rover to traverse the High Butte obstacle. Size the low gear ratio according to this torque value.

Outcomes/Conclusions: The outcome of this analysis was that the minimum torque required for the rover to traverse the High Butte obstacle will be **239 ft-lbf**. The corresponding low gear ratio required to produce this torque is **0.95 ~ 1.0**. The cogs within the drivetrain can be sized to support this value.

V.A.2.c. Engineering Analysis & Materials Selection for SS#3-P3, P4, P11 – Aluminum Driveshaft and Telescopic U-Joints

Gremillion

Type of Engineering Analysis Conducted: Stress analysis on driveshaft/driveshaft components

Outcomes/Conclusions: The outcome of this analysis was that the maximum compressive stress induced onto the driveshaft by the NuVinci N380 internally geared hub is **6.30 ksi**. This result yields a safety factor with respect to yield of **6.3**. This analysis proved that using an aluminum 6061-T6 solid rod with an outer diameter of 1.18 inches will be adequate for the worst-case loading conditions. This outer diameter was a constraint for the analysis, since Team 24 will be reusing the U-Joints from the 2019 Rover.

V.A.2.d. Engineering Analysis & Materials Selection for SS#3-P3, P5, P7, P10 – ASTM #40 Standard Roller Chain, Crankset, and Surly Singleator Chain Tensioner

Prejean

Type of Engineering Analysis Conducted: Maximum chain tension assuming a worst-case scenario load on the chain/cog system.

Outcomes/Conclusions: The maximum chain tension based on a worst-case scenario loading condition will be **~1050 lbf**. This results in a factor of safety relative to the ultimate stress of an ANSI Standard #40 roller chain of **2.98**.

V.A.2.e. Engineering Analysis & Materials Selection for SS#3- P12 – Pillow Block Bearing

Gremillion

Type of Engineering Analysis Conducted: Bearing sizing based on worst-case scenario loading conditions

Outcomes/Conclusions: The outcome of this analysis was that during the worst-case loading scenario the bearings will see a load of **683 lbf**. The radial ball bearings suitable for this load with a bore of 1.18 inches is an **L06 series** which has a load capacity of **753 lbf**. The 2019 Rover Team used 206 series flanged radial ball bearings which have a 1.18 inch bore and a rated load

capacity of **1214 lbf**; therefore, Team 24 will reuse the 2019 Rover Team's bearings to support the driveshaft.

V.A.2.f. Engineering Analysis & Materials Selection for SS#3- P7, P8 – Square-Tapered Bottom Bracket and Bottom Bracket Shells

Gremillion

Type of Engineering Analysis Conducted: Stress analysis incorporating local stress concentration due to hole cut into frame

Outcomes/Conclusions: The outcome of this calculation was that the frame will not yield at the crankset locations. The frame was modeled as a plate with a hole under uniform tension, and under the most strenuous loading conditions the maximum stress seen in the frame was **3.05 ksi**. This maximum stress results in a safety factor with respect to yield of **13.11**.

V.A.2.g. Engineering Analysis & Materials Selection for SS#3- P1, P2, P9 – Surly 36T/18T Cog & NuVinci N380 Internally-Geared Hub

Prejean

Type of Engineering Analysis Conducted: Drivetrain transmission diagram

Outcomes/Conclusions: The number of teeth of cogs 1-7 will be 36. The number of teeth on cogs 8-10 will be 18. The corresponding gear ratio between cogs 1-2, 3-4, and 5-6 will be **1.0**. The gear ratio between cogs 7-8 will be **0.5**. The gear ratio range between cogs 8-9 based on the internally geared hub will be **0.5-1.9**. The gear ratio between cogs 9-10 will be **1.0**.

V.A.2.h. Engineering Analysis & Materials Selection for SS#3- P6 – Hub Anchor

Prejean

Type of Engineering Analysis Conducted: Hub Anchor: Proof of concept

Outcomes/Conclusions: The hub anchor will be used to prevent translation of the internally-geared hub in the direction of the bracket slot.

V.A.3. Engineering Analysis for SS#4- Push-Pull Steering

Types of Engineering Analysis Conducted:

- Turning Radius Analysis
- Tie Rod Buckling Load
- Shear Force on Bolts
- Range of Motion Limiter Weld Analysis
- Steering Arm Weld Analysis

V.A.3.a. Engineering Analysis & Materials Selection for SS#4-P1-P11 – Steering Assembly

Donner

Type of Engineering Analysis Conducted: Turning Radius Analysis

Outcomes/Conclusions: An inner turning angle of **23°** and an outer turning angle of **16.3°** will ensure that the rover achieves a turning radius of 13 ft. This is within NASA's turning radius requirement of 15 ft or less. These angles are achieved by the tie rod linkage.

V.A.3.b. Engineering Analysis & Materials Selection for SS#4-P2 – Tie Rod

Donner

Type of Engineering Analysis Conducted: Tie Rod Buckling Load.

Outcomes/Conclusions: The tie rod will be 40 in. long with a cross section of $\frac{3}{4} \times 1$ in. This tie rod will be able to withstand just over **2100 pounds** of force before buckling. This is far more force than the tie rod is expected to receive. This analysis was done because of issues experienced by previous rover teams.

V.A.3.c. Engineering Analysis & Materials Selection for SS#4-P9 – Tie Rod Bolts

Donner

Type of Engineering Analysis Conducted: Shear Force on Bolts

Outcomes/Conclusions: Assuming the 3/8-inch aluminum bolts will not experience a force greater than 800 pounds, their factor of safety with respect to shear is **4.14**.

V.A.3.d. Engineering Analysis & Materials Selection for SS#4-P1,P5 – Steering Knuckle, ROM Limiter

Donner

Type of Engineering Analysis Conducted: Range of Motion Limiter Weld Analysis

Outcomes/Conclusions: Assuming the impact force experienced by the Range of Motion Limiter will not exceed 2000 pounds of force, the max stress on the fillet welds, considering its geometry, will not exceed **1500 psi**. Therefore, the welds will not fail.

V.A.3.e. Engineering Analysis & Materials Selection for SS#4-P3,P4- Steering Arm, Bolt Plate

Donner

Type of Engineering Analysis Conducted: Steering Arm Weld Analysis

Outcomes/Conclusions: Assuming the force experienced by the Steering Arm will not exceed 1000 pounds of force, the max stress on the fillet welds, considering its geometry, will not exceed **1500 psi**. Therefore, the welds will not fail.

V.A.4. Engineering Analysis for SS#5- Hydraulic Brakes

Types of Engineering Analysis Conducted:

- Stopping Distance and Stopping Time
- Forces on Caliper Pistons of braking system

V.A.4.a. Engineering Analysis & Materials Selection for SS#5P1 – Hydraulic Brakes

Guo

Type of Engineering Analysis Conducted: Stopping Distance and Stopping Time

Outcomes/Conclusions: Assuming an average rover speed of 10 mph and a combined driver and rover weight of 431.1 lbs, the braking distance for the rover would be **5.6 ft** and the stopping distance would be **0.8 s**.

Type of Engineering Analysis Conducted: Forces on Caliper Pistons of braking system

Outcomes/Conclusions: Assuming average male grip strengths of 112.75 lbf and 104.5 lbf, the maximum force applied to the caliper piston is **490.4 lbf**.

V.A.5. Engineering Analysis for SS#6- Double Wishbone Suspension

Types of Engineering Analysis Conducted:

- Impact analysis on the entire rover.
- Shock angle and mounting arm location calculations.
- Minimum thickness of suspension bending arms.

Type of Engineering Analysis Conducted: Determine the maximum impact force applied to the rover. The impact force will likely be the highest force the rover encounters so this force will determine the thickness of various members.

Outcomes/Conclusions: This analysis was able to show that the highest impact force seen on the rover will come from the male rider in the front. The max impact force at the front of the rover was calculated to be **1,341.31 lbf**.

Type of Engineering Analysis Conducted: Trigonometric calculations to determine the location of the shock mounting arm and the angle of the shock.

Outcomes/Conclusions: The shock will be mounted at **34.6°** and the mounting arm on the control arm will be **2.95"** horizontal from the steering knuckle towards the frame.

V.A.5.c. Engineering Analysis & Materials Selection for SS#6-P4 – Bottom A-frame suspension

Hebert

Type of Engineering Analysis Conducted: Bending stress analysis to determine the minimum thickness of the suspension arms.

Outcomes/Conclusions: The major outcome of this bending analysis was that the suspension arm thickness was determined to be safe at a **0.1 in.** thickness.

V.A.6. Engineering Analysis for SS#7- Seats

Types of Engineering Analysis Conducted:

- Euler's column analysis on the seat post.
- Stress analysis on bolts used to attach the seat to the frame.
- Stress analysis on brackets used to attach the seats.
- Calculate the maximum stress exerted on the rear seat handles.

V.A.6.a. Engineering Analysis & Materials Selection for SS#7-P3, P4 – Front seat bar, rear seat bar

Hebert

Type of Engineering Analysis Conducted: Euler's Column Analysis was conducted to determine the minimum width for the rear seat attachment bar.

Outcomes/Conclusions: Euler's analysis determined that the seat bar's minimum width had to be **1.93 in.**

V.A.6.b. Engineering Analysis & Materials Selection for SS#7-P10 - Bolts, 2 ½" long, Zinc-plated ¼"- 20 Thread

Hebert

Type of Engineering Analysis Conducted: Calculate the max shear stress the bolts used to attach the seat will face in our worst-case scenario.

Outcomes/Conclusions: The bolts selected to can withstand a shear stress of 90,000 psi, but will see a max of **24,490 psi** giving them a safety factor of **3.7**.

V.A.6.c. Engineering Analysis & Materials Selection for SS#7-P6, P7 – Top mounting bracket, bottom mounting bracket

Hebert

Type of Engineering Analysis Conducted: Calculate the max tensile force on the seat attachments.

Outcomes/Conclusions: The top seat bracket faces a max tensile stress of **18,182 psi** which means they have a safety factor of **2.2**. While the bottom seat bracket will see a max tensile stress of **8,333 psi** which means they have a safety factor of **4.8**.

V.A.6.d. Engineering Analysis & Materials Selection for SS#7-P12 – Rear seat handles

Hebert

Type of Engineering Analysis Conducted: Calculate the maximum yield stress exerted on the rear handles.

Outcomes/Conclusions: The maximum stress seen by the handles is **11,200 psi**, which gives the component a factor of safety of **3.125**.

VI. Manufacturing & Assembly

VI.A. Manufacturing Processes

Manufacturing the 2020 rover will consist of both manufacturing of individual parts and assembly of collections of parts to form the seven subsystems that compose the rover. The manufacturing techniques that will be used in realization of the prototype include CNC milling, lathing, sanding, welding and heat treating, water-jetting, grinding, 3D printing, and vacuum-bagging. The actual techniques required to manufacture each subsystem vary amongst those stated above and will be addressed in the following text during subsystem breakdown of the manufacturing processes.

The wheels and hubs will require wood-cutting, 3-D printing, vacuum-bagging, lathing, and milling. The body of the wheel (SS#1-P1) will be a sandwich structure composite using balsa wood as the core and carbon fiber as the composite. The balsa wood will be cut using standard wood-cutting techniques and the carbon fiber will also be cut to size. Due to the thickness of the balsa wood, sheets of balsa wood will have to be stacked and adhered together to the desired thickness of the wheel; carbon fiber will then be applied on top of the balsa wood and bonded using epoxy resin. Three sheets of carbon fiber will be attached at 0°, 45°, and 90° orientations; epoxy resin will be applied on top of every sheet. The bond will be finished using vacuum bagging to get rid of any air bubbles and excess epoxy, ensuring a good bond between the composite and the core. The aluminum hubs and hub connector (SS#1-P2,3, & 8) will be machined using the lathe and mill. The tread (SS#1-P8) will use a negative mold that is 3-D printed and then casted using a liquid rubber. The braking disk, nuts, and bolts (SS#1-P4,5, & 6) will be purchased.

The main manufacturing processes for the frame (SS#2-P1-11) include welding and water jet cutting. Water jet cutting will be used on the front and rear crankset supports (SS#2-P3,4) to create holes for the bottom bracket shells of the drivetrain. Water jet cutting will be used in the same manner for the front and rear crankset support angles as well as the holes in the front and rear frame section (SS#2-P1,2). Water jet cutting will be used to create all bracketing pieces for the drivetrain and rear bracketing assemblies. Once all frame pieces meet the necessary dimensions the front and rear crankset supports will be welded to the frame in their respective locations. Welding will be used to connect all pieces of the drivetrain and rear bracketing assemblies. The final frame subsystem will have the drivetrain and rear bracketing connected through welding the assemblies to their respective locations on the frame.

The majority of the components that compose the drivetrain subsystem will be purchased; thus, a manufacturing technique is not needed for them. However, aluminum welding will need to be utilized in order to fix the square-tapered bottom brackets (SS#3-P7) and bottom bracket shells (SS#3-P8) to the rover's main frame. The specific welding will take place circumferentially around the four bottom bracket shells bored into the frame. The weld type will consist of a circumferential fillet weld with a weld leg of 0.125 inches. The manufacturing drawing for this manufacturing process can be found in XI.C.3.c, Figure XI-18-Manufacturing Drawing of SS#3-P7,P8.

The various components that comprise the Push-Pull Steering Assembly must be manufactured from raw aluminum plates and rods. The most complex component, the Steering Knuckle (SS#4-P1), must be manufactured using CNC milling. The specifications for this component can be found in XI.C.4.c, Figure XI 24: Manufacturing Drawing of SS#4P1. The

Steering Handles (SS#4-P6,P7) will be manufactured by cutting a straight aluminum rod into sections of specific lengths with edges cut at specific angles, as specified in XI.C.4.c, Figure XI 29,30: Manufacturing Drawings of SS#4P6,P7. These sections will be joined with circumferential welds. The remaining components will be manufactured using standard milling and lathing techniques to achieve their desired geometries. Their specifications can be seen in their respective Manufacturing Drawings found in XI.C.4.c.

The manufacturing process for the suspension will consist of a combination of CNC'ing, water jetting, and welding. The top and bottom A-Frame's (SS#6-P1,P2) will be water jetted to the correct geometry. Then the A-Frame connectors (SS#6-P3) can be cut out using a CNC machine. Once all parts are cut, they will be welded together to form the double wishbone. The shocks (SS#6-P7, P8) will then be bolted to the wishbone connectors to finalize the suspension. The manufacturing processes for the brakes include reusing the braking system from last year's rover team in order to save time and overall cost. The final brake subsystem will consist of a metal rotor, lever, caliper piston and hose filled with an incompressible fluid.

The seats manufacturing process will be slightly different for the rear seat and front seat due to the collapsible volume portion of the competition. Both seat bars (SS#7-P3, P4) will be cut using a saw and mill to get them to the proper length/angles. The front seat bar (SS#7-P2) will be welded directly to the frame (SS#2-P2), while the rear seat bar is bolted into the rear seat holder (SS#7-P8). The rear seat holder will be welded together then to the frame. The bottom seat to frame attachment (SS#7-P5) will be an Aluminum bar that is cut in half then welded to the frame. This where the seats will attach to the mounting brackets (SS#7-P6, P7) which will each be cut using a CNC machine.

VI.B. Assembly Process

The Frame subsystem, SS#2, has multiple tie points to the suspension and drivetrain as discussed in future paragraphs.

The drivetrain subsystem, SS#3, will require tie points into several subsystems. Firstly, the drivetrain will be tied into the frame by the use of bottom bracket shells (SS#3-P7). The frame will have holes bored at four specified locations on the frame for each bottom bracket. The bottom bracket shells will be tied into the frame at these locations through welding. Once the bottom bracket shells are secured to the frame, the bottom brackets and corresponding cranksets will be inserted. The next phase of the drivetrain assembly process will be located on the bottom drivetrain bracket located underneath the single bar frame.

The drivetrain bracket assembly is utilized to house the NuVinci N380 Internally-Geared Hub (SS#3-P9) and the driveshaft (SS#3-P11). The internally geared hub will be inserted into the bottom drivetrain bracket through the use of the hub support brackets. Similarly, the driveshaft will be inserted into the driveshaft support brackets. Two 206 series flanged Pillow-Block Bearings (SS#3-P12) will be attached to the outside of the driveshaft support bracket and will be utilized to support the driveshaft. The driveshaft will be mounted in two telescopic u-joints (SS#3-P4) via a set screw. The u-joints will be responsible for tying the drivetrain into the wheel (SS#1) subsystem.

The u-joints will tie into the wheel subsystem via the wheel hub connector (SS#1-P3). The u-joints will fit over the wheel hub shaft and will be secured via both set screws and keyway/key

mates. This assembly process will ensure that the drivetrain subsystem will convert driver input into useful rover output.

The suspension acts as intermediary between most of the subsystems. The suspension will be connected to the frame using the wishbone connectors (SS#6-P7). These connectors will be bolted to the frame. Two different joints will be used to connect the suspension to the steering knuckles (SS#4-P1). The front suspensions systems will be connected to the steering knuckle using ball joints (SS#6-P6) to allow for the steering motion to occur. While the rear suspensions systems will be connected to the rear knuckles using clevis rods (SS#6-P5).

The Push-Pull Steering design will tie the suspension, wheels, and drivetrain together. This will be facilitated by the steering knuckle (SS#4-P1). First, the wheel hub connector (SS#1-P3) will slide through the steering knuckle's u-joint cut-out to connect with the drivetrain's telescopic u-joint (SS#3-P4). Next, the top and bottom suspension control arms (SS#6-P3,P4) screw into the steering knuckle within their respective cut-outs through the control arms' ball joints.

The hydraulic braking system (SS#5-P1) will connect directly to the hub of the wheel (SS#1) in order to minimize wear and lighten overall load to the rim. The lever and piston system will attach to the steering handle via a rubber wrap that will go around the handle body.

The wheel subsystem, SS#1, will be assembled by bolting the inner wheel hub (SS#1-P2) and the outer wheel hub (SS#1-P5) onto the main body of the wheel (SS#1-P1) using the outer diameter holes. The outer wheel hub (SS#1-P5) will then be bolted to the hub connector (SS#1-P3) using the inner diameter holes. This assembly will allow the wheels to attach to the steering subsystem, drivetrain subsystem, and the braking subsystem; this ensures that the wheel subsystem will traverse the demanding obstacle course and support the load of the riders and rover weight.

Each seat will be connected to the frame at two points. The front seat bar (SS#7-P3) will be welded to the front frame bar (SS#2-P1) while the rear seat holder (SS#7-P8) is welded to (SS#2-P2). The bottom seat to frame attachments (SS#7-P5) will be manufacture, by cutting a rod in half to create two separate pieces. Each of these pieces will be drilled to fit bolts (SS#7-P10) then they will be welded to the frame (SS#2-P1, P2). Once all the pieces are cut and welded together the seat's assembly can be put together using 6 bolts (SS#7-P9, P10).

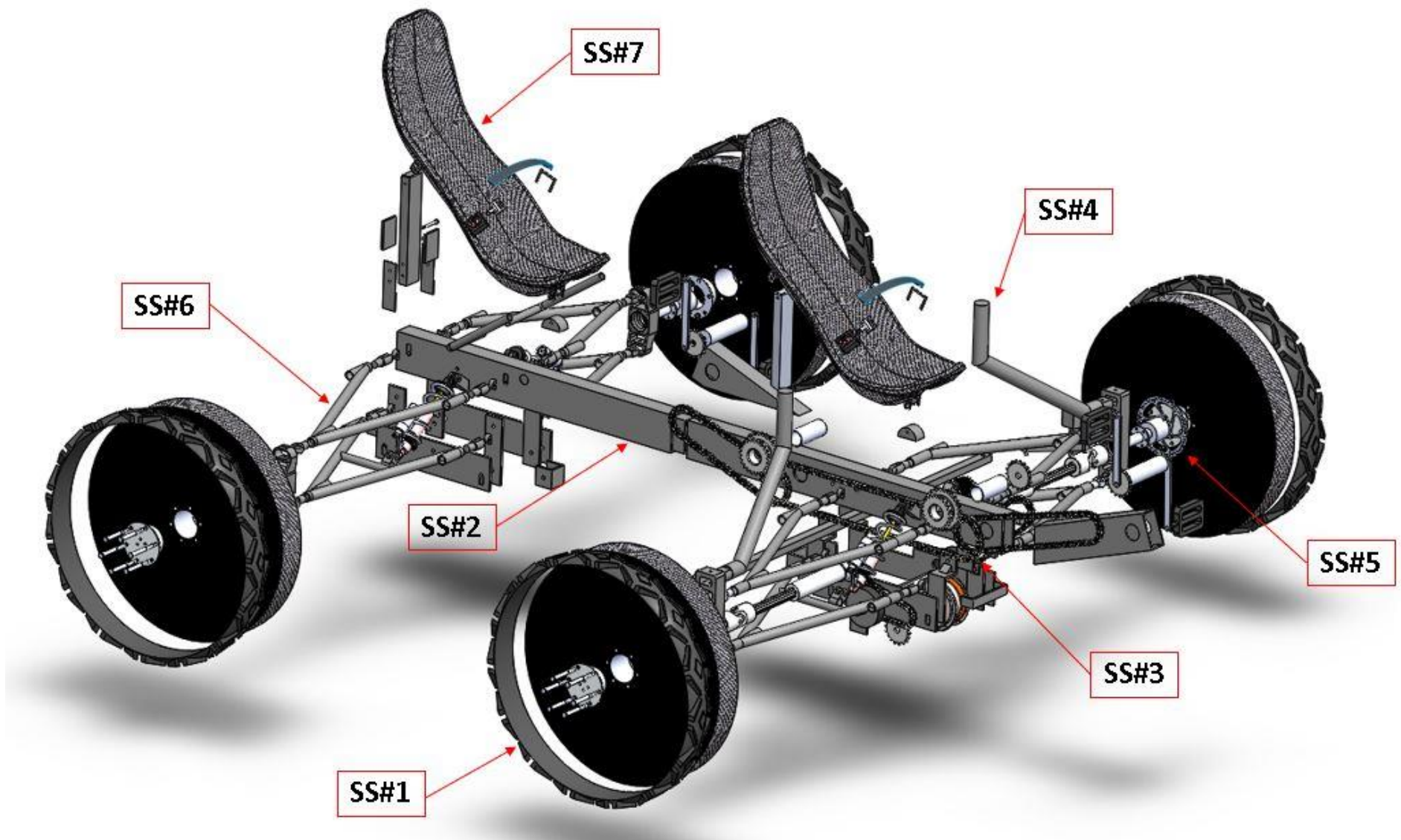


Figure VI.B.1: Exploded View Assembly Drawing of Prototype – Sub-System Level

VII. Safety Considerations

VII.A. Operational Safety

The NASA Human Exploration Rover Challenge 2020 Guidebook identifies all safety guidelines that must be adhered to in order for Team #24 to participate in the competition. Prior to competition start, all safety requirements will be checked for compliance by each team. Disqualification will result if the requirements are not strictly adhered to.

The NASA Human Exploration Rover Challenge 2020 Guidebook identifies four major safety categories: Seat Restraints, Protrusions, Passenger Attire, and Injury and Signals [1].

The Seat Restraint category discusses the requirement of seat restraints for each of the two rover riders. The Guidebook states, “Failing to provide robust, practical seat restraints can result in not being allowed to compete on the course [1].”

The Protrusion category states that all sharp edges and protrusions must be eliminated by padding or guards as necessary.

The Passenger Attire section identifies specific personal protective equipment to be worn by the rover riders during competition. These include: eye protection, commercially-manufactured head protection, full fingered gloves, long sleeved and long-torso shirts, long pants, and socks/shoes [1].

The Injury and Signals section defines a rover rider who is unable to continue in competition as one who is: “injured, bleeding, or incapacitated [1].” As well, riders are expected to develop and utilize some form of signaling technique to communicate between the riders when all extremities are clear from moving parts and ready to continue the competition. The drivers will be expected to communicate their communication plan with the safety personnel prior to competition start.

VII.B. Safety During Manufacturing, Assembly and Testing

All safety guidelines outlined by the “MIE Advanced Manufacturing & Machining Facility (AMMF) Rules and Usage” will be adhered to in their entirety by Team #24 when manufacturing, assembling, and testing the LSU 2020 NASA Rover [5].

The “MIE Advanced Manufacturing & Machining Facility (AMMF) Rules and Usage” identifies five safety instructions for all persons in the AMMF [5]. These include:

1. Safety glasses and a name tag must be worn at all times in the AMMF;
2. No shorts, no loose clothes, no open-toed shoes or sandals can be worn;
3. Long hair must be fastened above ear level, preferably under hat or net;
4. Abide by all safety postings and ask for help when in doubt;
5. After hours use requires supervision – you cannot work alone. No guests are allowed after hours. All users must sign in and out.

Along with these guidelines for general AMMF use, more specific safety precautions will be taken based on the current job scope. Most manufacturing processes to be used in the creation of the rover prototype (milling, lathing, welding) were learned and practiced in ME 3633 –

Manufacturing Processes. Safety protocol executed in this class will be reciprocated during prototype creation.

VIII. Testing, Validation, and/or Implementation

VIII.A. Design-Phase Testing

The competition requires one male and one female driver for each team. To select the best drivers for the rover, strength and endurance testing was performed at the LSU Recreational Center. For endurance testing, each team member rode stationary bikes to measure their performance over an eight-minute period which yielded an average velocity (EFS, Endurance Factor). For strength testing, each team member performed ten repetitions on a leg press. This data was used to calculate each team members strength to weight ratio (SFS, Strength Factor).

Five of the fourteen course obstacles were identified as requiring strength to overcome, while all fourteen obstacles require endurance since the course must be completed within eight minutes. This allowed the team to place a 65% weight on the endurance factor and a 35% weight on the strength factor. The results of each test were nondimensionalized and a formula was created to score each candidate based on their strength and endurance factors, see appendix XII.C.2 Design Phase Testing Results. Each candidates' results were evaluated and ranked which yielded the best male and female candidate to drive the rover for the 2020 competition.

Team 24 had the opportunity to test the 2019 rover design. The main goals of testing were to acquire some meaningful data to be used in analysis of the 2020 rover, determine what worked well in the 2019 design and determine what needed to be improved from the 2019 rover. The old rover was able to reach an average speed of 7.92 mph, an average breaking distance of 4.17 ft, and a minimum turning radius of 14.0 ft. Team 24 considered the suspension and breaking from the 2019 rover to be used in the 2020 rover due to the success during the testing. The main complications from testing the 2019 rover were found in the front and rear drivetrains. The rear drivetrain never worked during testing so a secondary driver could not be used during testing. Additionally, the front drivetrain hub consistently came out of place from the slots in the frame. These complications led Team 24 to place a heavy emphasis on design and analysis of the 2020 drivetrain.

Team 24 also conducted testing on the wheels of the 2019 rover to determine a baseline of forces to move the rover on different terrains. Terrains tested included concrete, gravel, grass as well as a 10° and 20° incline to simulate obstacles that the team will face during the actual competition. The conclusion included max forces of 49.1 lbf, 71.7 lbf, 68.0 lbf, 103.8 lbf, and 170.5 lbf for the respective different terrains (Appendix XII.C.2.). An impact study was also conducted over a 6 in. bump. The major observation made was the hub of the wheel coming out of its position and compressing the inner foam material of the wheel. See Appendix XII.C.2. for reference images. The information gathered was used as a baseline in the drivetrain and wheel analysis and selection.

VIII.B. Preliminary Testing, Validation and/or Implementation Plan

Team 24 have set a team manufacturing completion deadline of March 16th to ensure a month of testing for the rover. This will allow the team to address any issues in the design and

account for these issues before competition April 16th-17th. The following are several tests that will be conducted on the completed rover.

Welds will be inspected using 3 types of non-destructive testing (NDT). The first NDT being a visual inspection on all welds to detect any major surface flaws. Then a liquid penetrant test will be conducted on all welds throughout the design. The liquid penetrant, seen in XII.C.3, will be used for a more in-depth surface flaw detection. The final NDT conducted will be phased array ultrasonic testing (PAUT) using LSU's equipment seen in Figure XII-XII.C.1: Liquid Penetrant Non-Destructive Testing [6]. PAUT will provide a detailed inspection of the weld below the surface and will only be used on the bottom A-frame suspension arms (SS#6-P4) because these welds will face the largest force throughout the rover.

The team will also construct an obstacle course per NASA's description of the actual competition course. The team will traverse the course on the rover multiple times in order to ensure that the rover can compete all 14 obstacles. The testing will also allow the drivers to practice completing the necessary tasks and completing the course within the 8-minute time constraint.

IX. Project Management

IX.A. Schedule and Milestones

The most significant milestone for Team 24 is the NASA Human Exploration Rover Challenge as seen in Figure IX-1 below. The Challenge takes place on April 17 and 18, 2020 in Huntsville Alabama. To be successful during the competition, the Team imposed a milestone to complete manufacturing by March 16, 2020. This would allow Team 24 a month to adequately test and improve the rover for competition. In order to complete manufacturing by March 16, 2020 Team 24 provided a Bill of Materials (BOM) on November 15, 2019. Additionally, Team 24 has provided a manufacturing plan on December 11, 2019. If the Team had not provided a BOM or manufacturing plan by the specified dates, the manufacturing completion date would need to be pushed back. Therefore, the BOM and manufacturing plan were the critical path for the project to be completed successfully. The current critical path is completion of manufacturing by March 16, 2020. The most significant point of failure for the project is inadequate time to test and improve the newly designed rover.

NASA Rover Challenge

Team 24

SIMPLE GANTT CHART by Vertex42.com
<https://www.vertex42.com/ExcelTemplates/simple-gantt-chart.html>

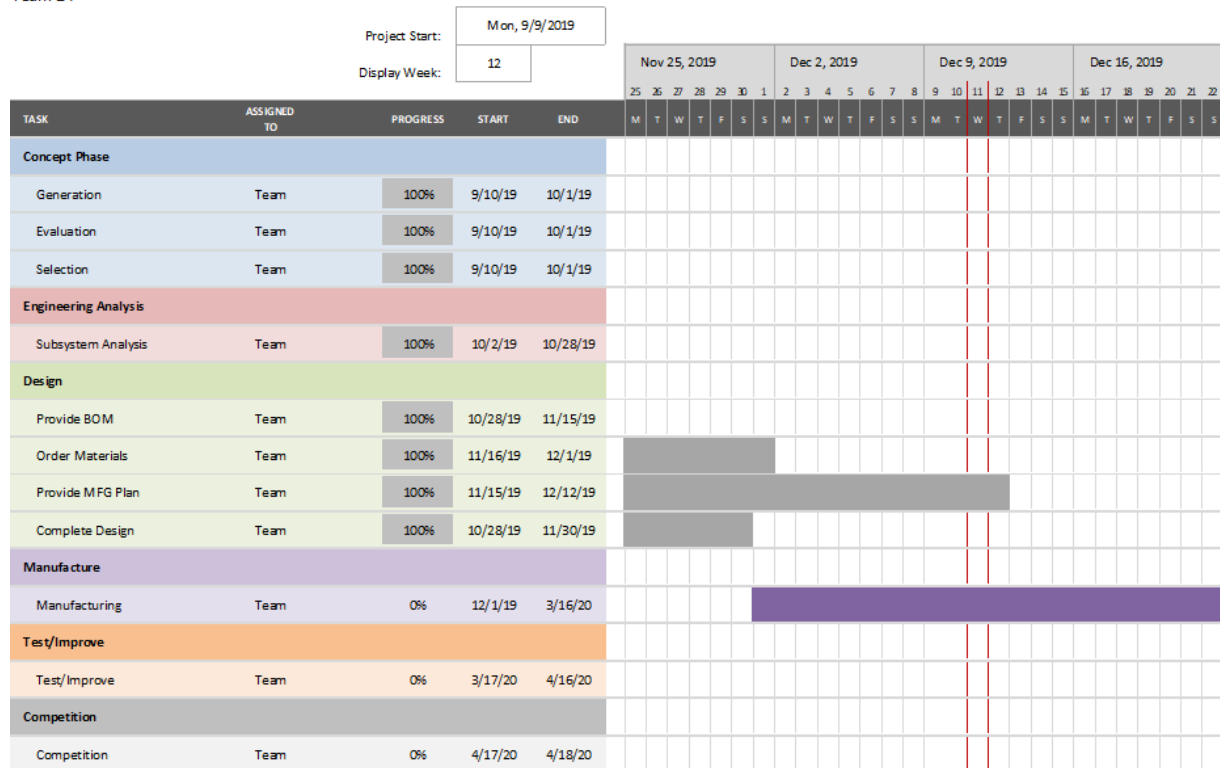


Figure IX.A.1: Gantt Chart with Project Timeline

IX.B. Budget

Team 24 has been given a sponsor-imposed budget of \$7,000. The Team received a donation from ExxonMobil of \$1,000 which increased the overall budget to \$8,000. The additional funding is justified by a more complete redesign of the rover that was not undergone in past rover designs. Based on the bill of materials, Team 24 was able to create a budget breakdown which can be seen in Figure IX.B.1: 2020 Team 24 Fall Final Budget below. The contingency for the project is ~ 11% of the entire budget which is adequate for this phase of the project. The team determined that the wheels, drivetrain, and manufacturing would be the most cost intensive items which was also indicated in the 2019 Team's Final Budget. The 2019 Team Final Budget along with more budget details can be seen in Appendix XII.D.2 - Budget Details.

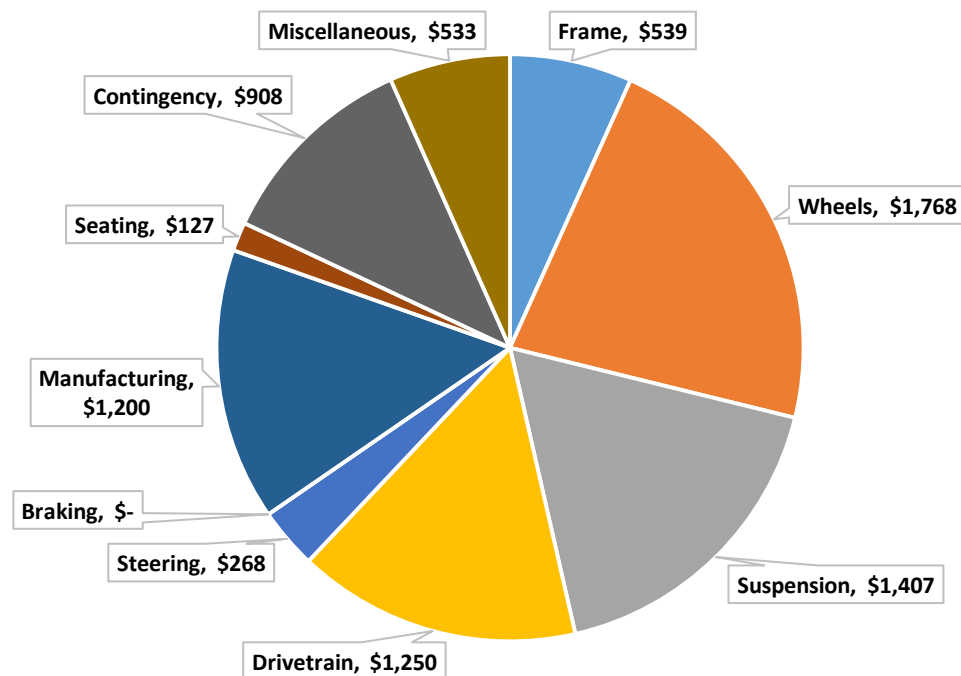


Figure IX.B.1: 2020 Team 24 Fall Final Budget

X. Summary and Conclusions

The 2020 LSU NASA Rover consists of seven subsystems that will support a successful, competitive design. These subsystems will work in unison to complete the team-defined functions for success: traverse a demanding obstacle course, support the load of two riders plus vehicle weight, store collected samples/task items, convert rider input (pedaling) into useful output, maintain durability throughout the competition, and easy & quick assembly.

All rover subsystem designs were justified through various types of engineering analysis conducted by Team 24. The broad range of analysis conducted yielded safety factors greater

than that of industry standards, proving adequacy and feasibility of concepts for integration into the design.

Manufacturing the 2020 rover will consist of both manufacturing of individual parts and assembly of collections of parts to form the seven subsystems that compose the rover. The manufacturing techniques that will be used in realization of the prototype include CNC milling, lathing, sanding, welding and heat treating, water-jetting, grinding, 3D printing, and vacuum-bagging.

In summary, Team 24 is working to create a robust, four-wheeled, tandem, recumbent, off-road quadricycle that will win the 2020 NASA Human Exploration Rover Challenge. Since the project was assigned, Team 24 has been working tirelessly to ensure that the 1st place trophy for the 2020 NASA Human Exploration Rover Challenge is taken home to LSU.

XI. Appendix

XI.A. Quality Function Deployment (QFD) - HoQ

During the functional requirement meeting, the team ranked the functions and qualitative constraints from highest to lowest priority, resulting in the corresponding weights in the House of Quality. The letters next to the functions specify the type of constraint: 'T' is a team constraint, 'S' is a sponsor constraint, and 'C' is competition constraints. "Safe", "Man-Powered", "Adhering to Competition Guidelines", "Custom Wheels", "Support the load of two riders plus vehicle weight", and "Compact Design" were given the maximum weight of 5 because these functions and constraints are requirements to compete. "Fast" and "Traverse a demanding obstacle course" were also given a weight of 5 because these functions and constraints will be how the team places first. "Maintain durability throughout the competition", "Lightweight", "Simple", "Maneuverable", "Versatile", "Powerful", and "Cost Effective" were given a weight of 4 because they were functional requirements that the team will improve from last year's iteration that are not required by competition rules. "Ergonomic", "Braking", and "Stable" were given a weight of 3 because they are accessories for the rover that may help in performance or comfort. "Aesthetic" and "Storage" were given a weight of 2 as they will not affect the performance of the rover significantly. "Easy and Quick Assembly" was also given a weight of 2 as it is only worth 2 points.

Row #0	Max Relationship Value	Relative Weight	Weight / Importance	Functions & Qualitative Constraints ("Whats")	Quantitative Constraints with Units ("Hows")	Column #0																
						Direction of Improvement: Minimize (↓), Maximize (↑), or Target (⊕)																
1	9	6.0	5.0	Safe (C & T)	Rover Speed (mph)	↑																
2	9	4.8	4.0	Maintain derability throughout the competition (T)	Rover Weight (lb)	↓																
3	9	4.8	4.0	Lightwieght (C & T)	Rover Turning Radius (ft)	↓																
4	9	4.8	4.0	Simple (T)	Rover Ground Clearance (ft)	⊕																
5	9	6.0	5.0	Fast (C & T)	Rover Assembly Time (s)	↓																
6	9	2.4	2.0	Easy and Quick Assembly (C & T)	Rover Assembly Steps (#)	↓																
7	9	6.0	5.0	Human-Powered (C)	Compact Length (ft)	⊕																
8	9	6.0	5.0	Support the load of two riders plus the vehicle weight (T)	Width (ft)	⊕																
9	9	6.0	5.0	Traverse Demanding Obstacle (T)	Compact Height (ft)	⊕																
10		3.6	3.0	Ergonomic (T)	Competition Time (min)	↓																
11		3.6	3.0	Braking (T)	Competition Obstacles Completed (#)	↑																
12		6.0	5.0	Adhere to Competition Guidelines (T & S & C)	Competition Tasks Completed (#)	↑																
13		4.8	4.0	Manuverable (T)	Rover Acceleration (ft/s^2)	↓																
14		4.8	4.0	Convert Rider Input (Pedaling) to useful output (T)	Center of Gravity (ft)	↑																
15		4.8	4.0	Cost Effective (S)	Torque Produced (ft-lb)	↓																
16		6.0	5.0	Compact (C)	Stopping Distance (ft)	↓																
17		6.0	5.0	Custom Wheels (C)	Cost (\$)	↓																
18		2.4	2.0	Aesthetic (T)	Parts to Manufacture (#)	↑																
19		3.6	3.0	Stable (T)	Coefficient of Traction	⊕																
20		4.8	4.0	Versatile (T)	Sharp Edges (#)	⊕																
21	9	2.4	2.0	Store collected sample/task items (T)	Seat Restraints (#)	⊕																
21				Target or Limit Value for Quantitative Specs	Energy Storage Systems (#)	⊕																
22				Difficulty (0=Easy, 10=Extremely Difficult)	Malfunctions During Competition (#)	⊕																
				Max Relationship Value	Degrees of Freedom in Steering (#)	⊕																
				Weight / Importance	Degrees of Freedom in Power Input (#)	⊕																
				Relative Weight	Wheels (#)	⊕																

Figure XI.A.1: The House of Quality for the project.

Row #0	Max Relationship Value	Relative Weight	Weight / Importance	Functions & Qualitative Constraints ("Whats")	Quantitative Constraints with Units ("Hows")	Column #0														Competitive Analysis				
						Direction of Improvement: Minimize (↓), Maximize (↑), or Target (⊕)														(0=Worst, 5=Best)				
1	9	6.0	5.0	Safe (C & T)	Stopping Distance (ft)	↓																		
2	9	4.8	4.0	Maintain derability throughout the competition (T)	Cost (\$)	↓																		
3	9	4.8	4.0	Lightwieght (C & T)	Parts to Manufacture (#)	↑																		
4	9	4.8	4.0	Simple (T)	Coefficient of Traction	⊕																		
5	9	6.0	5.0	Fast (C & T)	Sharp Edges (#)	⊕																		
6	9	2.4	2.0	Easy and Quick Assembly (C & T)	Seat Restraints (#)	⊕																		
7	9	6.0	5.0	Human-Powered (C)	Energy Storage Systems (#)	⊕																		
8	9	6.0	5.0	Support the load of two riders plus the vehicle weight (T)	Malfunctions During Competition (#)	⊕																		
9	9	6.0	5.0	Traverse Demanding Obstacle (T)	Degrees of Freedom in Steering (#)	⊕																		
10		3.6	3.0	Ergonomic (T)	Degrees of Freedom in Power Input (#)	⊕																		
11		3.6	3.0	Braking (T)	Wheels (#)	⊕																		
12		6.0	5.0	Adhere to Competition Guidelines (T & S & C)	Direction of Movement (#)	⊕																		
13		4.8	4.0	Manuverable (T)	Safety Factor	⊕																		
14		4.8	4.0	Convert Rider Input (Pedaling) to useful output (T)	Seating Angle (°)	⊕																		
15		4.8	4.0	Cost Effective (S)	Rover Climbing Angle (°)	⊕																		
16		6.0	5.0	Compact (C)	Rover Angle of Deflection (°)	⊕																		
17		6.0	5.0	Custom Wheels (C)	Tools (#)	⊕																		
18		2.4	2.0	Aesthetic (T)	Our Rover																			
19		3.6	3.0	Stable (T)	LSU 2019 Rover																			
20		4.8	4.0	Versatile (T)	Puerto Rico (2019 College Winner)																			
21	9	2.4	2.0	Store collected sample/task items (T)	Germany (2019 High School Winner)																			
21				Target or Limit Value for Quantitative Specs																				
22				Difficulty (0=Easy, 10=Extremely Difficult)																				
				Max Relationship Value																				
				Weight / Importance																				
				Relative Weight																				

Modified from GFD Online <https://www.gfdonline.com/>

Figure XI.A.2: The House of Quality for the project.

- “Tools (#)” is positively correlated to “Competition Tasks Completed (#)” and “Parts to Manufacture (#)” because the rover requires a tool to be manufactured for every task that will be attempted.

XI.B. Concept/Solution Evaluation and Selection Supplement

XI.B.1. Concept/Solution for Function F1 – Traverse a Demanding Obstacle Course - Details

XI.B.1.a. Concept Selected - Details

Metric	Weight	Four-Wheel Solid Disks	Four-Wheel I-Beam	Hybrid Four Wheel	Three Wheel Design
Stability	0.3	4	4	4	2
Traction	0.2	4	4	4	2
Manufacturability	0.1	4	2	1	4
Weight	0.2	2	4	3	4
Cost	0.2	4	1	3	4
Total	1	3.6	3.2	3.3	3

The decision matrix above prioritizes the stability the rover will have while on the course because safety is the number one priority for Team 24. If the riders do not feel safe doing an obstacle, then they will by-pass it. Traction is the next most important metric because the wheel's main purpose is to translate rotational displacement into translation displacement and if the wheel does not have enough traction, the wheels will slip and the rover will not move; slipping is also dangerous while braking.

Metric	Weight	Rack & Pinion	Push-Pull
Ability to Fit Frame	0.20	1	3
Maneuverability	0.20	2	2
Manufacturability	0.15	1	2
Cost	0.15	1	3
Durability	0.15	2	1
Weight	0.15	1	2
Total	1	1.35	1.65

The decision matrix above prioritizes the steering assembly's ability to fit the rover frame because, if it can't be incorporated into the rover frame, it cannot be implemented into the design. Maneuverability is the next most important metric because a steering systems main purpose is to facilitate maneuverability. Next, Manufacturability, Cost, durability, and Weight were considered as secondary metrics because these metrics are not vital to the design itself.

XI.B.1.b. Concepts Considered - Details

Three-wheel design and hybrid four-wheel design were also considered. Three-wheel design offered less stability than a four-wheel design and was not recommended to repeat by the 2019 rover team; they stated that they were afraid to go over certain obstacles because of the design. Hybrid four-wheel utilizes the 2019 wheels, but due to design changes, this idea was quickly scrapped.

The standard rack and pinion was considered. However, it failed on the main metric. Team 24 felt that incorporating a steering column into the rover's design along with the single beam frame would present unnecessary design challenges.

XI.B.2. Concept/Solution for Function F2 – Support the Load of Two Riders Plus Vehicle Weight - Details

XI.B.2.a. Concept Selected - Details

Metric	Weight	4-Bar Trapezoidal	3-Bar Triangular	Single Beam
Manufacturability	0.3	1	1	3
Weight	0.25	2	1	3
Cost	0.2	1	2	3
Strength	0.2	3	2	1
Accessibility	0.1	1	2	3
Comfort	0.05	3	2	1
Total	1	1.85	1.65	2.8

A single beam frame design was selected based off several key factors. Team 24 intends on having the rover manufactured by March 16th, 2019 and the frame is expected to be one of the most, if not the most, difficult parts to manufacture. That is why the decision matrix seen above gives the frame's manufacturability the most significance. After testing and breaking LSU's 2019 rover, an emphasis on having quickly accessible parts to allow for ease of maintenance was sought. A single beam structure allows for other subsystems to be incorporated simply throughout the frame. Having this type of frame also reduces the overall weight of the rover significantly. Along with the frame design selection came the decision to have the riders sit recumbently on the frame. This allows for a low center of gravity, which will keep the riders safely on the rover. Team 24 incorporated simplified cycling biomechanics in the final design to increase efficiency for the selected riders. Further analysis and considerations for the frame's materials selection can be found in the analysis section of the report.

Metric	Weight	Spring	Spokes	I-Beam	Solid
Compressive Strength	0.3	2	2	3	4
Manufacturability	0.2	2	2	2	4
Durability	0.2	1	2	3	3
Weight	0.2	2	2	4	3
Cost	0.1	3	2	3	2
Total	1	1.9	2	3	3.4

Solid disk wheels were selected based off its compressive strength, manufacturability, and durability. Team 24 sought to tackle every obstacle on the course and the wheels must maintain

structural integrity through it all. For obstacles that the wheel will see repeated impacts, compressive strength will be important. Team 24 also plans to finish manufacturing a month before competition start so manufacturability was second most important metric. The 2019 rover used a solid sandwich structure composite wheel and created a step-by-step video tutorial. Durability was also an important metric as it needs to roll for at least the 8-minute time limit.

Metric	Weight	Double Wishbone	MacPherson Strut	Leaf Spring
Stability	0.25	3	2	2
Footprint	0.25	3	1	1
Weight	0.20	2	3	1
Cost	0.15	3	2	1
Manufacturability	0.15	2	3	1
Total	1	2.65	2.1	1.25

The decision matrix for the suspension, seen above placed an emphasis on three key metrics: stability, footprint, and weight. Team 24 sought to have the suspension provide stability throughout the rover not only for the riders' sake, but to also reduce the amount of stress seen on other components in the design. The footprint of the suspension came down to which design best fit in with other aspect of the rover. The weight of the suspension was given slightly less significance because it is still important that Team 24 reduce the overall weight, but the stability and geometric footprint were the most important design qualities sought.

XI.B.2.b. Concepts Considered - Details

4-Bar Trapezoidal and 3-Bar Triangular frames were considered in addition to the single beam frame. These types of frames proved to be more difficult to manufacture and much heavier than a single beam frame. Additionally, both the high school and college teams that placed first in the 2019 competition had single beam frames.

Spring, spokes, and I-beam cross-section wheels were considered in addition to the solid disk wheels. These wheels were difficult to manufacture using the available technology and did not provide much advantage over the solid disk wheels.

XI.B.3. Concept/Solution for Function F3 – Store Collected Samples/Task Items - Details

XI.B.3.a. Concept Selected - Details

Metric	Weight	4-Bar Trapezoidal	3-Bar Triangular	Single Beam
Manufacturability	0.3	1	1	3
Weight	0.25	2	1	3
Cost	0.2	1	2	3
Strength	0.2	3	2	1
Accessibility	0.1	1	2	3
Comfort	0.05	3	2	1
Total	1	1.85	1.65	2.8

Metric	Weight	Circular/ Sphere Design	Cap Shovel Design	3D Printed Scoop design	Vacuum Pump Design
Collection of solid and liquid sample	0.3	2	3	4	1
No Cross-Contamination	0.2	2	2	3	1
Lightweight	0.2	3	2	3	1
Fast Deployment	0.1	1	1	5	4
Easy Use	0.1	2	3	5	3
Low # of moving parts	0.05	2	3	5	1
Low # of parts	0.05	2	2	3	1
Total	1	2.8	3.6	4.2	2.2

Metric	Weight	Cross-Beam	Plunger + Digger	Plunger + Scoop	Plunger + Cap
Lightweight	0.3	3	3	3	3
Fast Extraction	0.2	5	4	2	1
Fast Containment	0.2	5	4	3	1
Fast Deployment	0.1	5	3	3	1
Easy Use	0.1	5	3	3	3
Easy Manufacturing	0.05	5	3	1	3
Affordable	0.05	5	3	1	3
Total	1	4.4	3.4	2.6	2.0

XI.B.3.b. Concepts Considered – Details

4-Bar Trapezoidal and 3-Bar Triangular frames were considered in addition to the single beam frame. These types of frames proved to be more difficult to manufacture and much heavier than a single beam frame. Additionally, both the high school and college teams that placed first in the 2019 competition had single beam frames.

For the solid and liquid soil collection task, concepts considered included circular, cap shovel, 3D printed scoop and vacuum pump. For the core sample extraction task, concepts considered included the cross-beam, plunger, scoop and cap design.

XI.B.4. Concept/Solution for Function F4 – Convert Rider Input (Pedaling) Into Useful Output - Details

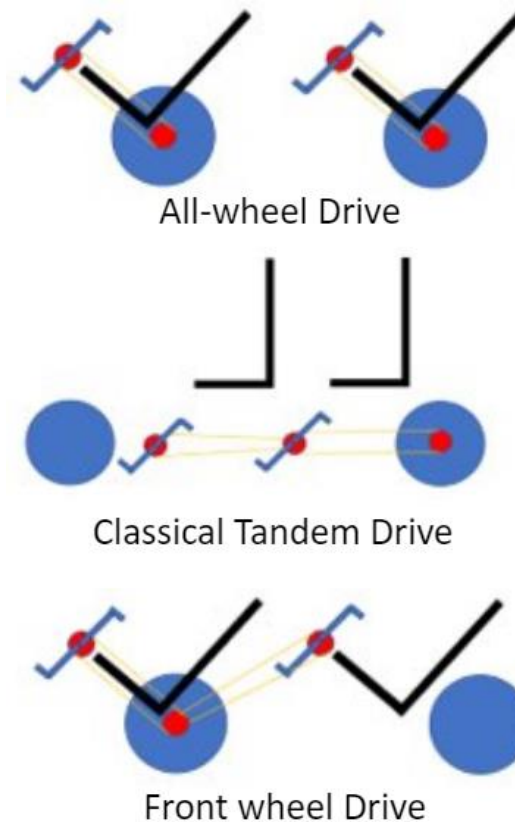
XI.B.4.a. Concept Selected - Details

Metric	Weight	All-Wheel	Tandem	Front-Wheel
Available Output Power	0.30	3	3	3
Simplicity	0.30	2	3	1
Manufacturability	0.20	2	2	2
Cost	0.10	1	2	2
Weight	0.10	1	2	2
Total	1	2.1	2.6	2.0

The concept selected in order to convert rider input into useful output was a combination of the tandem and front-wheel concepts. The team made this decision because the tandem concept guaranteed additive power to a single drivetrain, and the front-wheel concept allowed

free rover real estate towards the rear in order to leave room for the hinge mechanism used to complete the optional NASA-implemented volume constraint of 125ft³. Available output power and simplicity were the heaviest weighted metrics for the drivetrain decision matrix. Available output power was rated as shown because the primary function of any drivetrain is to transmit power through a system effectively. Simplicity was weighted heaviest because one of the team's top qualitative constraints was to keep the design as simple as possible. Manufacturability was rated with a 0.20 weight because the team was willing to sacrifice manufacturability in order to guarantee simplicity and functionality. However, manufacturability was more important than cost and weight of the drivetrain. Both cost and weight were weighted the lowest because of the high budget allocation for the drivetrain and because the weight is constrained to be the weight of commercially-available parts.

XI.B.4.b. Concepts Considered – Details



XI.B.5. Concept/Solution for Function F5 - Maintain Durability Throughout the Competition - Details

XI.B.5.a. Concept Selected - Details

One of the main considerations for selection of materials beyond the decision matrix was the ease of procurement. It was important to consider materials that could be purchased and delivered as soon as possible to be able to manufacture the rover by the team-imposed deadline.

XI.B.5.b. Concepts Considered - Details

Material Selection				
Metric	Weight	Carbon Fiber	Aluminum	Steel
Manufacturability	0.3	1	5	5
Weight	0.3	5	4	2
Strength	0.2	5	4	4
Weldability	0.1	1	5	5
Cost	0.1	3	4	5
Total	1	3.2	4.4	3.9

XI.B.6. Concept/Solution for Function F6 - Easy and Quick Assembly - Details

XI.B.6.a. Concept Selected - Details



Figure XI.B.6.1: Seat Quick Assembly Features

XI.B.6.b. Concepts Considered – Details



Figure XI.B.6.2: Seat Stays (Volae, 2019)

XI.C. System Description/Product Architecture Supplement

XI.C.1. Sub-System SS#1- Solid Disk Wheel

The function of the solid disk wheels is to convert the rotational displacement of the driveshaft into translational displacement of the overall rover in order to traverse a demanding obstacle course. Utilizing a solid disk design ensures that the wheels can handle the load of the two riders and the rover weight. By incorporating a curved rubber tread, the wheels will have traction on different terrains and mitigate some impact forces seen by the wheel. Using a sandwich structure composite, the wheels will be relatively lightweight and allow the riders to accelerate the rover with relative ease.

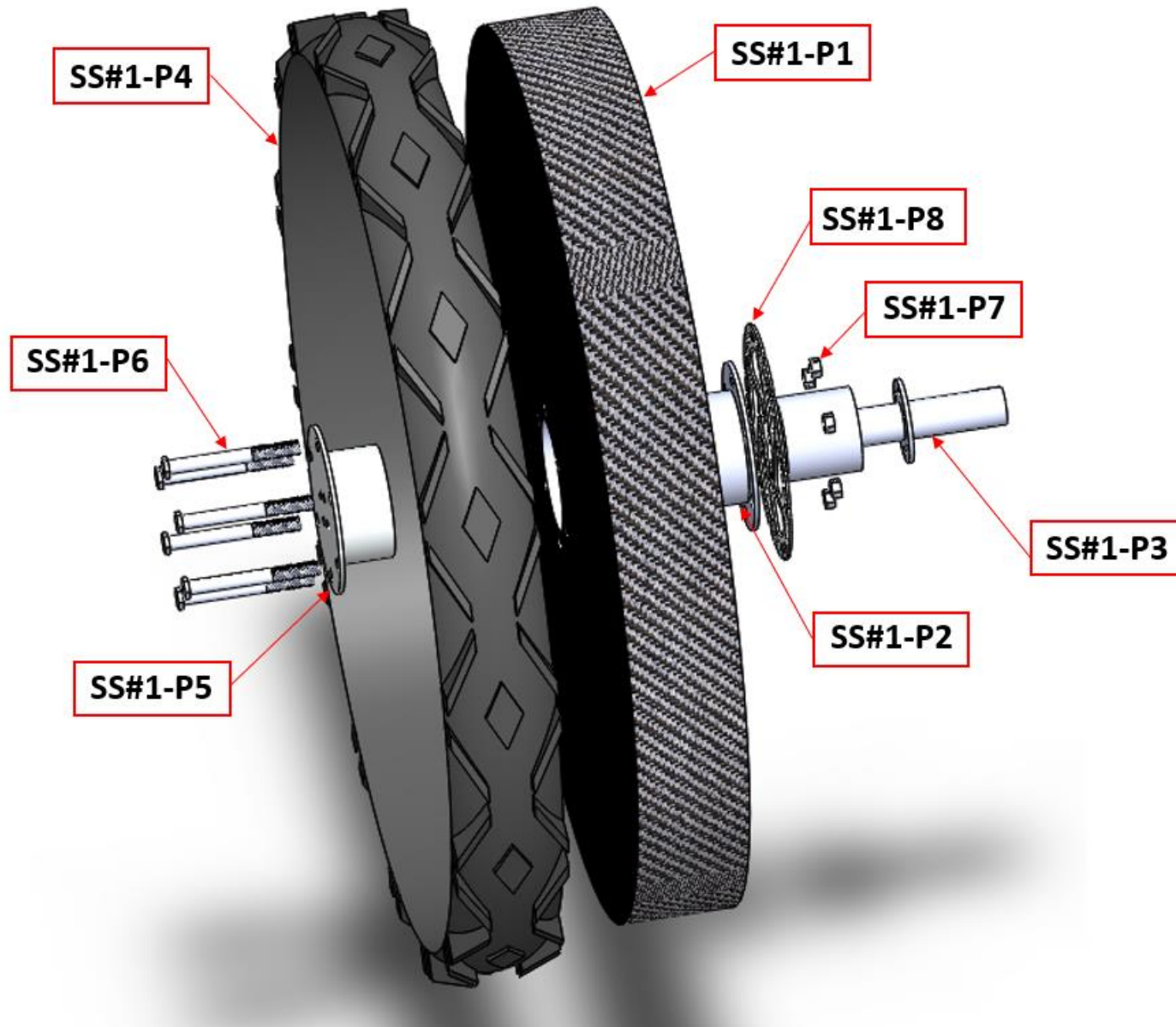


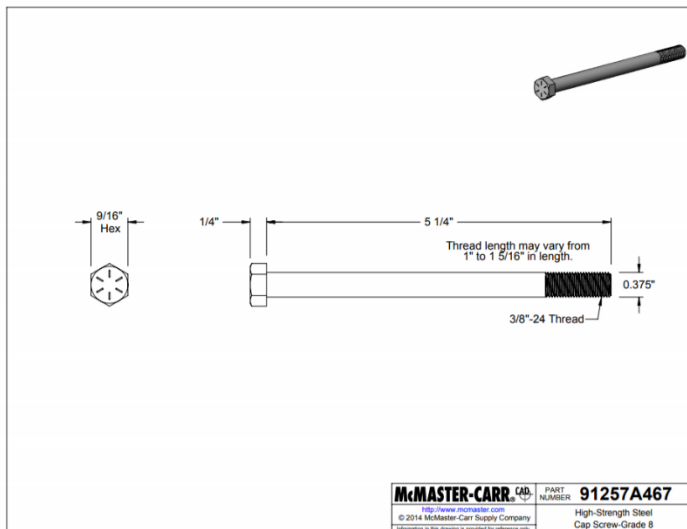
Figure XI.C.1: Exploded View Assembly Drawing of Sub-System SS#1 - Solid Disk Wheels

XI.C.1.a. Comprehensive Parts List for SS#1 – Solid Wheels

Table XI-1: List of Parts for Sub-System SS#1 – Solid Wheels

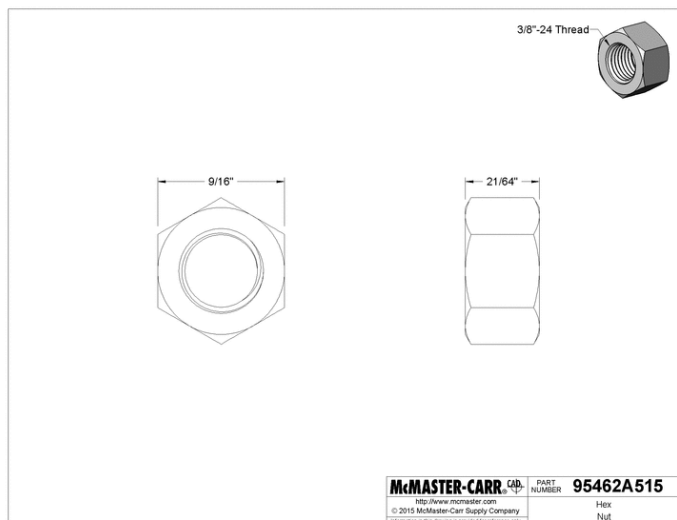
Part #	Quantity	Material	Name
SS#1-P1	4	Balsa & Carbon Fiber	Wheel
SS#1-P2	4	Al 6061-T6	Inner Wheel Hub
SS#1-P3	4	Al 6061-T6	Wheel Hub Connector
SS#1-P4	4	PMC 770	Tread
SS#1-P5	4	Al 6061-T6	Outer Wheel Hub
SS#1-P6	40	N/A	3/8 in. Bolt
SS#1-P7	24	N/A	3/8 in. Nut
SS#1-P8	2	N/A	Braking Disc

XI.C.1.b. Off-The-Shelf Parts and Component Specifications for SS#1 - Solid Disk Wheels



Head Type	Hex
Drive Style	External Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	3/8"-24
Screw Size Decimal	0.375"
Equivalent	
Thread Type	UNF
Thread Fit	Class 2A
Length	5 1/4"
Threading	Partially Threaded
Min. Thread Length	1"
Thread Spacing	Fine
Head	
Width	9/16"
Height	1/4"
Fastener Strength	Grade 8
Grade/Class	
Material	Steel
Finish	Zinc Yellow-Chromate Plated
Tensile Strength	150,000 psi
Hardness	Rockwell C33
Specifications Met	ASME B18.2.1, SAE J429
RoHS	RoHS 3 (2015/863/EU) noncompliant

Figure XI.C.2: Specifications for part SS#1-P6 - 3/8"-24 Thread Size, 5 1/4" Long Hex Head Screw [21]



Material	Steel
Fastener Strength	Grade 8
Grade/Class	
Thread Size	3/8"-24
Thread Type	UNF
Thread Spacing	Fine
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	9/16"
Height	21/64"
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Standard
System of Measurement	Inch
RoHS	RoHS 3 (2015/863/EU) compliant

Figure XI.C.3: Specifications for part SS#1-P7 - 3/8" Thread Size Hex Nut [22]



SHIMANO SLX RT66 6-BOLT ROTOR

The Shimano SLX RT66 6-BOLT Rotor features a lightweight and sleek design that is super reliable.

SHIMANO SLX RT66 6-BOLT ROTOR SPECIFICATIONS

- Hub/Disc Compatibility: 6-Bolt Disc
- Pad Compatibility: All Pad Compounds
- Rotor Size (mm): 203mm, 160mm, 180mm
- Rotor Type: One Piece
- Includes rotor fixing bolts and non-turn washers

Figure XI.C.4: Specifications for part SS#1-P8 – Braking Disc [23]

XI.C.1.c. Engineering (Manufacturing) Drawings of All Parts of SS#1 - Solid Disk Wheels

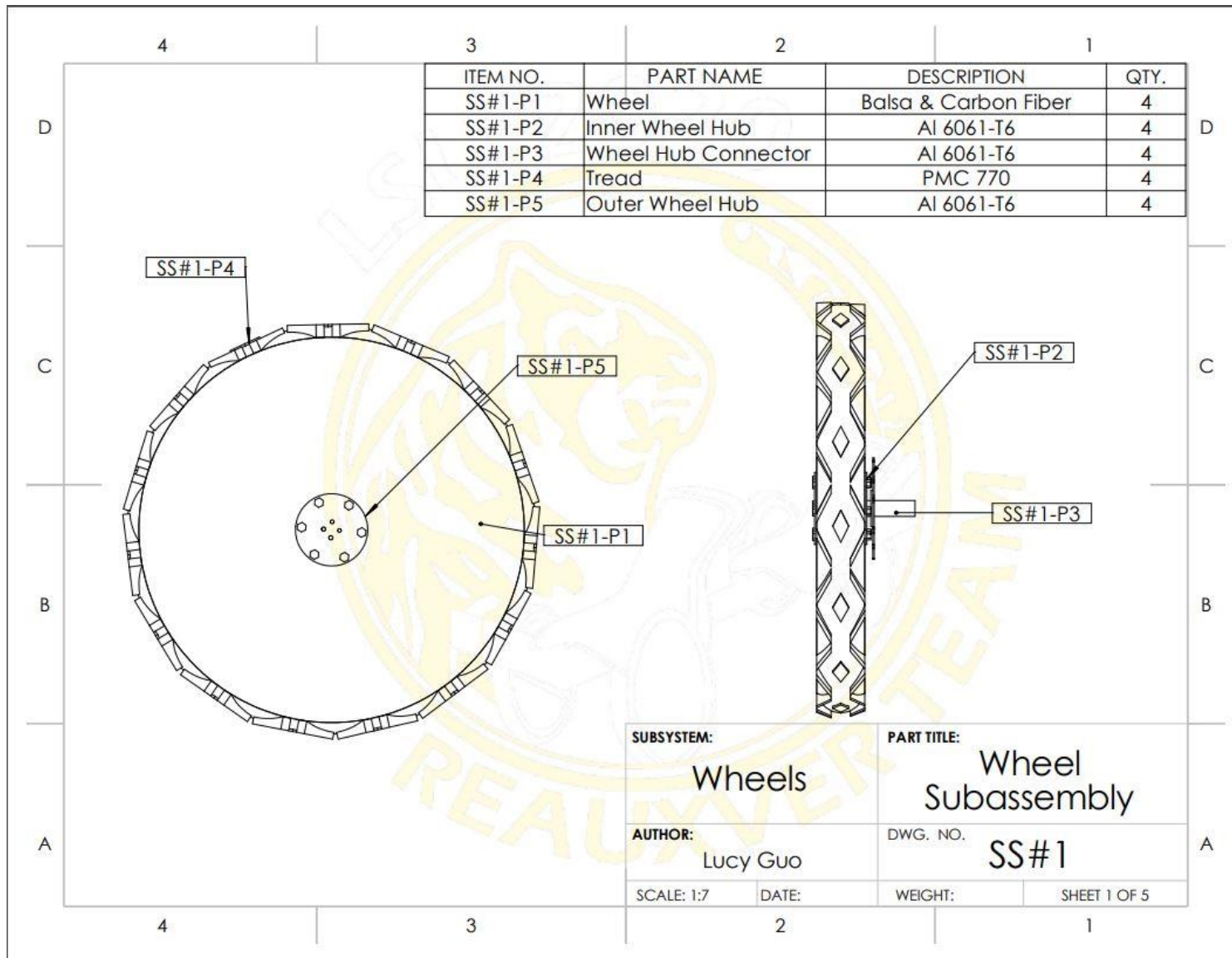


Figure XI.C.5: Manufacturing Drawing of part SS#1 - Wheel

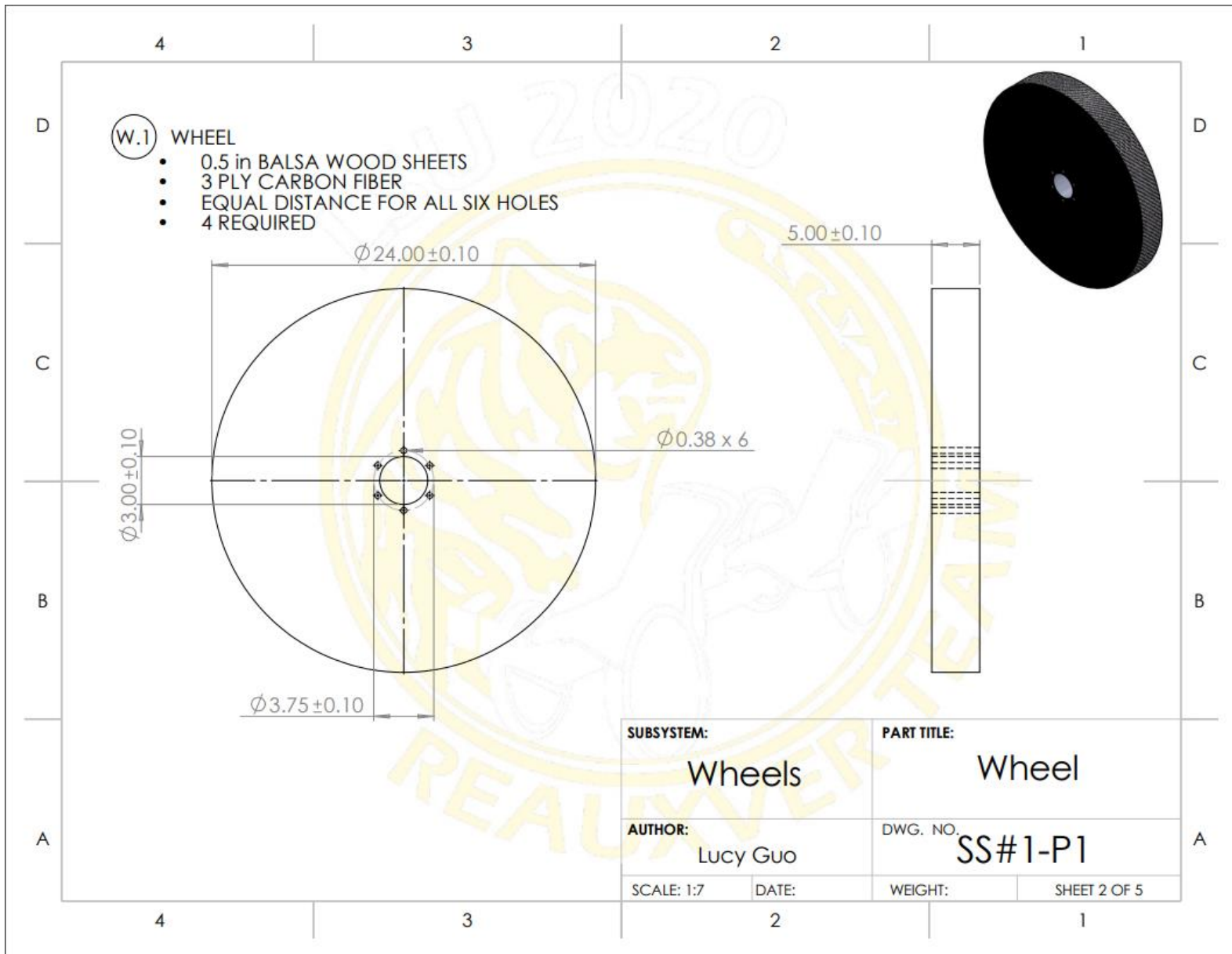


Figure XI.C.6: Manufacturing Drawing of part SS#1-P1 - Wheel

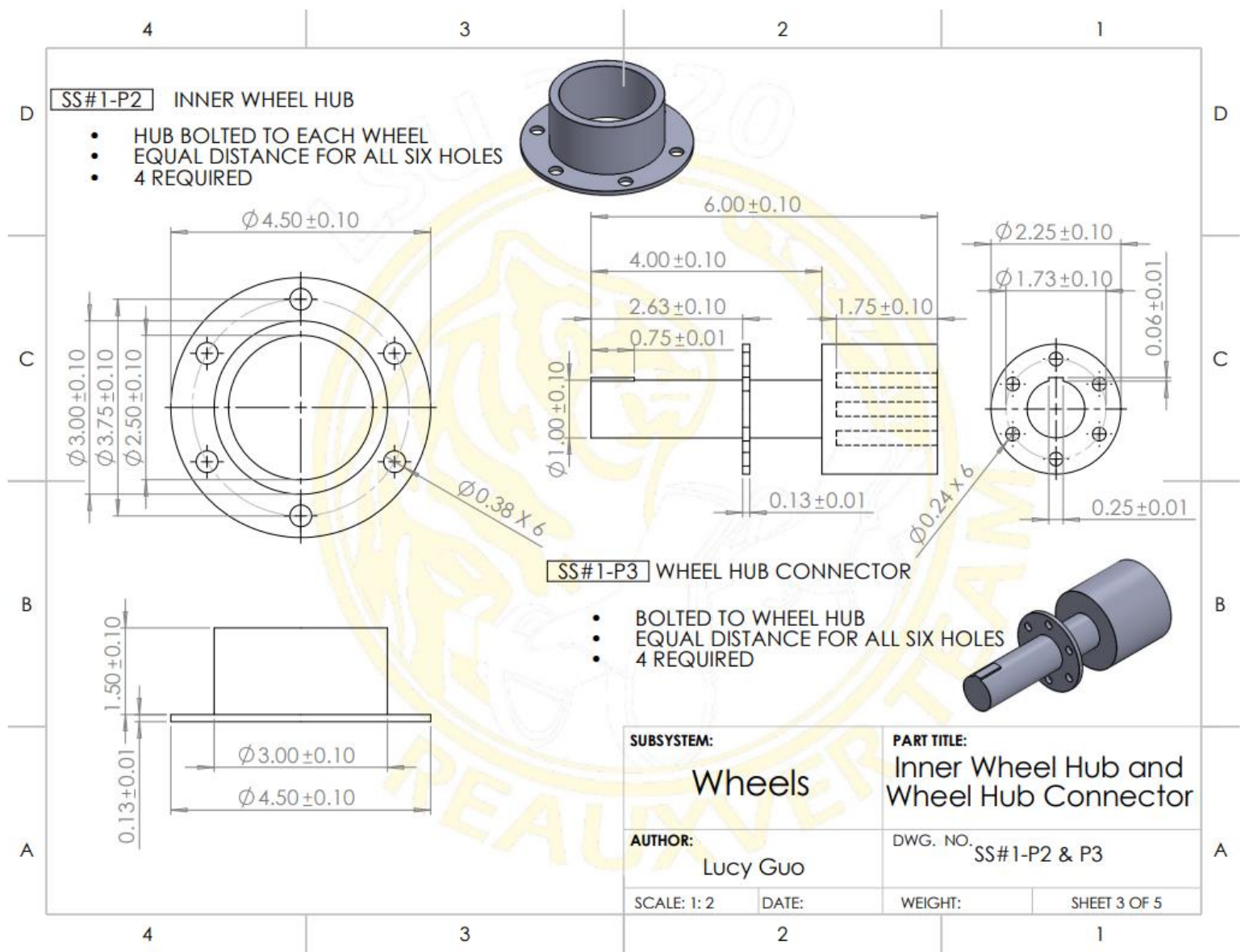


Figure XI.C.7: Manufacturing Drawing of part SS#1-P2&P3 – Inner Wheel Hub and Wheel Hub Connector

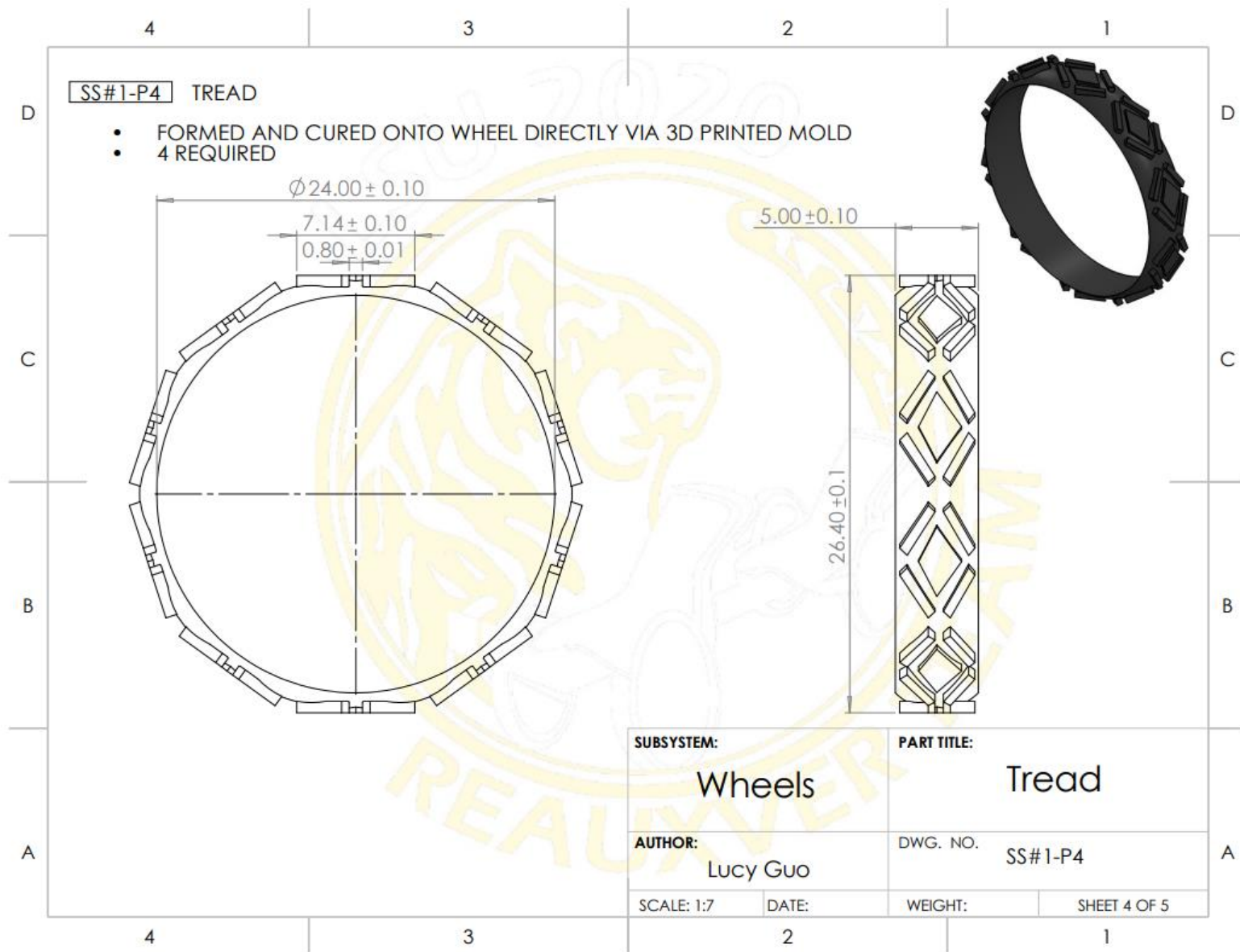


Figure XI.C.8: Manufacturing Drawing of part SS#1-P4 – Tread

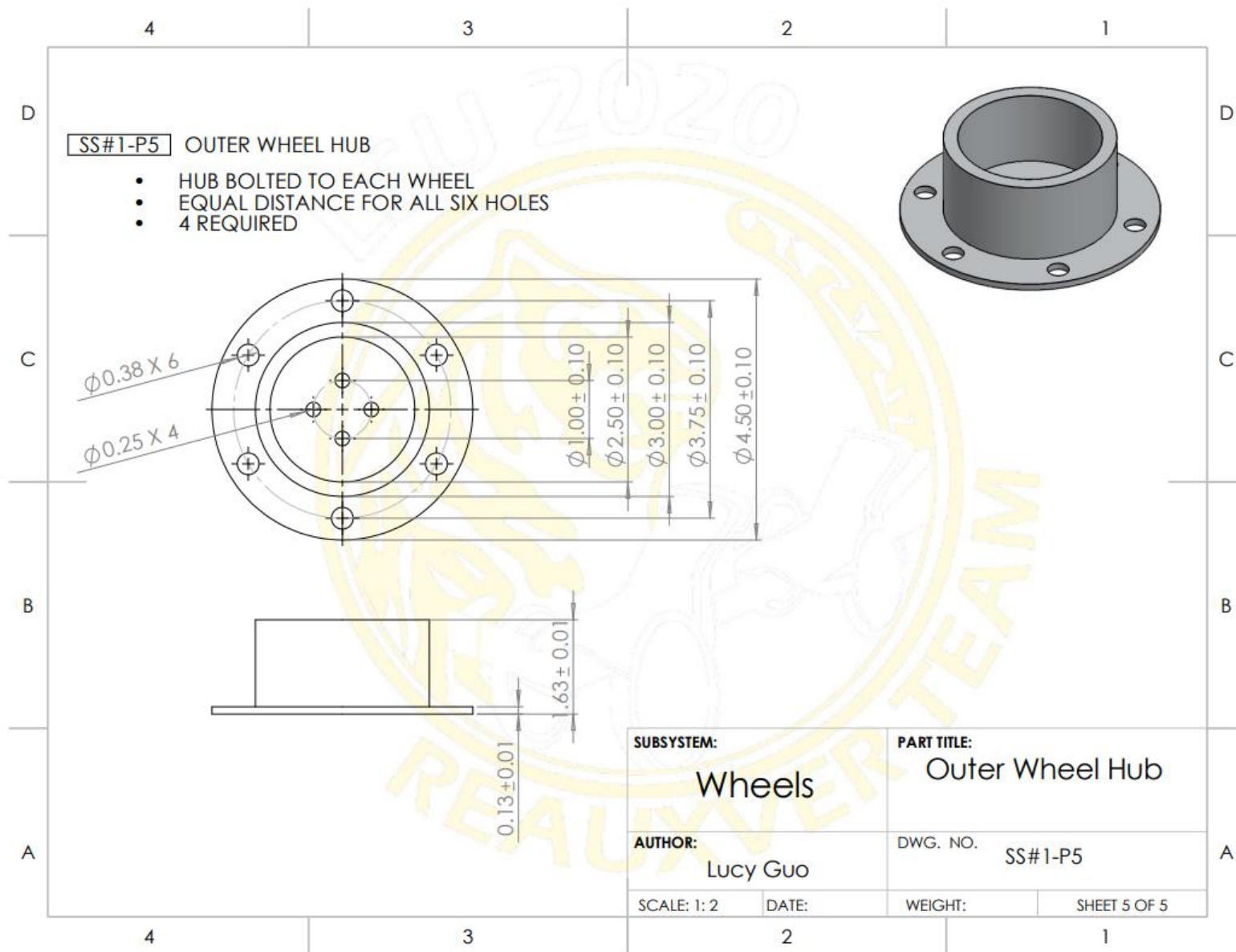


Figure XI.C.9: Manufacturing Drawing of part SS#1-P5 – Outer Wheel Hub

XI.C.2.Sub-System SS#2- Single Bar Frame

The functions of the single bar frame are to support the load of two riders plus the vehicle weight, store collected samples/task items, maintain durability throughout competition, and be easy and quick to assemble. The single bar frame accomplishes all required functions while reducing overall weight of the rover. This sub-system ensures the rover will withstand static and dynamic forces while having a long fatigue life.

Not Shown:

SS#2-P13

SS#2-P14

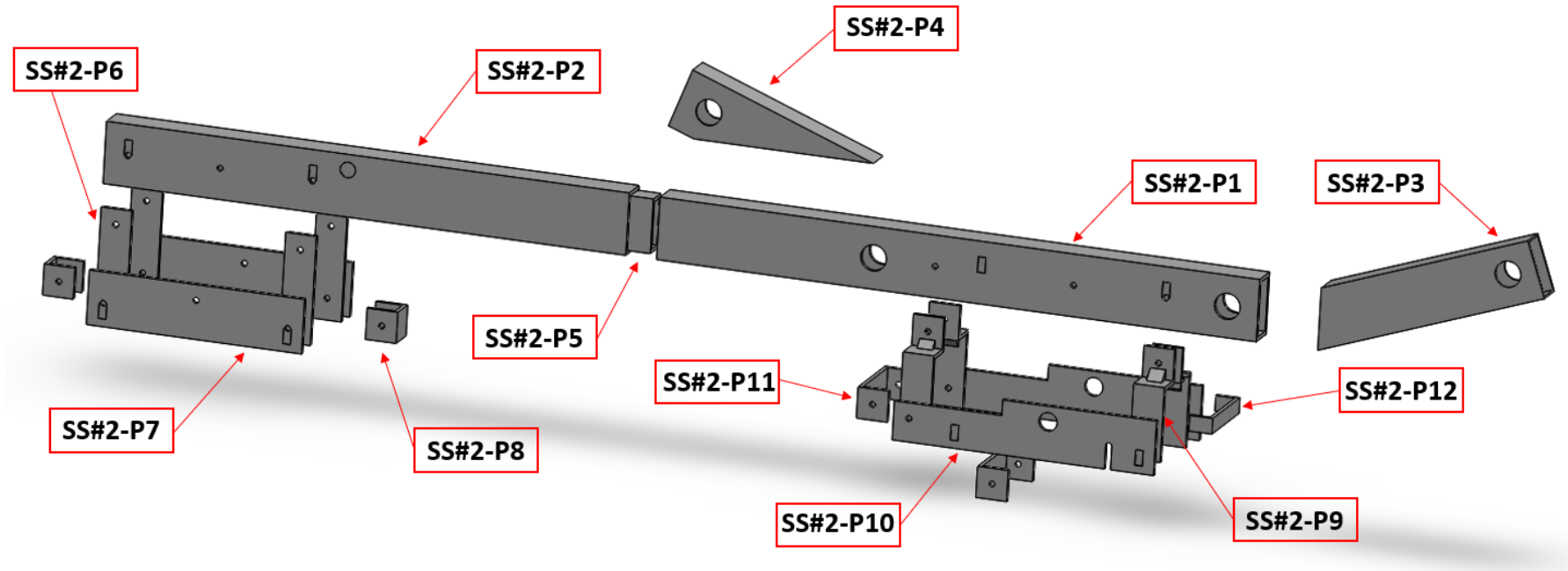


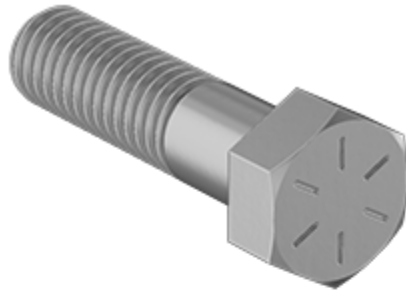
Figure XI.C.10: Exploded View Assembly Drawing of Sub-System SS#2 - Single Bar Frame

XI.C.2.a. Comprehensive Parts List for SS#2- Single Bar Frame

Table XI-2: List of Parts for Sub-System SS#2 – Single Bar Frame

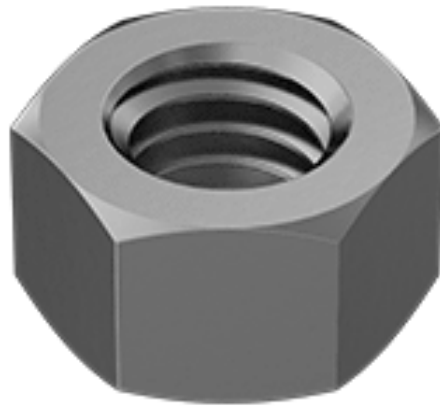
Part #	Quantity	Material	Name
SS#2-P1	1	Al 6061-T6	Front Frame Section
SS#2-P2	1	Al 6061-T6	Rear Frame Section
SS#2-P3	1	Al 6061-T6	Front Crankset Support
SS#2-P4	1	Al 6061-T6	Rear Crankset Support
SS#2-P5	1	Al 6061-T6	Frame Connection Piece
SS#2-P6	4	Al 6061-T6	Rear Bracketing Assembly Legs
SS#2-P7	2	Al 6061-T6	Rear Bracketing Assembly Tie-in
SS#2-P8	2	Al 6061-T6	Rear Bracketing Assembly Supports
SS#2-P9	4	Al 6061-T6	Drivetrain Bracketing Assembly Legs
SS#2-P10	2	Al 6061-T6	Drivetrain Bracketing Assembly Tie-in
SS#2-P11	2	Al 6061-T6	Drivetrain Bracketing Assembly Supports – Large
SS#2-P12	1	Al 6061-T6	Drivetrain Bracketing Assembly Supports – Small
SS#2-P13	8	Grade 8 Steel	3/8" - 16 Thread Size, 5" L, Fully Threaded Hex Head Bolts
SS#2-P14	8	Grade 8 Steel	3/8" Hex Nut

XI.C.2.b. Off-The-Shelf Parts and Component Specifications for SS#2 – Single Bar Frame



Head Type	Hex
Drive Style	External Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	1/4"-20
Screw Size Decimal Equivalent	0.25"
Thread Type	UNC
Thread Fit	Class 2A
Length	5"
Threading	Partially Threaded
Min. Thread Length	3/4"
Thread Spacing	Coarse
Head	
Width	7/16"
Height	11/64"
Fastener Strength	
Grade/Class	Grade 8
Material	Steel
Finish	Zinc Yellow-Chromate Plated
Tensile Strength	150,000 psi
Hardness	Rockwell C33
Specifications Met	ASME B18.2.1, SAE J429
RoHS	RoHS 3 (2015/863/EU) compliant

Figure XI.C.11: Specifications for part SS#2-P13 - 3/8" - 16 Thread Size, 5" L, Fully Threaded Hex Head Bolts



Material	Zinc Yellow-Chromate Plated Steel
Fastener Strength	Grade 8
Grade/Class	
Thread Size	3/8"-16
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	9/16"
Height	21/64"
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Standard
System of Measurement	Inch
RoHS	RoHS 3 (2015/863/EU) noncompliant

Figure XI.C.12: Specifications for part SS#2-P13 - 3/8" - 16 Thread Size, 5" L, Fully Threaded Hex Head Bolts

XI.C.2.c. Engineering (Manufacturing) Drawings of All Parts of SS#2- Single Bar Frame

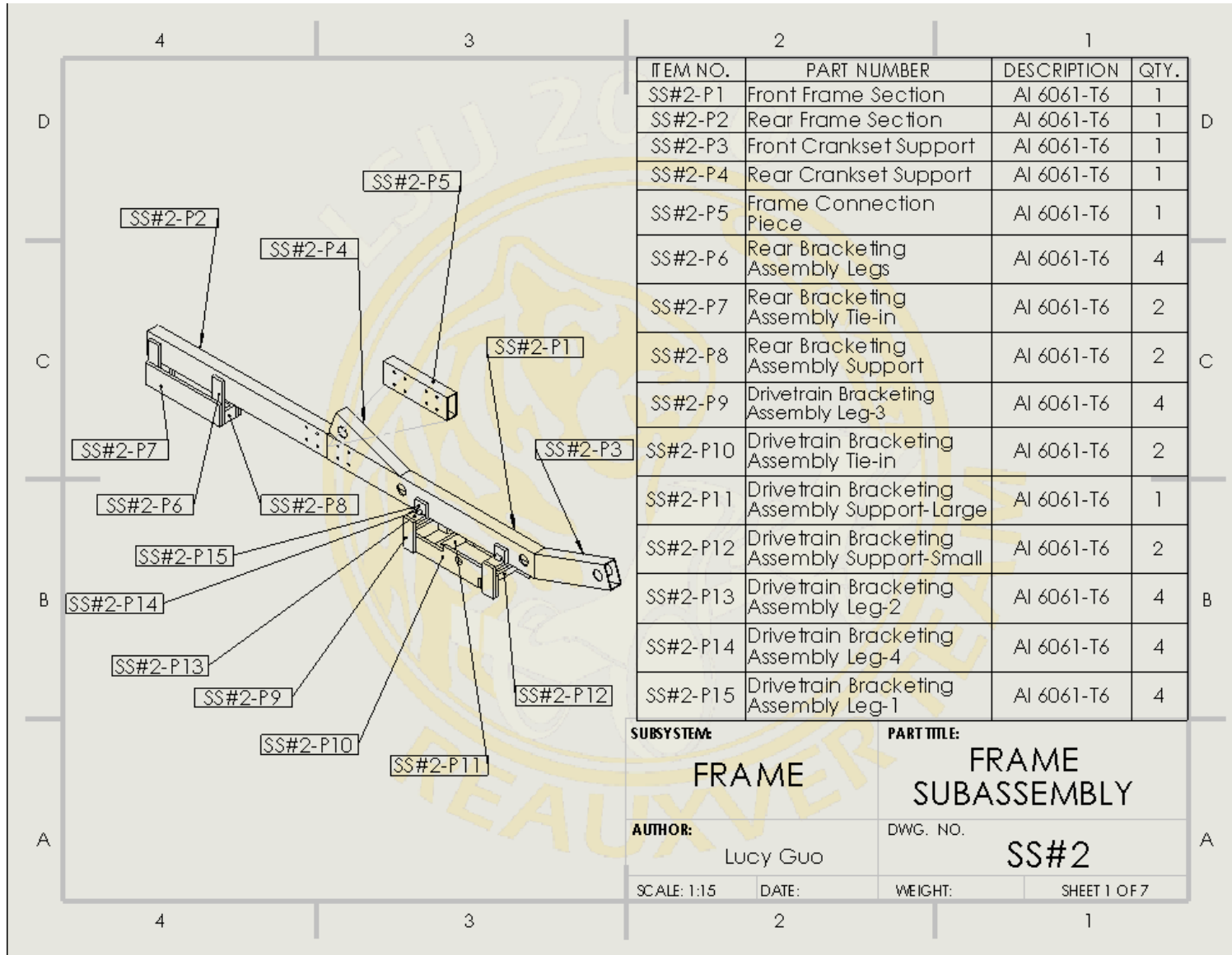


Figure XI.C.13: Manufacturing Drawing of part SS#2 – Frame Subassembly

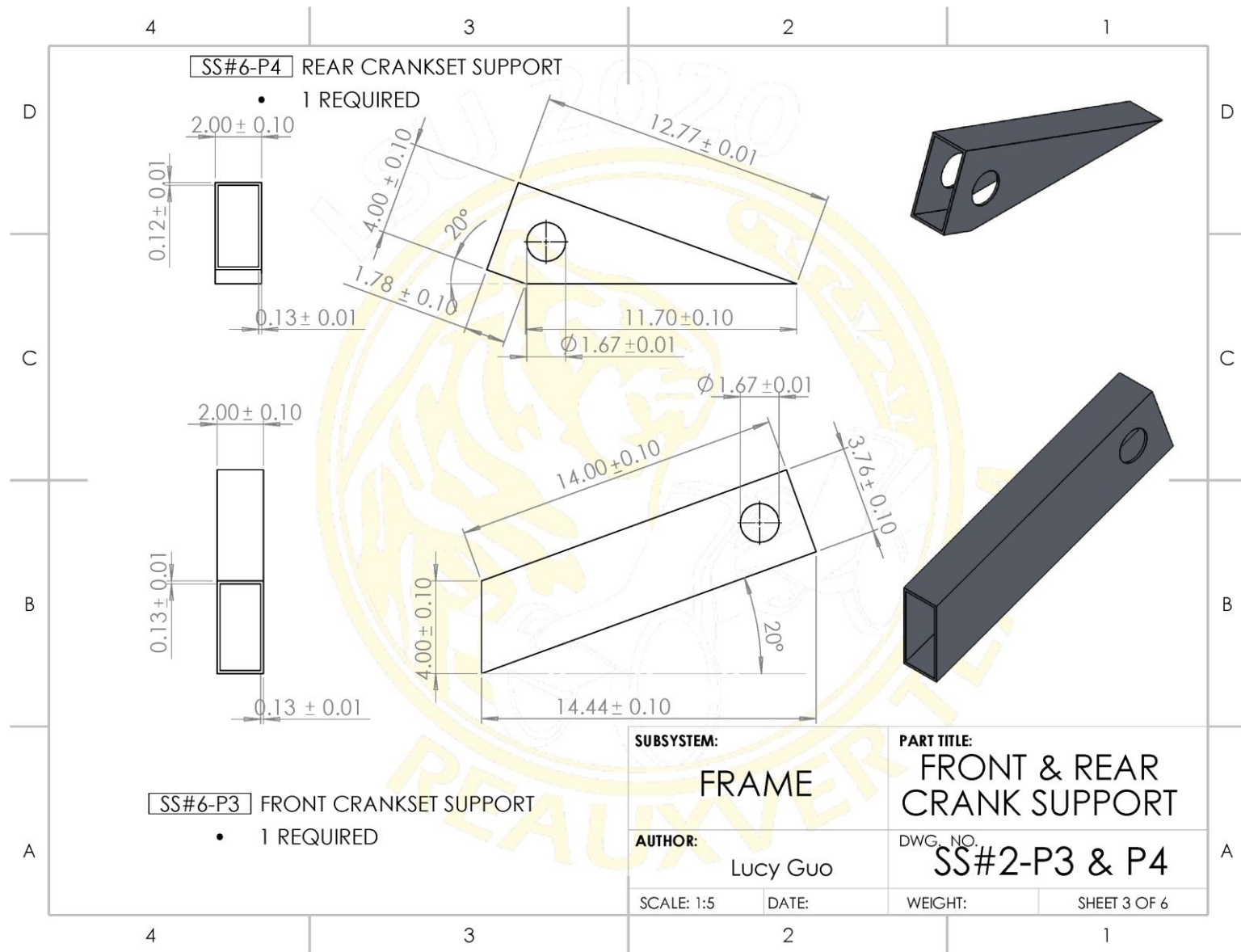


Figure XI.C.15: Manufacturing Drawing of part SS#2-P3&P4 – Front and Rear Crank Support

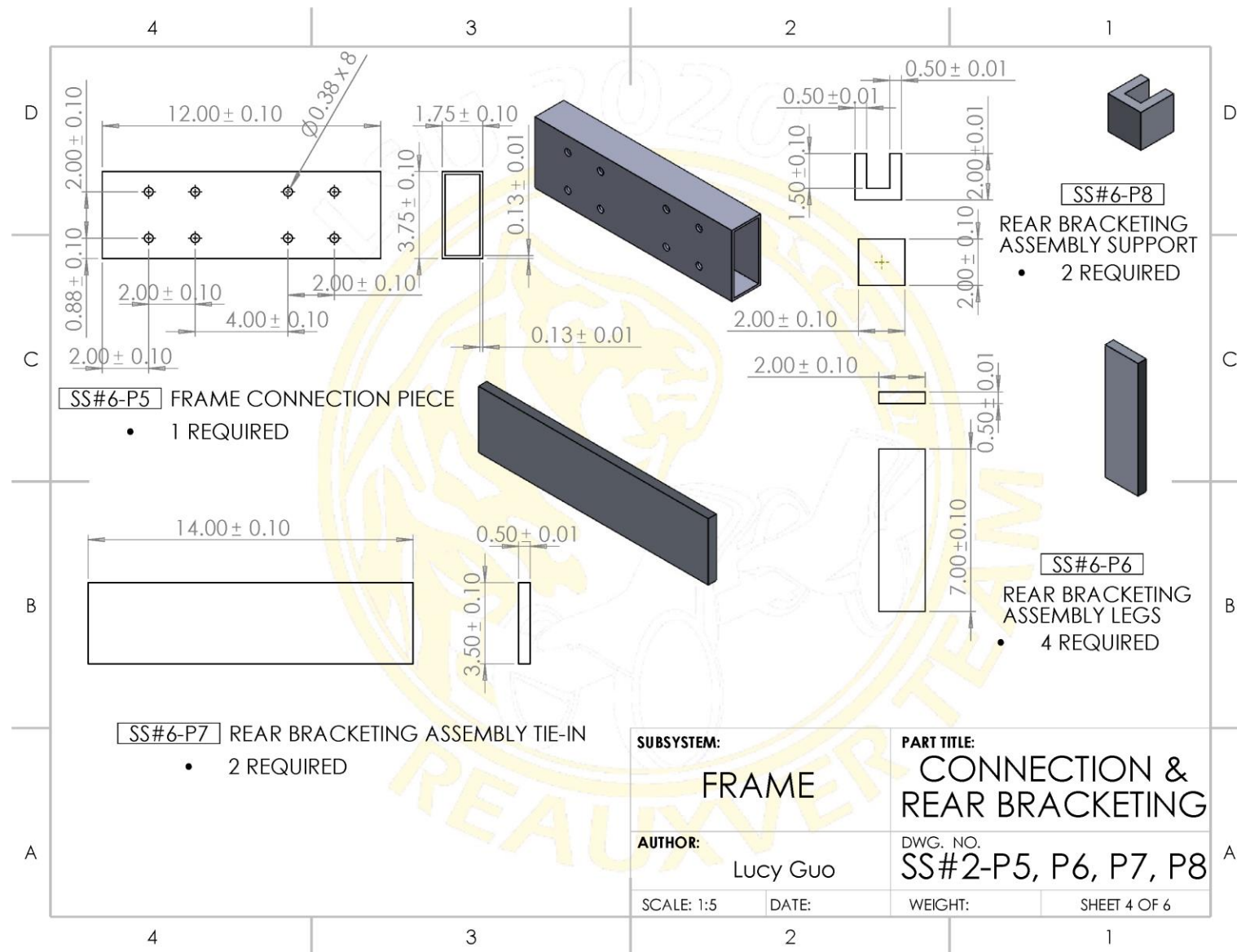


Figure XI.C.16: Manufacturing Drawing of part SS#2-P5,6,7,8 – Connection and Rear Bracketing

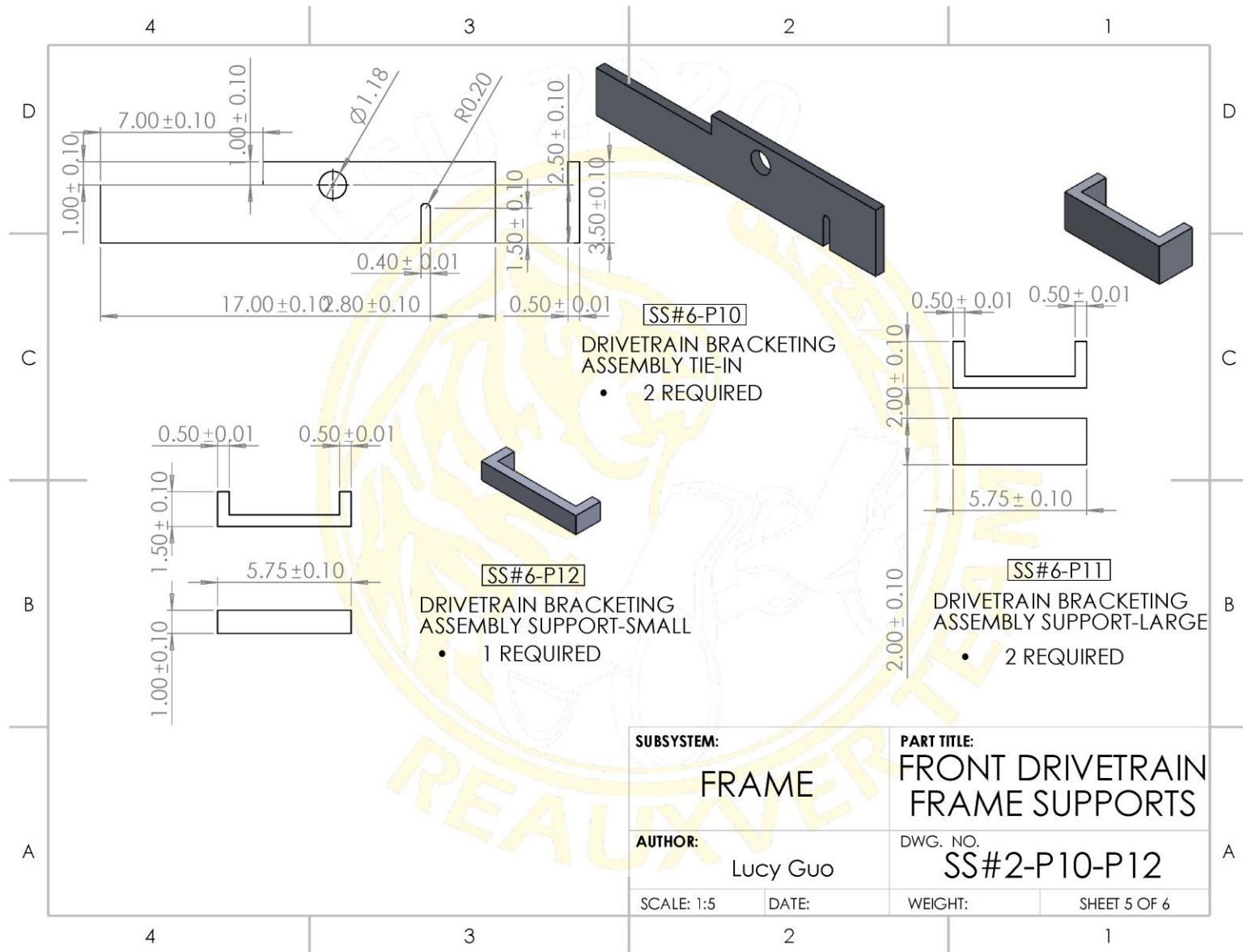


Figure XI.C.17: Manufacturing Drawing of part SS#2-P10,P11,P12 – Front Drivetrain Frame Supports

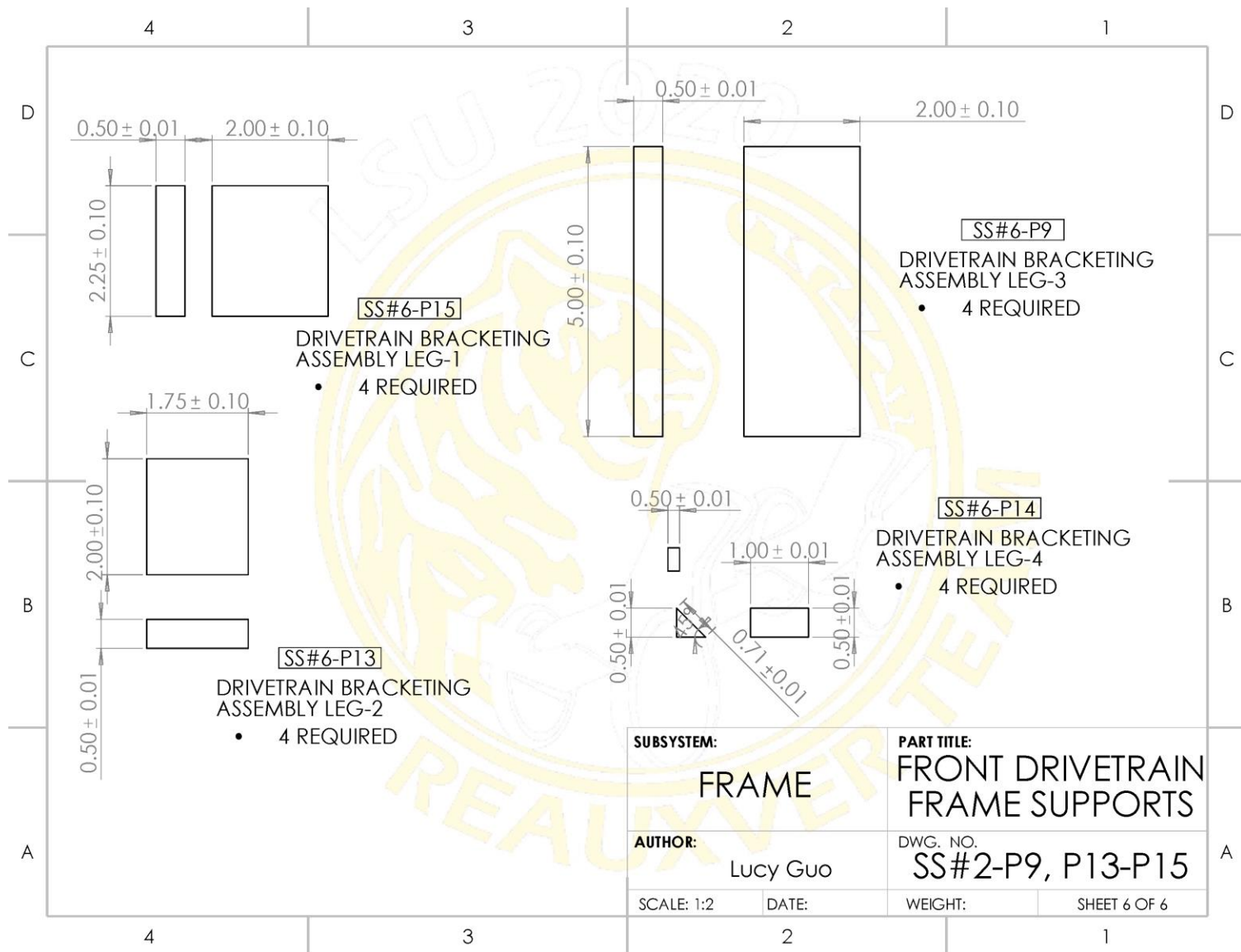


Figure XI.C.18: Manufacturing Drawing of part SS#2-P9,P13,P14,P15 – Front Drivetrain Frame Supports

XI.C.3.Sub-System SS#3- Front Tandem Drivetrain

The function of the front tandem drivetrain is to convert driver input power into useful rover output power to traverse a demanding obstacle course. Utilizing a classical tandem drive design ensures additive power from both drivers. By Incorporating a front-wheel drive system into the classical tandem design, the drivetrain will have a smaller footprint. This will allow for the opportunity to incorporate a folding mechanism into the rear portion of the frame, which was otherwise unachievable by solely utilizing the classical tandem drive design.

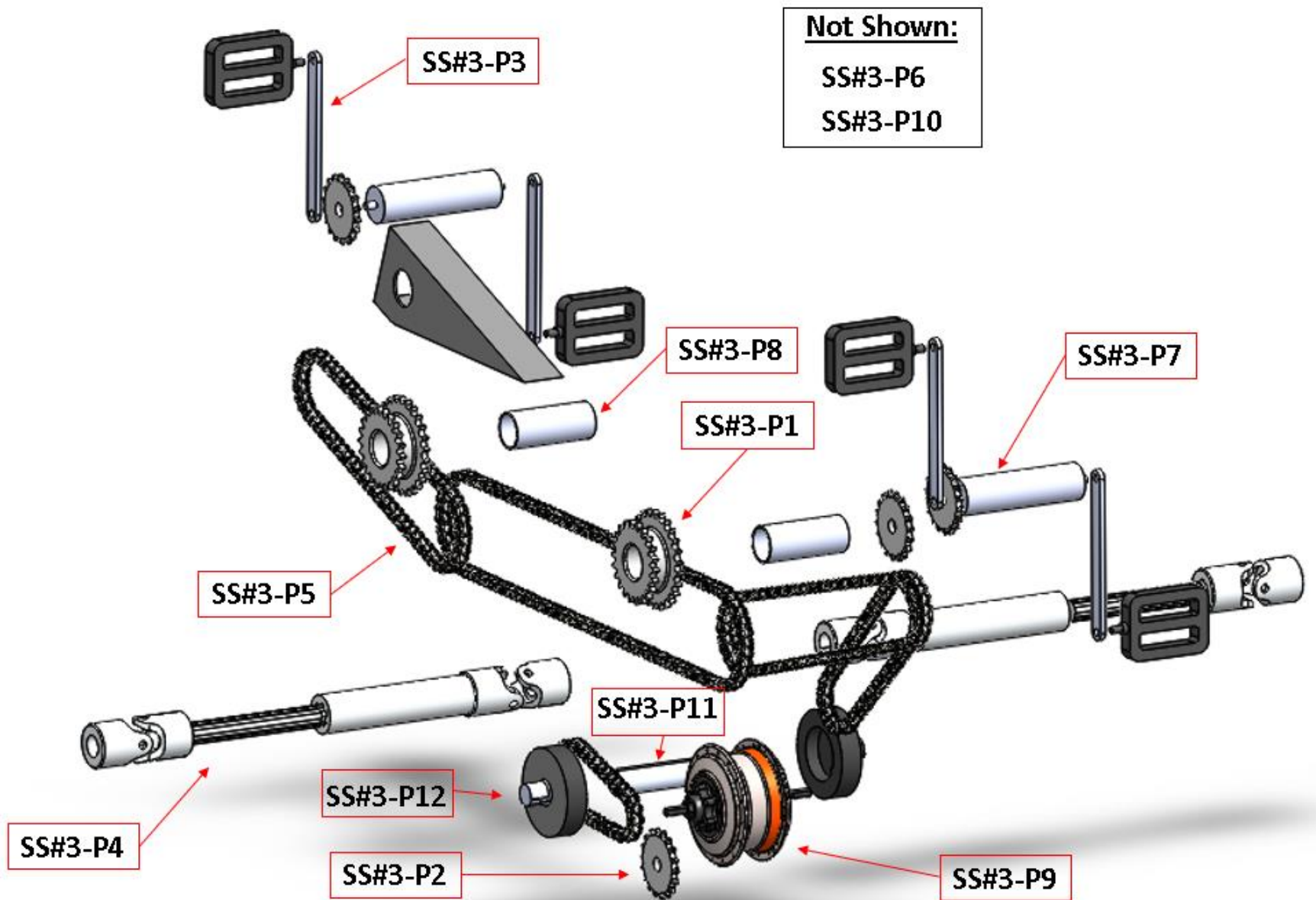


Figure XI.C.19: Exploded View Assembly Drawing of Sub-System SS#3- Front Tandem Drivetrain

XI.C.3.a. Comprehensive Parts List for SS#3- Front Tandem Drivetrain

Table XI-3: List of Parts for Sub-System SS#3 – Front Tandem Drivetrain

Part #	Quantity	Material	Name
SS#3-P1	7	Steel	Surly 36T Cog
SS#3-P2	2	Steel	Surly 18T Cog
SS#3-P3	4	Steel	7" Crankset
SS#3-P4	2	Steel	Telescopic U-Joint
SS#3-P5	5	Steel	ASTM #40 Standard Roller Chain
SS#3-P6	2	Steel	Hub Anchor
SS#3-P7	4	Steel	Square-Tapered Bottom Bracket
SS#3-P8	4	Aluminum	Bottom Bracket Shells
SS#3-P9	1	N/A	NuVinci N380 Internally-Geared Hub
SS#3-P10	3	Aluminum	Surly Singleator Chain Tensioner
SS#3-P11	1	Al 6061 T6	Driveshaft
SS#3-P12	2	Steel	206 Series Pillow-Block Bearing



[7]

SURLY STAINLESS STEEL CHAINRING

Surly's Stainless Steel Chainrings are perfect for singlespeeds, fixed gears or riders simply wanting a stronger, stiffer, prettier chainring. Although they are slightly heavier than alloy chainrings, they are considerably more resilient to impact and will last over twice as long as alloy chainrings. Also, once worn in one direction, their symmetrical design allows them to be flipped for double the lifespan, making them more economical than alloy rings! These non-ramped, non-pinned chainrings have a beautiful stainless finish that will upgrade the class of any bike. These are compatible with most standard bolt circle diameter cranksets, and can be used just fine with multiple front ring setups, although shifting will be slightly slower than with a ramped chainring. These fantastic rings come in a variety of bolt patterns, teeth options, and BCD. Check the options menu for the chainring that will fit your application.

FEATURES AND INFORMATION

- Incredibly strong and durable!
- No ramps or shift gates
- For use with 6-10 speed chains, 1/2"x3/32" pitch
- Use of Surly rings with 9 and 10 speed chains will have a tight fit and require a short break in period
- Made from 304 stainless

Figure XI.C.20: Specifications for part SS#3-P1,2 – Surly 36T/18T Cog



[8]

SHIMANO ACERA 7/8 SPEED CRANKSET

Shimano Acera 7/8 Speed Crankset features many of Shimano's features that can also be seen on higher end components. The patented Hyperdrive system will provide quiet, easy and trouble free shifting. Available in two arm lengths and two gearing options. Choose the 22/32/42 option for standard mountain bike use and the 28/38/48 for taller road like/touring gears. Also available with a chain guard.

SPECIFICATIONS

- Rings: 22/32/42 or 28/38/48
- Spindle Interface Type: Square Taper JIS
- Spindle Length: 122.5 mm
- Crank Arm Length: 170mm or 175mm
- Chainring BCD: 104/64
- Bolt Pattern: 4-Bolt
- Chain Compatibility: 7/8-Speed
- Pedal Spindle: 9/16"
- Color: Black
- Crank Intended Use: Mountain

Figure XI.C.21: Specifications for part SS#3-P3 – 7" Crankset



[9]

SRAM PC-1 Single Speed chains. An excellent choice for all singlespeed and track bicycle applications, including older 1 speed bikes like beach cruisers, etc. as well as today's high-end singlespeed MTBs.

- 332 grams
- 114 links
- For 1/8" single speed specific cogs and chainrings only. Do NOT use on drivetrains designed for use with multi-speed 3/32" chains.

Figure XI.C.22: Specifications for part SS#3-P5 – ASTM #40 Standard Roller Chain



[10]

SHIMANO UN55 BOTTOM BRACKET

High quality Shimano sealed bottom bracket (square taper). Extra strong Aluminum (not plastic) non drive cup. Please choose width and spindle length.

- English Threaded

Specifications

Bottom Bracket Shell Width	68mm, 70mm, 73mm
Bottom Bracket/Frame Interface	English, Italian
Spindle Interface Type	Square Taper - JIS
Spindle Length (mm)	107, 110, 113, 115, 118, 122.5, 127

Figure XI.C.23: Specifications for part SS#3-P7 – Square-Tapered Bottom Bracket



[11]

Nfinity 380 / Nfinity 380se	
Hub	N380/N380se
Hub Colors	Silver or black
CVP weight	2.450 grams
Speed ratio	Infinitely variable within ratio range
Ratio Range	380% nominal, 0.5 underdrive to 1.9 overdrive
Controller	C8
Other compatible controller	C3, C8s and N360 shifter (all MY)
Shifting	Variable twist grip, ¾ turn
Hub Interface	Nfinity Hub Interface 380
Other compatible HI's	-
Spoke Configurations	32 or 36 hole
Brake Configurations	Disc, rim, or roller
CVP weight	2450 grams
Drop-Out Width	135mm, horizontal or vertical
Axle Length	183mm
Axle Threads	M10 x 1
Chainline	49.0 ± 0.5mm
Beltline (Gates)	45.5 ± 0.5 mm
Beltline (Conti-Tech)	46.2 ± 0.5 mm
Sprocket Sizes	16-20 tooth (chain)
	20-28 Tooth (belt)
Sprocket type	Standard 9-spline, flat on one side
Sprocket ratio definition	Front sprocket teeth / Rear sprocket teeth
Sprocket ratio minimum (Pedal bikes)	1.8 (Table in section 5.2)
Sprocket ratio minimum (Front motor e-Bikes)	1.8 (Table in section 5.2)
Sprocket ratio minimum (Mid motor e-Bikes)	2.0 (Table in section 5.2)
Sprocket ratio minimum (Bosch GEN2 eBikes)	0.8 (Table in section 5.2)
Spoke flange diameter	125.0 mm
Center to left spoke flange	26.0 mm
Center to right spoke flange	26.0 mm

Figure XI.C.24: Specifications for part SS#3-P9 – NuVinci N380 Internally-Geared Hub



[12]

- Tensioner allows use of 1-speed drivetrain on vertical drop frame
- Mounts to derailleur hanger, self aligning floating pulley
- Adjustable chain line
- anti-chain drop hoop
- This part requires an 18mm Cone wrench to make it work. See the the Surly wrench below. Its the perfect fit. If you are self-installing, make sure to have one on hand. Make sure to read the instructions.
- Weight: 157 g

Figure XI.C.25: Specifications for part SS#3-P10 – Surly Singleator Chain Tensioner

XI.C.3.c. Engineering (Manufacturing) Drawings of All Parts of SS#3- Front Tandem Drivetrain

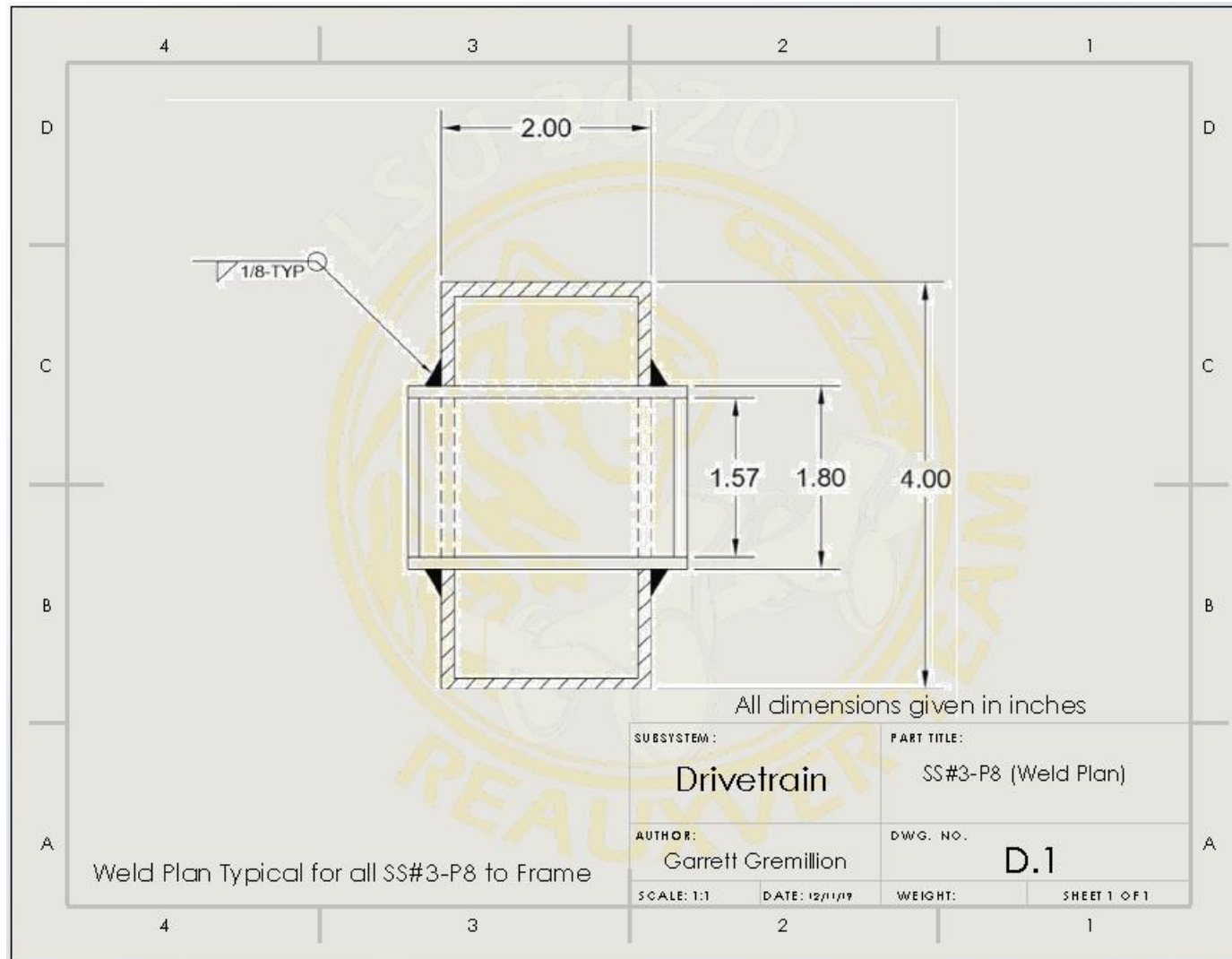


Figure XI.C.26: Manufacturing Drawing of part SS#3-P8

Note: All other components of the drivetrain will be or have been purchased/reused.

XI.C.4.Sub-System SS#4- Push-Pull Steering

The function of The Push-Pull steering design is to allow the rover to maneuver through the simulated surfaces of another world. It requires that the driver push and pull on the Steering Handles (SS#4 P6 & P7) to directly influence the motion of the wheels. Appropriate wheel angles are guaranteed by the Tie Rod linkage. (SS#4-P2 & P3)

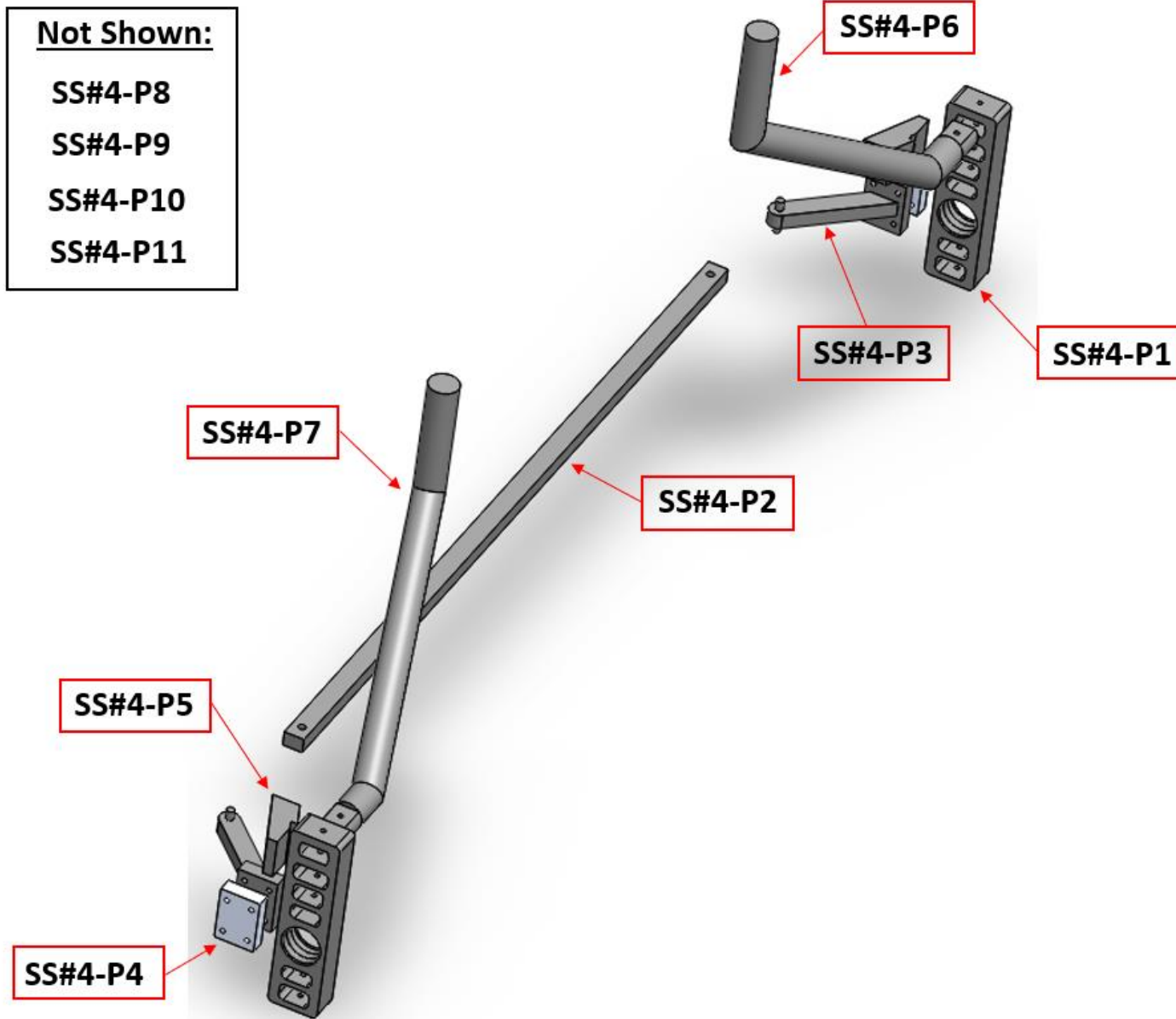


Figure XI.C.27: Exploded View Assembly Drawing of Sub-System SS#4- Push-Pull Steering

XI.C.4.a. Comprehensive Parts List for SS#4- Push-Pull Steering

Table XI-4: List of Parts for Sub-System SS#4 – Push-Pull Steering

Part #	Quantity	Material	Name
SS#4-P1	2	Al 6061-T6	Steering Knuckle
SS#4-P2	1	Al 6061-T6	Tie Rod
SS#4-P3	2	Al 6061-T6	Steering Arm
SS#4-P4	4	Al 6061-T6	Bolt Plate
SS#4-P5	2	Al 6061-T6	Range of Motion Limiter
SS#4-P6	2	Al 6061-T6	Left Steering Handle
SS#4-P7	2	Al 6061-T6	Right Steering Handle
SS#4-P8	10	Steel	Bolt Plate Bolt
SS#4-P9	2	Steel	Tie Rod Bolt
SS#4-P10	10	Steel	Bolt Plate Nut
SS#4-P11	2	Steel	Tie Rod Nut

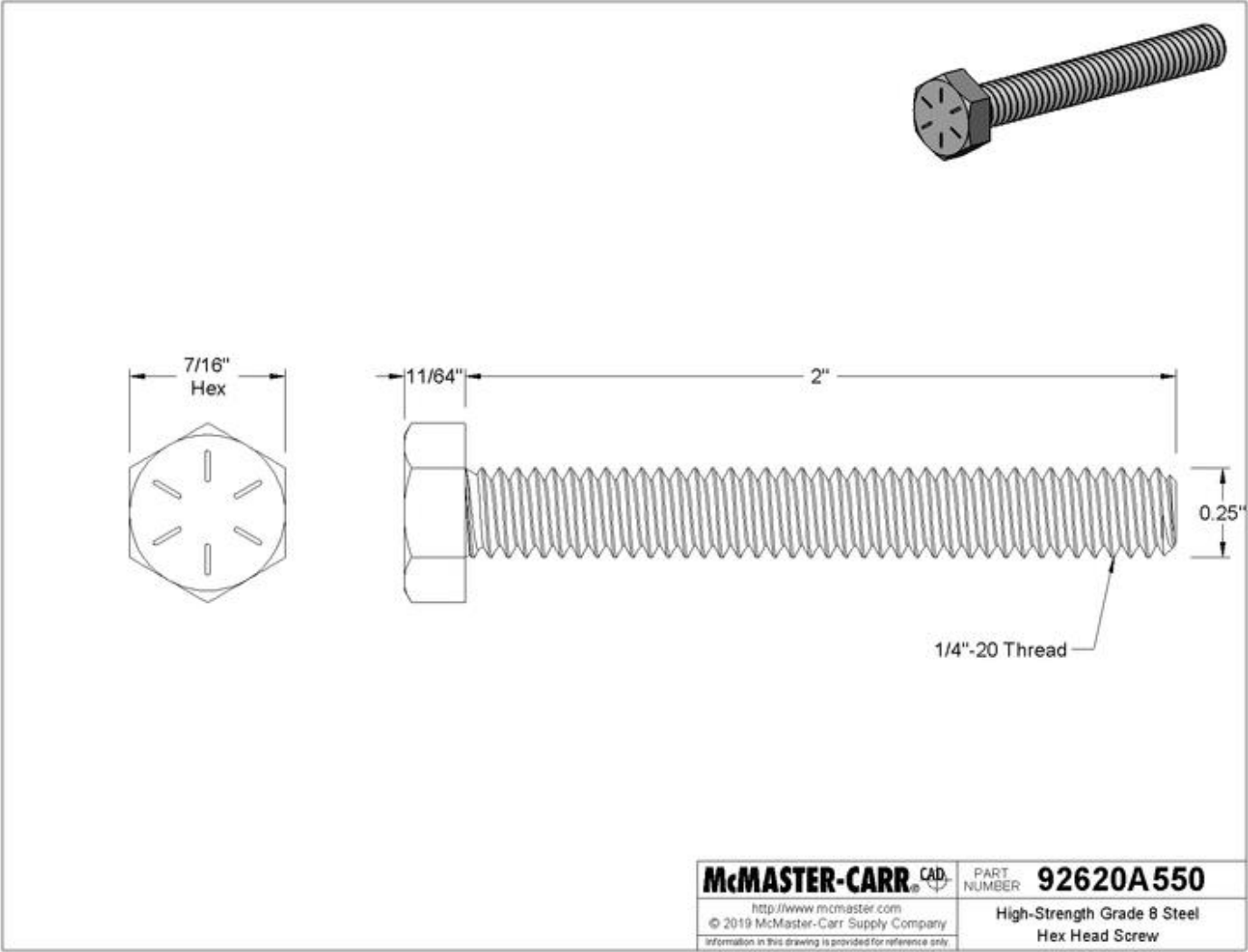


Figure XI.C.28: Specifications for part SS#4-P8 – Bolt Plate Bolt - 1/4"-20 Thread Size, 2" Long Hex Head Screw

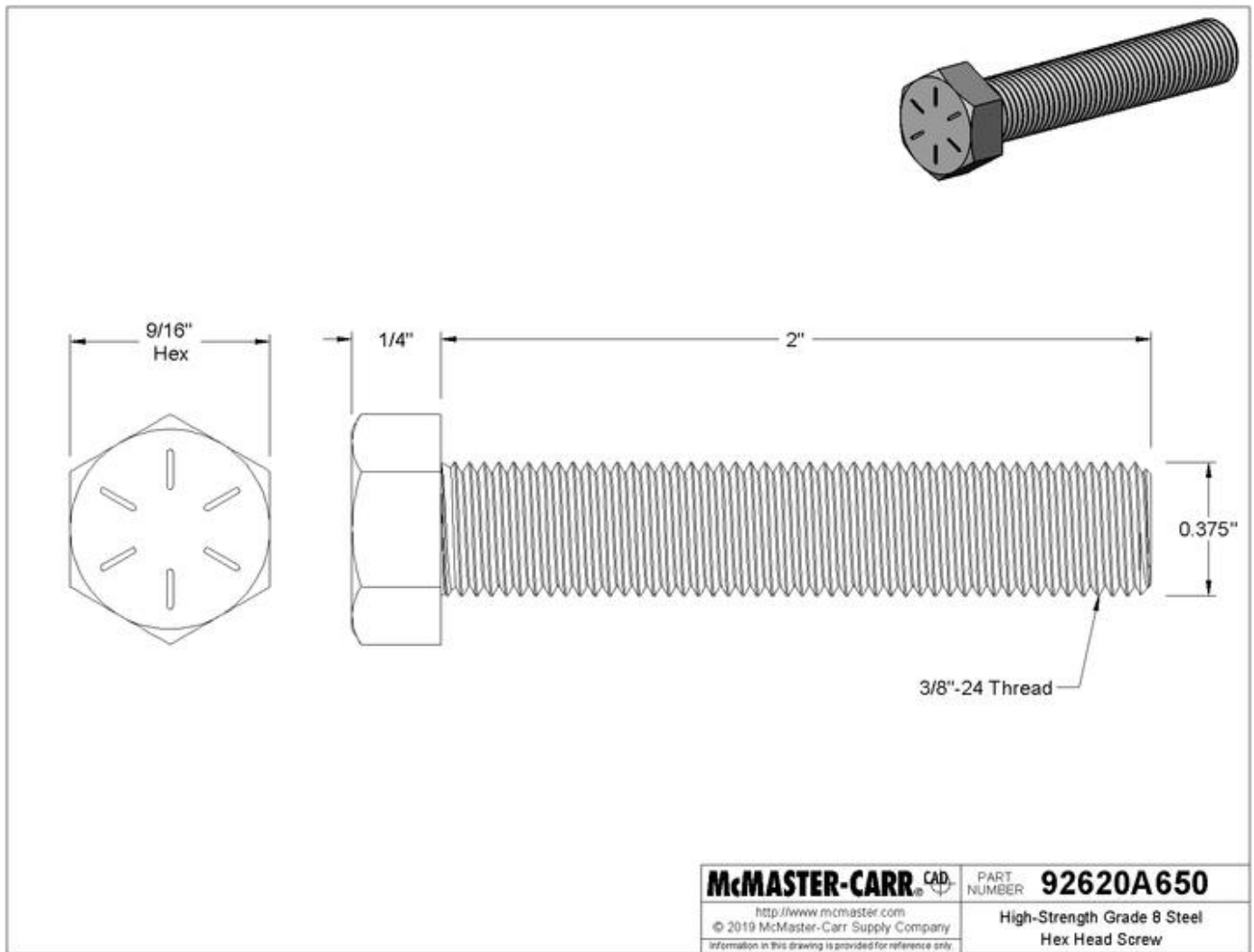


Figure XI.C.29: Specifications for part SS#4-P9 – Tie Rod Bolt - 3/8"-24 Thread Size, 2" Long Hex Head Screw

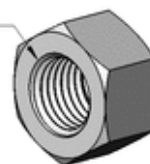
1/4"-20 Thread



McMASTER-CARR CAD	PART NUMBER 95462A029
http://www.mcmaster.com © 2015 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Hex Nut

Figure XI.C.30: Specifications for part SS#4-P10 – Bolt Plate Nut - 1/4"-20 Thread Size Hex Nut

3/8"-24 Thread



McMASTER-CARR CAD
<http://www.mcmaster.com>
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.

PART NUMBER	95462A515
	Hex Nut

Figure XI.C.31: Specifications for part SS#4-P11 – Tie Rod Bolt - 3/8"-24 Thread Size Hex Nut

XI.C.4.c. Engineering (Manufacturing) Drawings of All Parts of SS#4- Push-Pull Steering

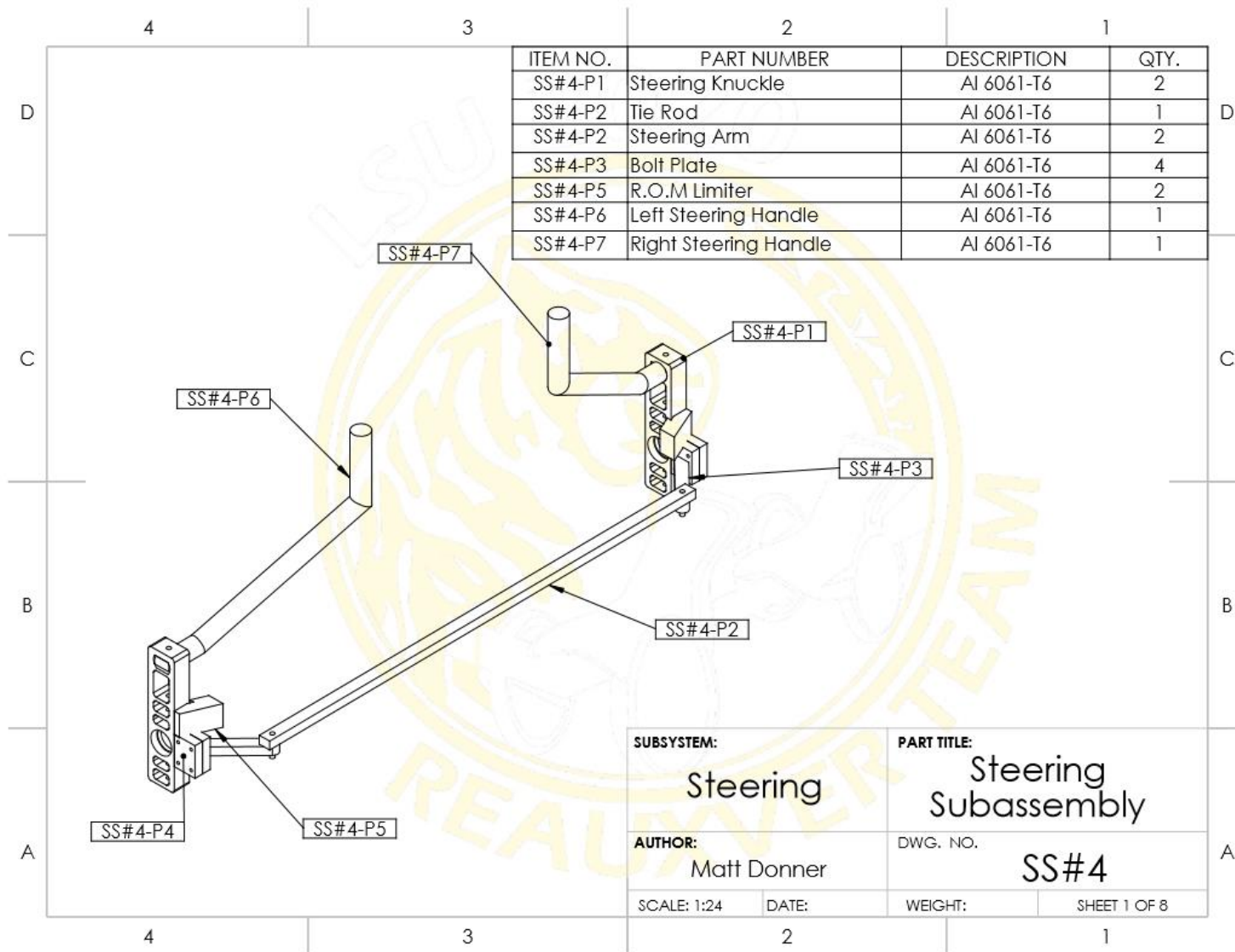


Figure XI.C.32: Manufacturing Drawing of part SS#4 - Steering Subassembly

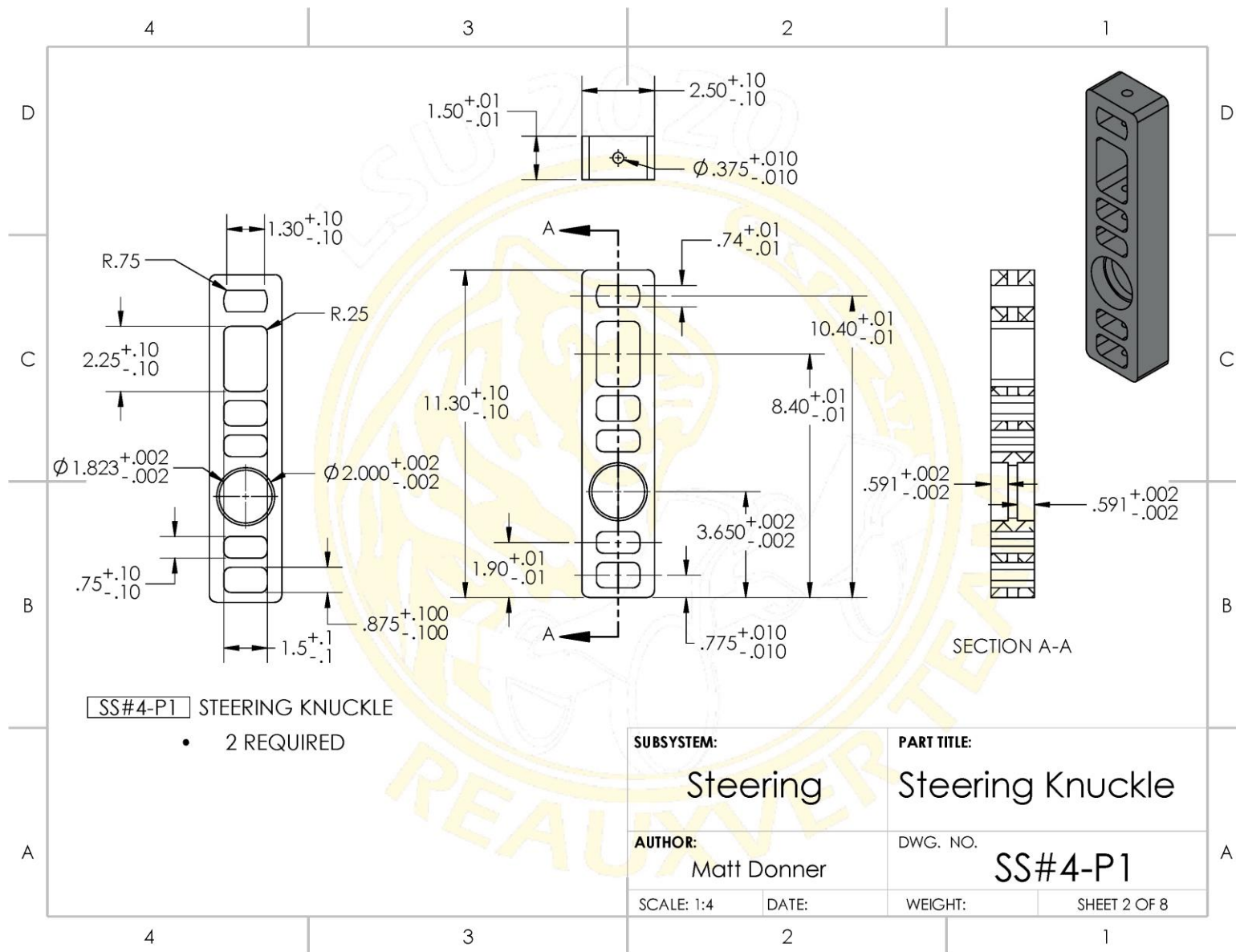


Figure XI.C.33: Manufacturing Drawing of part SS#4-P1 - Steering Knuckle

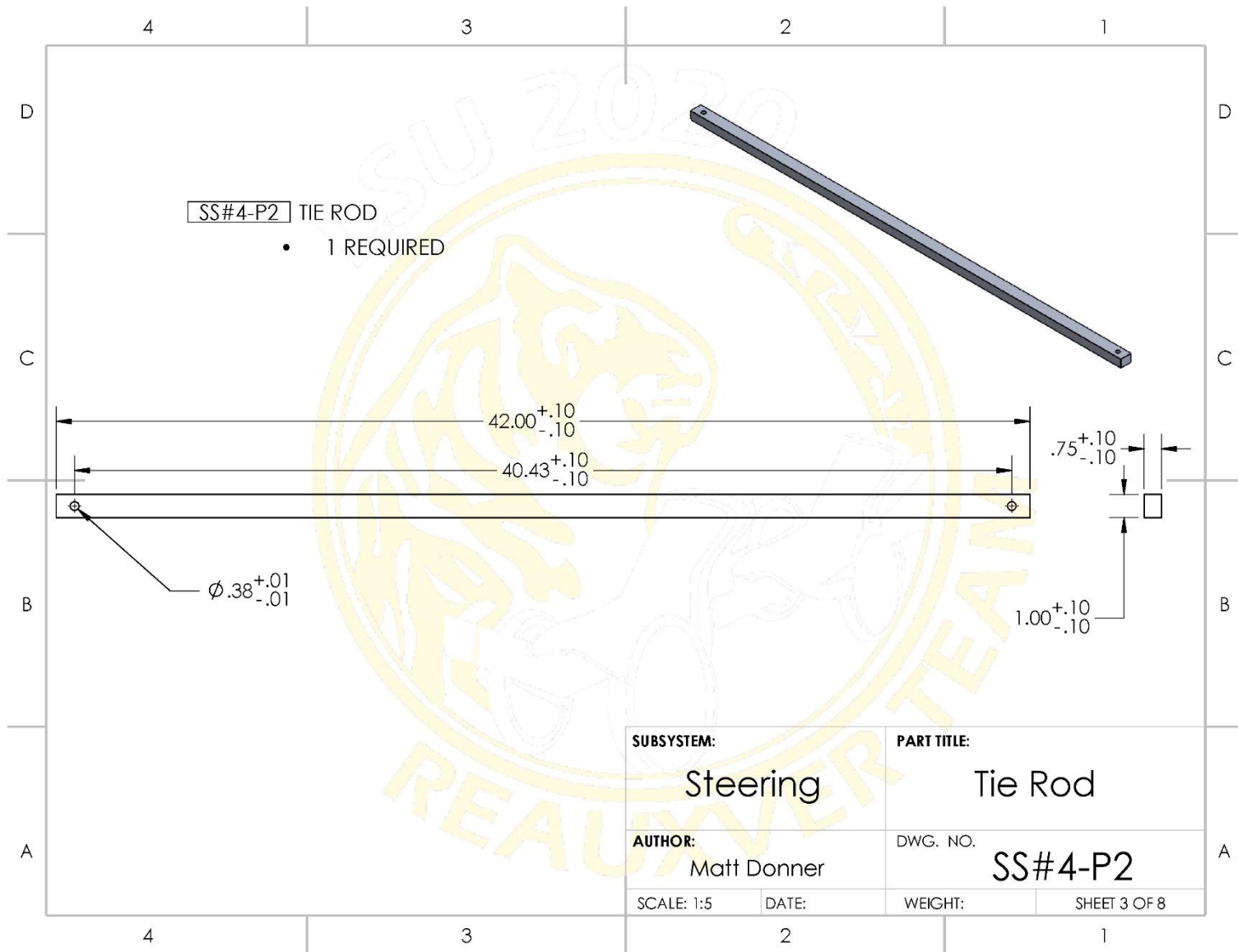


Figure XI.C.34: Manufacturing Drawing of part SS#4-P2 - Tie Rod

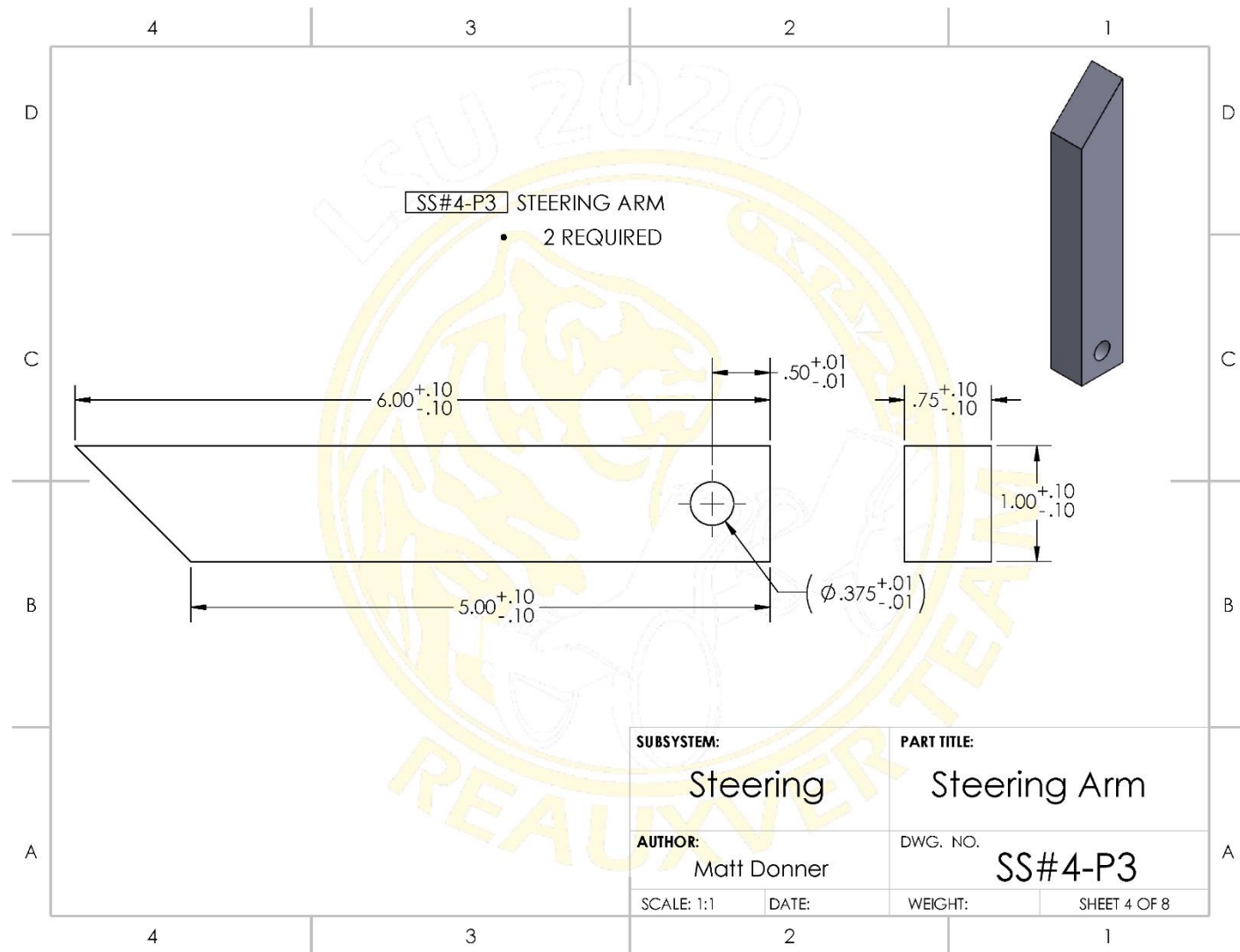


Figure XI.C.35: Manufacturing Drawing of part SS#4-P3 – Steering Arm

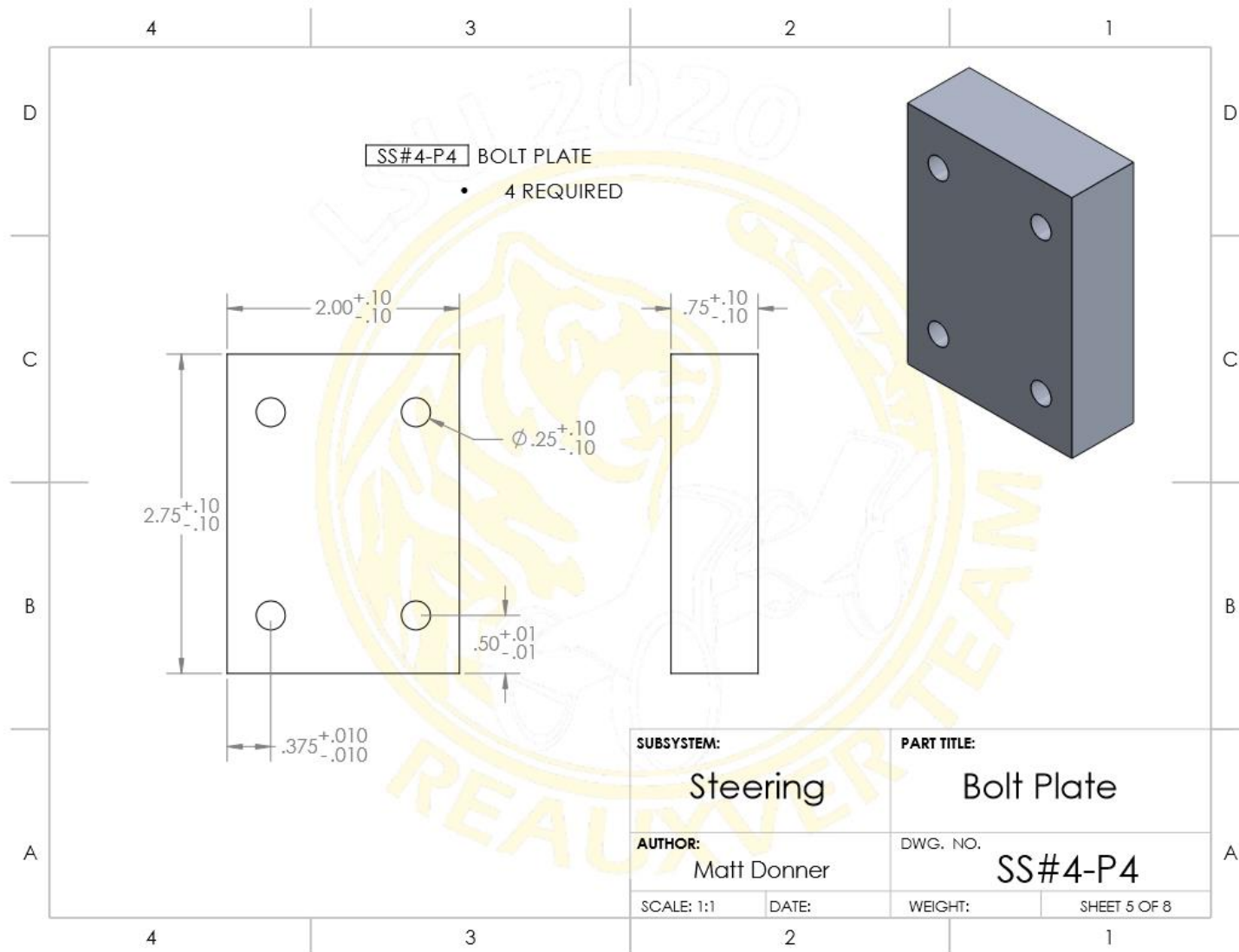


Figure XI.C.36: Manufacturing Drawing of part SS#4-P4 – Bolt Plate

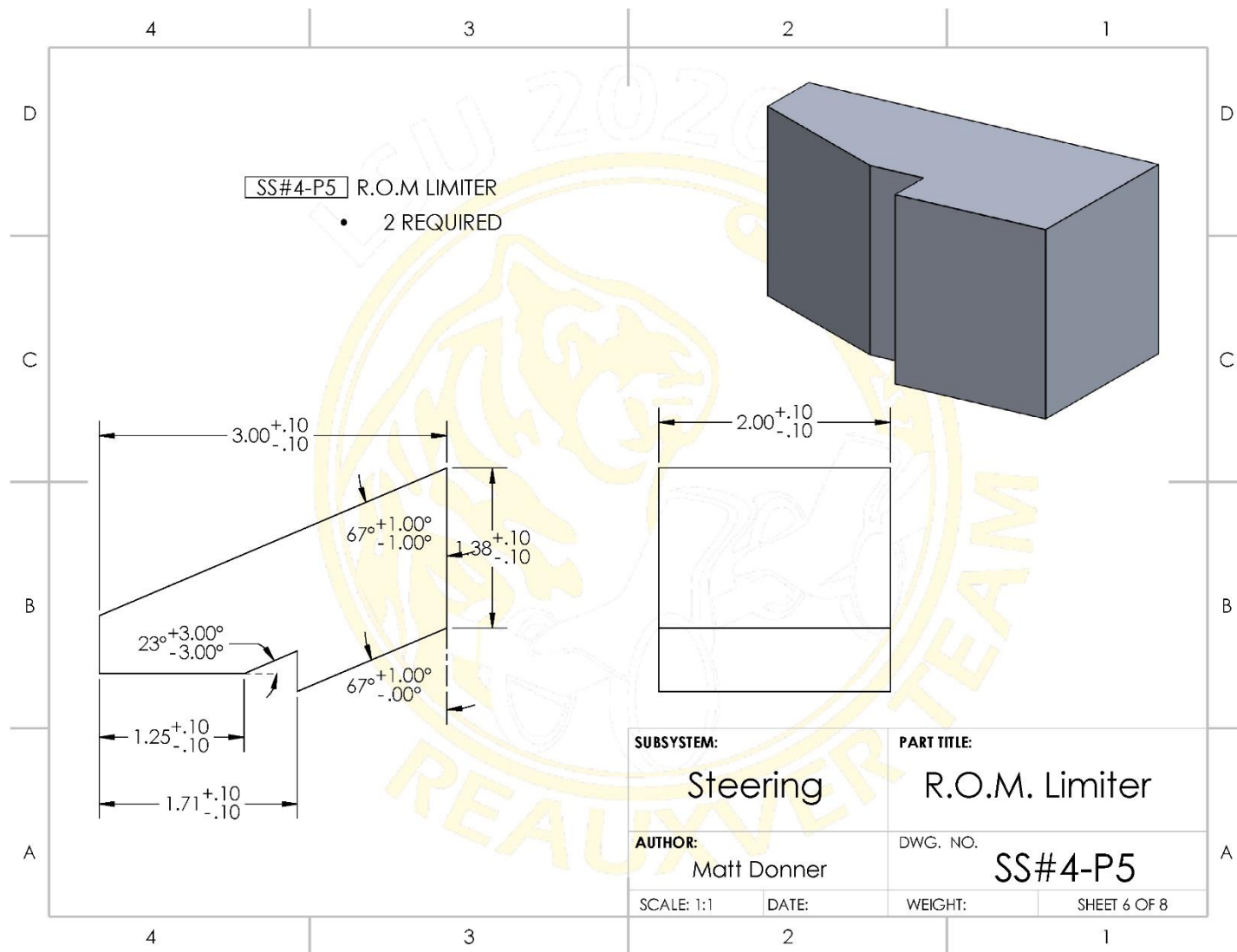


Figure XI.C.37: Manufacturing Drawing of part SS#4-P5 – Range of Motion Limiter

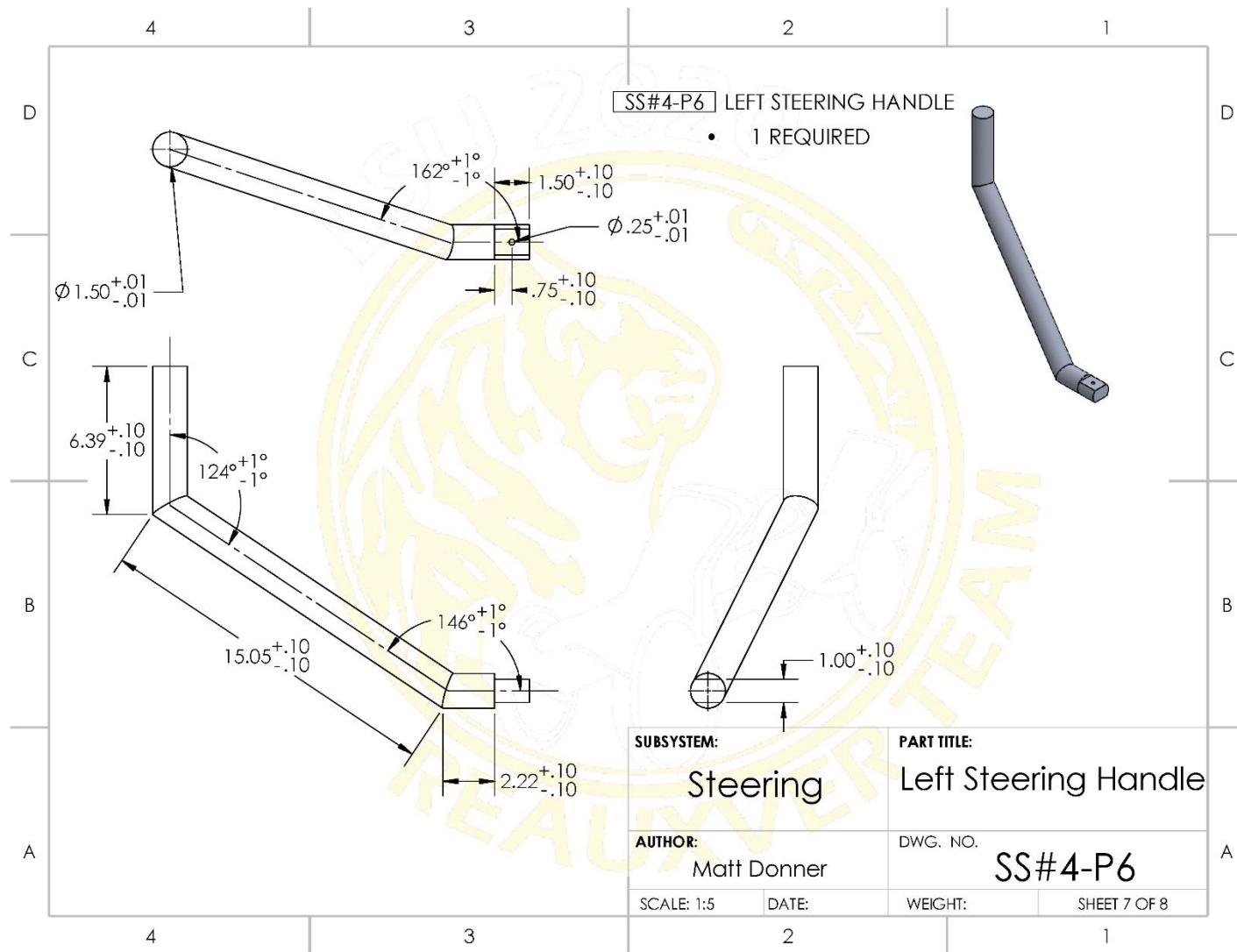


Figure XI.C.38: Manufacturing Drawing of part SS#4-P6 - Left Steering Handle

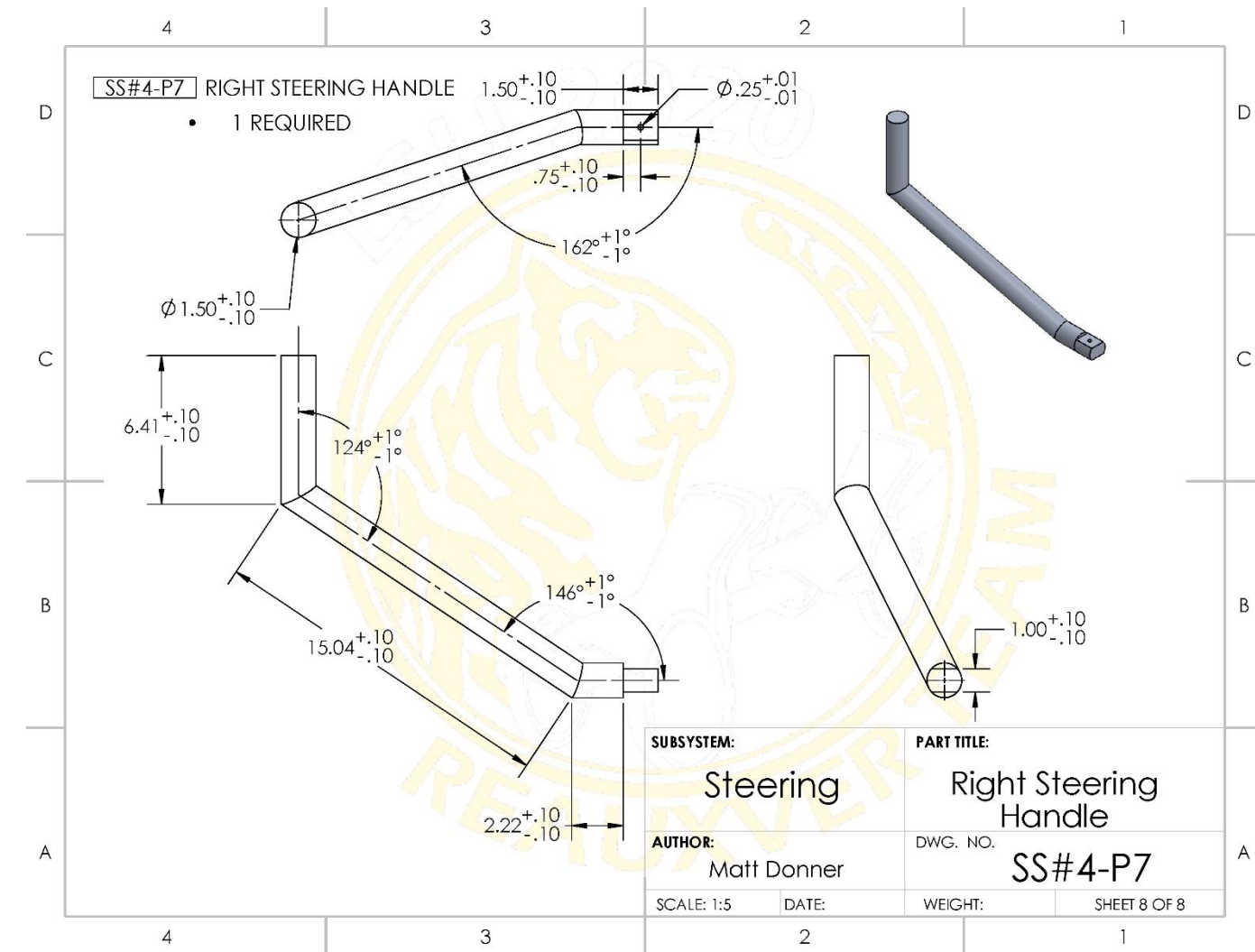
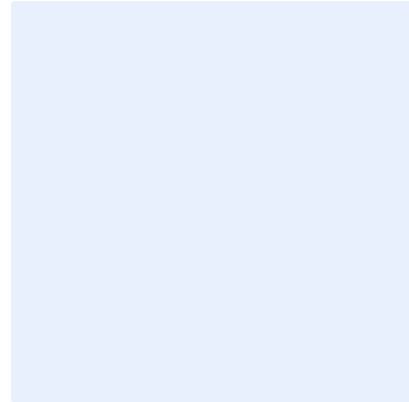


Figure XI.C.39: Manufacturing Drawing of part SS#4-P7 – Right Steering Handle

XI.C.5.Sub-System SS#5- Hydraulic Brakes

The hydraulic brakes will consist of a brake level, master cylinder, the physical disc brake, the brake caliper, brake fluid, and a piston inside of the master cylinder. As the brake level is squeezed by the rider, the piston is actuated inside of the master cylinder. As the piston rises, it forces fluid toward the brake caliper, which squeezes the brake disc, and thus slows the rover's wheels. The braking rotor is attached directly to the hub for efficient braking power (Figure XI.C.5)



XI.C.5.a. Comprehensive Parts List for SS#5- Hydraulic Brakes

Table XI-5: List of Parts for Sub-System SS#5 – Hydraulic Brakes

Part #	Quantity	Material	Name
SS#5-P1	2	Stainless Steel	Shimano Deore M6000 braking system



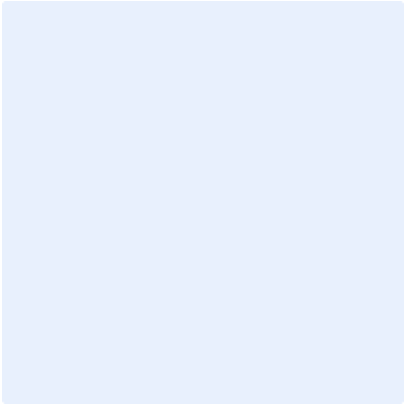
SHIMANO SLX RT66 6-BOLT ROTOR

The Shimano SLX RT66 6-BOLT Rotor features a lightweight and sleek design that is super reliable.

SHIMANO SLX RT66 6-BOLT ROTOR SPECIFICATIONS

- Hub/Disc Compatibility: 6-Bolt Disc
- Pad Compatibility: All Pad Compounds
- Rotor Size (mm): 203mm, 160mm, 180mm
- Rotor Type: One Piece
- Includes rotor fixing bolts and non-turn washers

Figure XI.C.540: Component Specifications for SS#5-P1 - Shimano Braking System



XI.C.6.Sub-System SS#6- Double Wishbone Suspension

The function of the double wishbone suspension is to absorb as much of the impact forces seen throughout the obstacle course as possible. The double wishbone suspension allows for other subsystems of the rover to be incorporated easily into the design because of its geometric configuration. This configuration also increases the negative camber, providing better stability and handling for the rover.

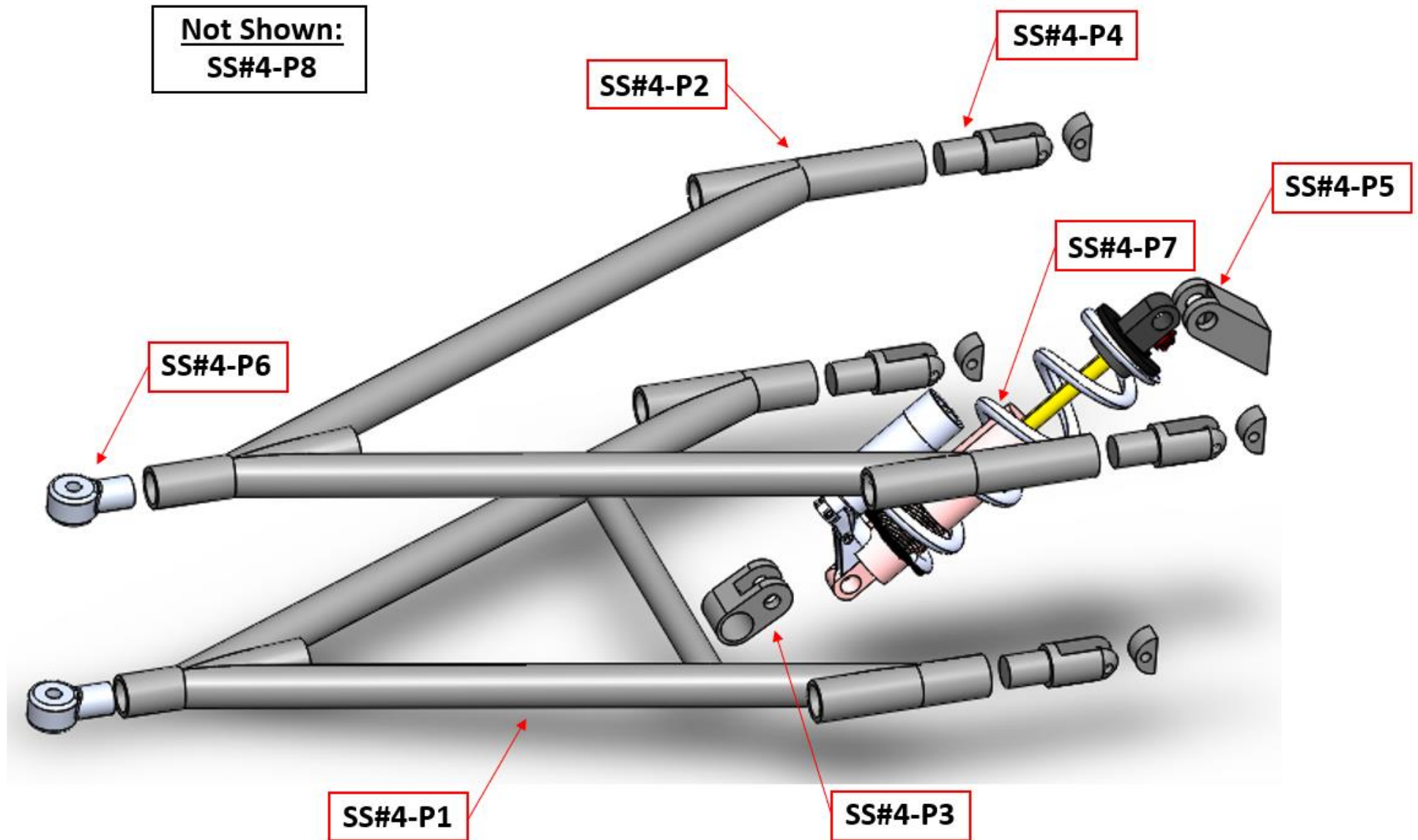


Figure XI.C.41: Exploded View Assembly Drawing of Sub-System SS#6- Double Wishbone Suspension

XI.C.6.a. Comprehensive Parts List for SS#6- Double Wishbone Suspension

Table XI-6: List of Parts for Sub-System SS#6 – Double Wishbone Suspension

Part #	Quantity	Material	Name
SS#6-P1	4	Al 6061-T6	Bottom A-Frame
SS#6-P2	4	Al 6061-T6	Top A-Frame
SS#6-P3	4	Al 6061-T6	A-Frame Connector
SS#6-P4	16	Al 6061-T6	Wishbone Connector
SS#6-P5	4	Steel	Clevis Rod Ends
SS#6-P6	4	Steel	Ball Joint Rod Ends
SS#6-P7	2	N/A	Rockshox Vivid R2C (Front)
SS#6-P8	2	N/A	Rockshox Vivid Air R2C (Rear)

XI.C.6.b. Off-The-Shelf Parts and Component Specifications for SS#6– Double Wishbone Suspension

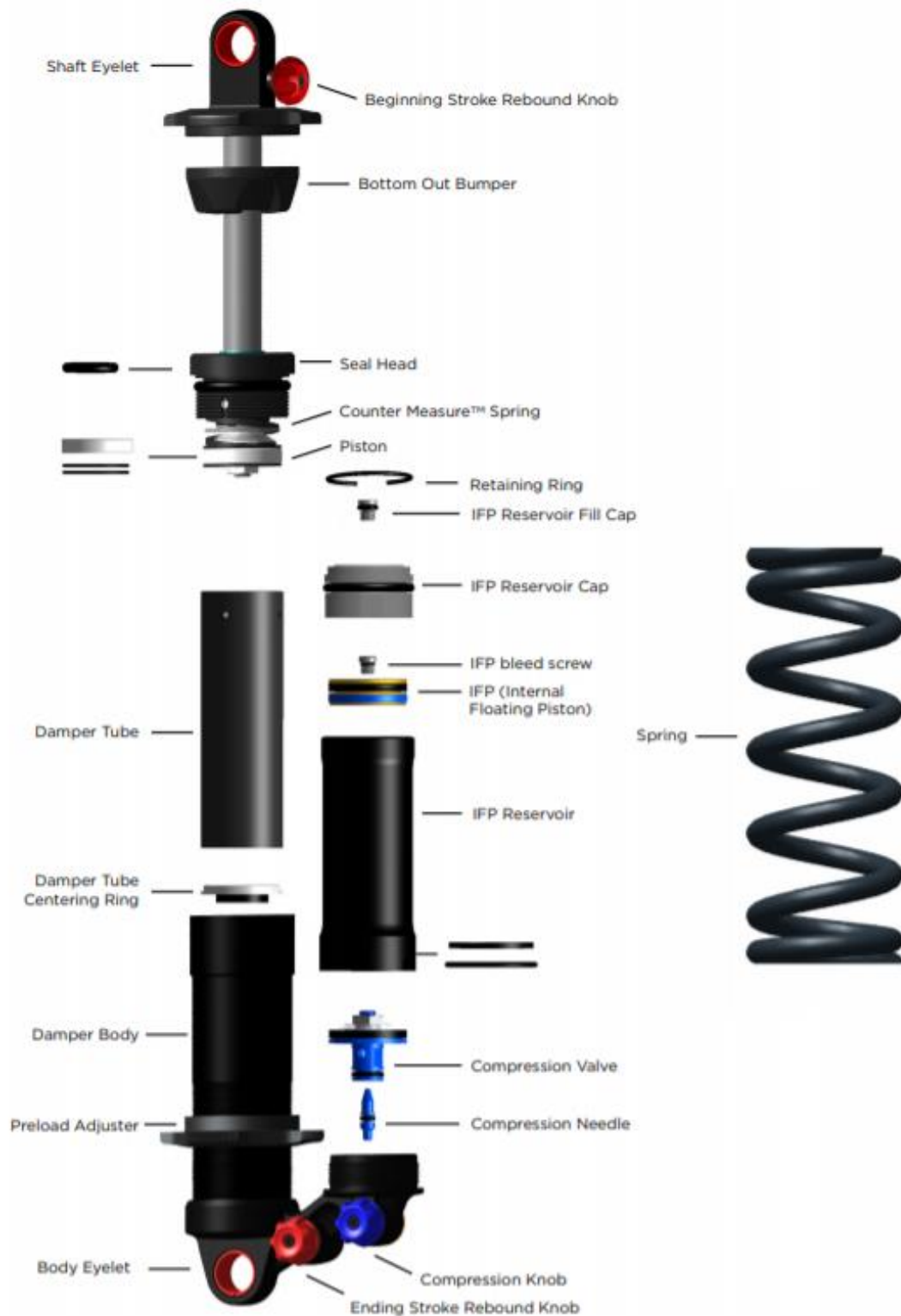


Figure XI.C.42: Specifications for part SS#6-P7 – Rockshox Vivid R2C (Vivid Service Manual, 2014) [13]

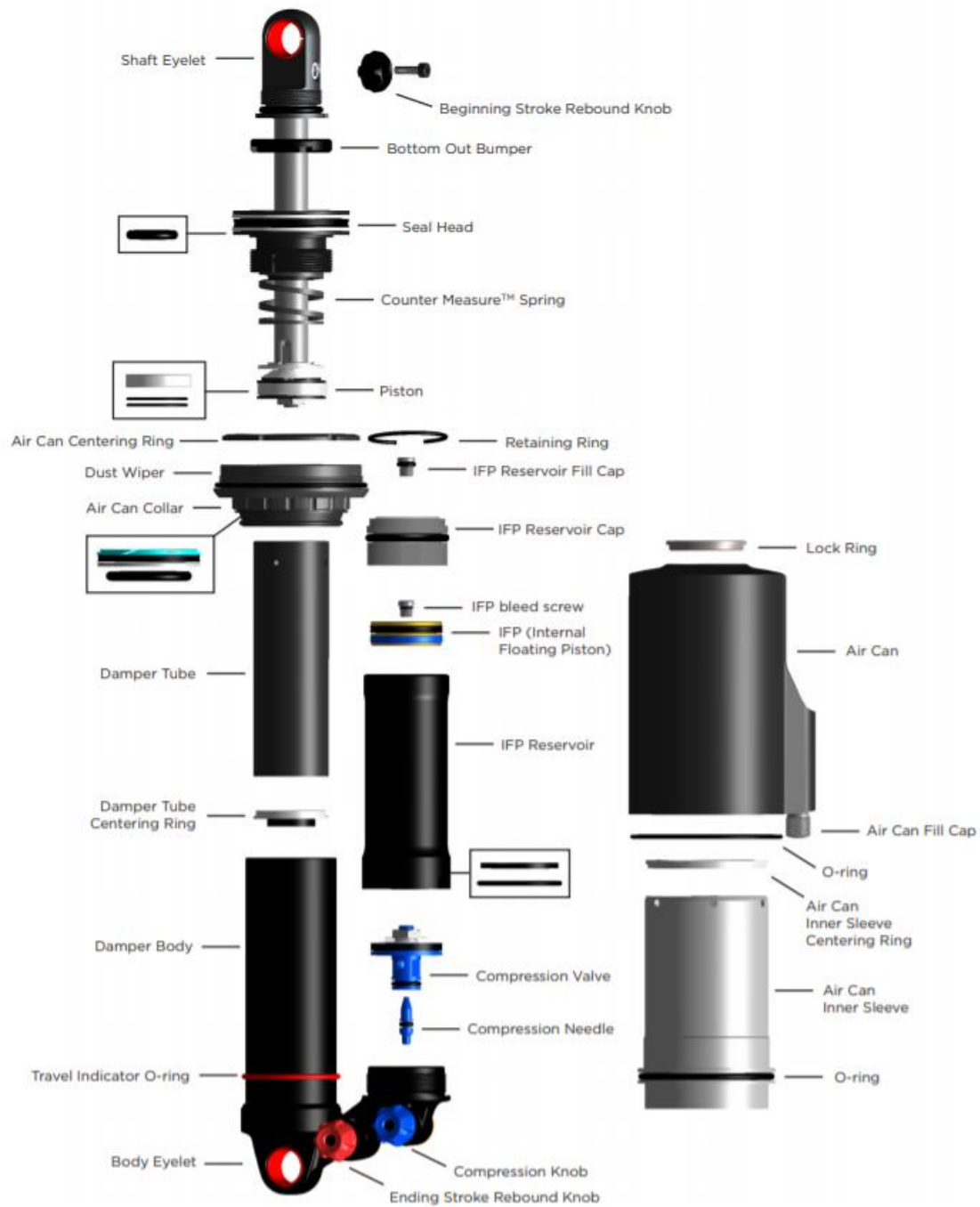


Figure XI.C.43: Specifications for part SS#6-P8 – Rockshox Vivid Air R2C (Vivid Air Service Manual, 2014) [13]



Shank Thread Size	1/4"-20
ID	1/4"
Jaw Opening Width	1/4"
Overall Width	5/8"
Shank Center Length	2"
Shank Thread Length	3/4"
Jaw Center Length	1 1/4"
Tensile Strength	58,000 psi
Material	Carbon Steel
Rod End Type	Clevis
Includes	Clevis Pin, Cotter Pin
Shank Threading	Fully Threaded
Shank Gender	Female
Shank Thread Direction	Right Hand
RoHS	RoHS 3 (2015/863/EU) compliant

Figure XI.C.44: Specifications for part SS#6-P5 – Clevis Rod End (McMaster-Carr, 2019) [14]



Shank Thread Size	1/4"-28
ID	1/4"
ID Tolerance	-0.0005" to 0.0025"
Maximum Ball Swivel	27°
Overall Width	3/4"
Overall Thickness	3/8"
Shank Center Length	1 9/16"
Shank Thread Length	1"
Static Radial Load Capacity	2,800 lbs.
Material	Zinc-Plated Carbon Steel
Ball Material	Chrome-Plated Bearing Steel
Lubrication	Required
Rod End Type	Ball Joint
Shank Threading	Fully Threaded
Shank Gender	Male
RoHS	RoHS 3 (2015/863/EU) compliant

Figure XI.C.45: Specifications for part SS#6-P6 – Ball Joint Rod End (McMaster-Carr, 2019) [15]

XI.C.6.c. Engineering (Manufacturing) Drawings of All Parts of SS#6- Double Wishbone Suspension

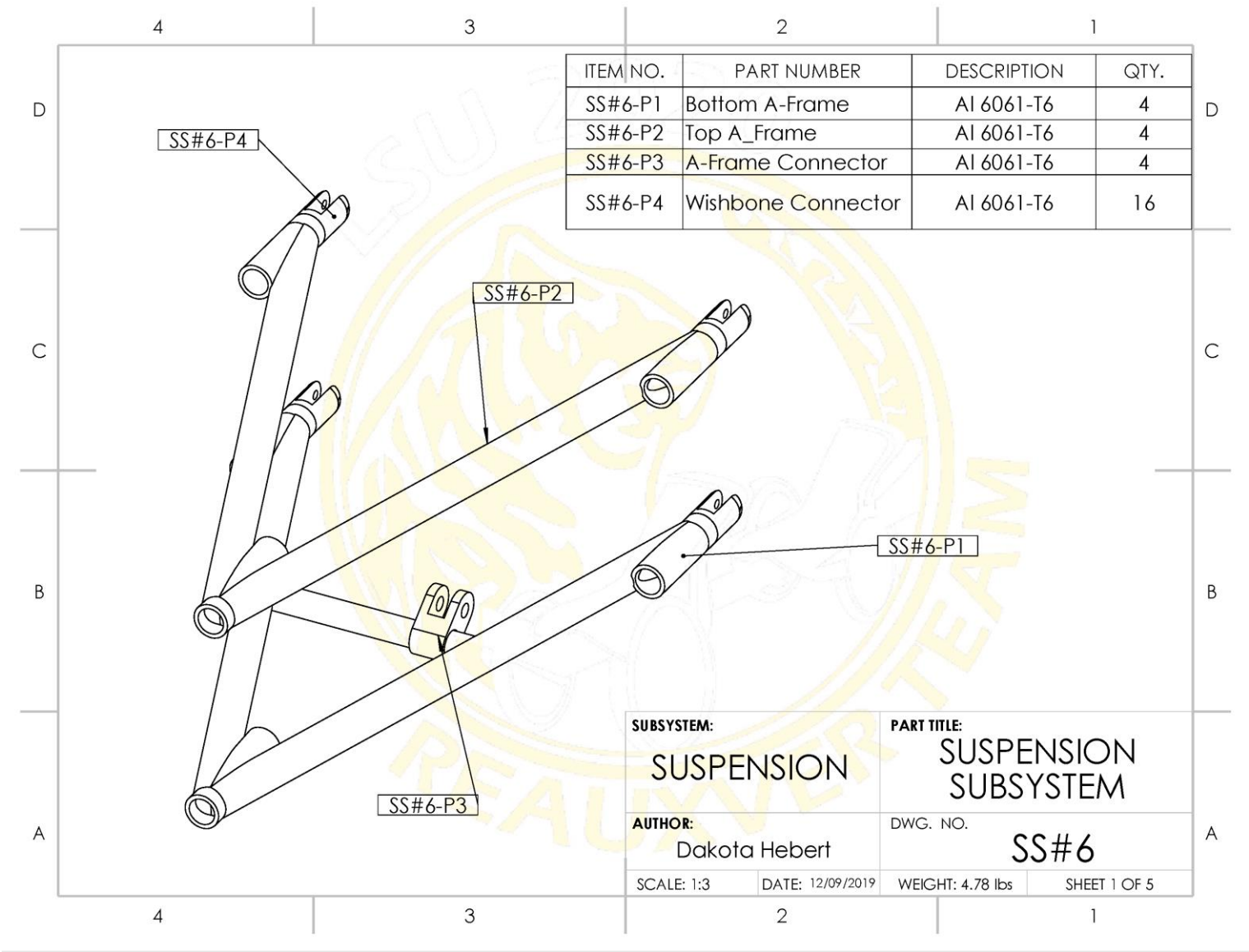


Figure XI.C.46: Manufacturing Drawing of part SS#6 – Suspension Subassembly

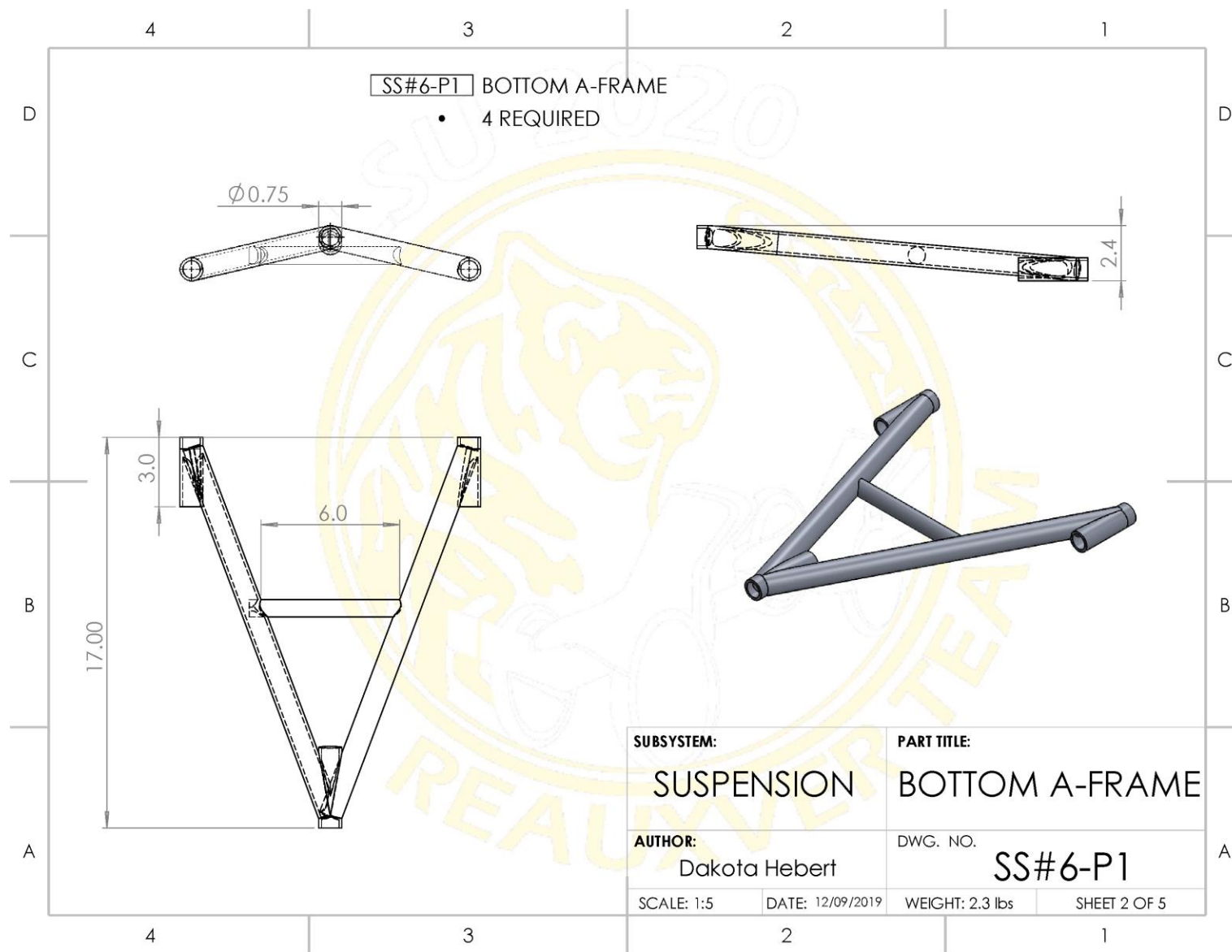


Figure XI.C.47: Manufacturing Drawing of part SS#6-P1 – Bottom A-Frame

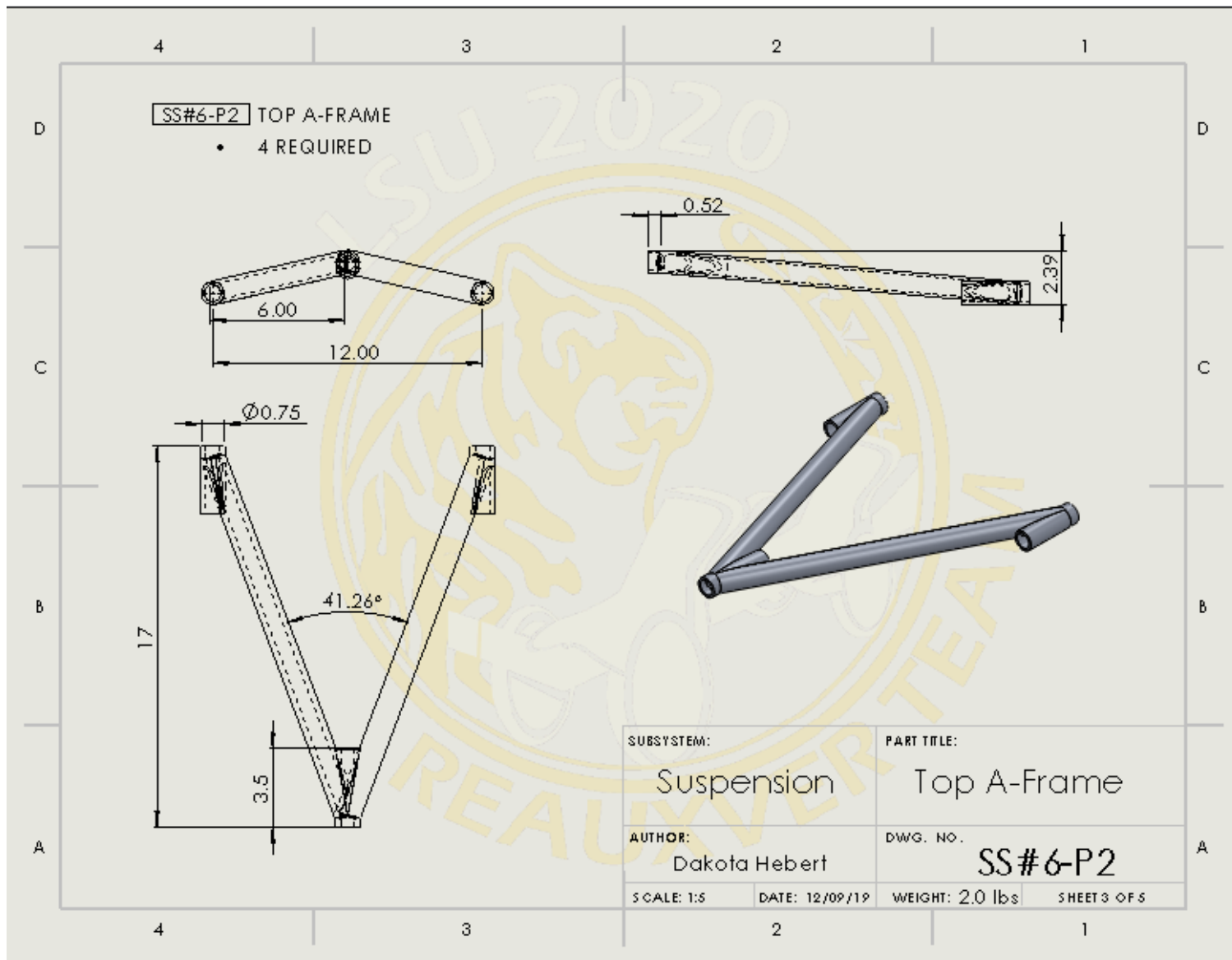


Figure XI.C.48: Manufacturing Drawing of part SS#6-P4 – Top A-Frame

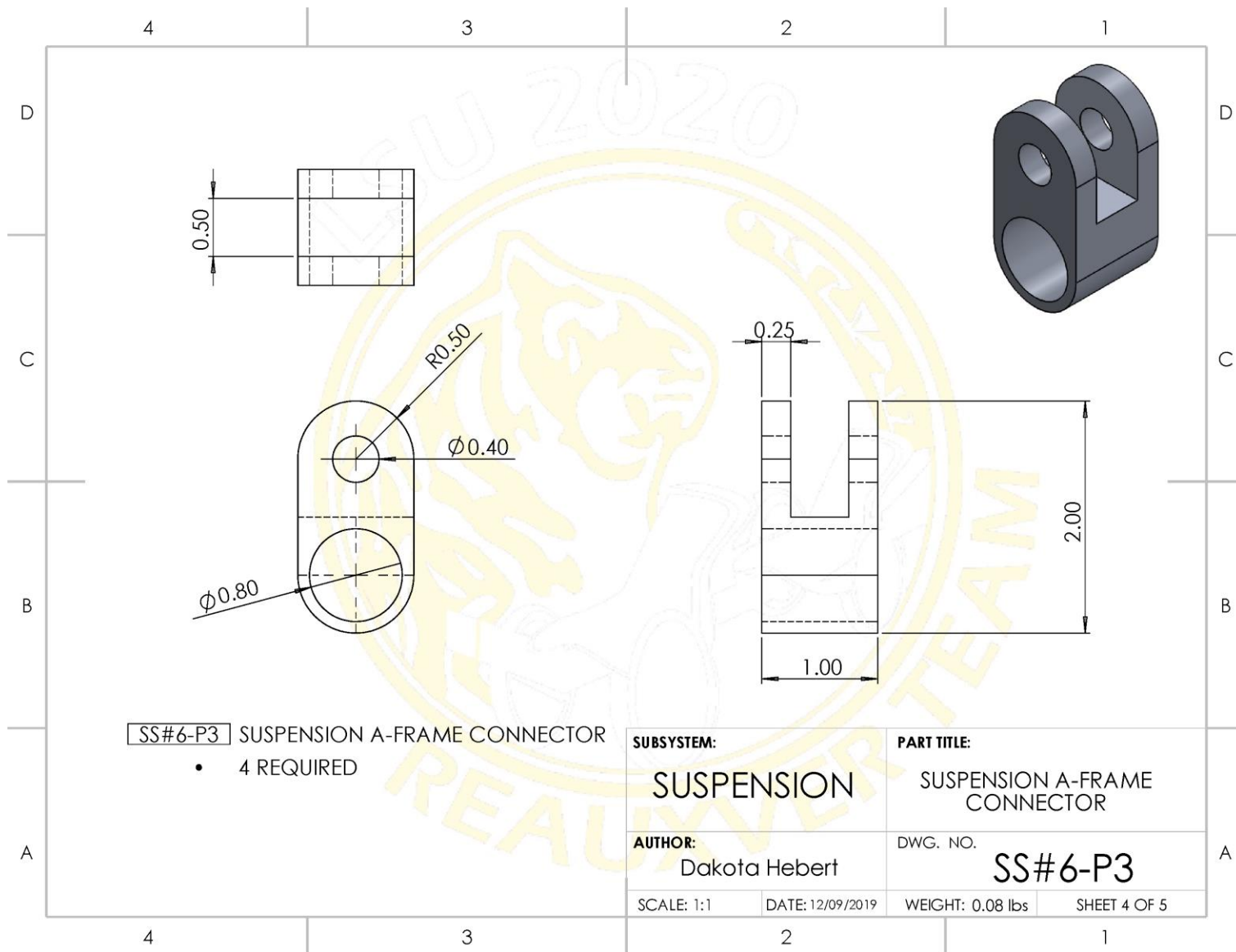


Figure XI.C.49: Manufacturing Drawing of part SS#6-P3 – Suspension A-Frame Connector

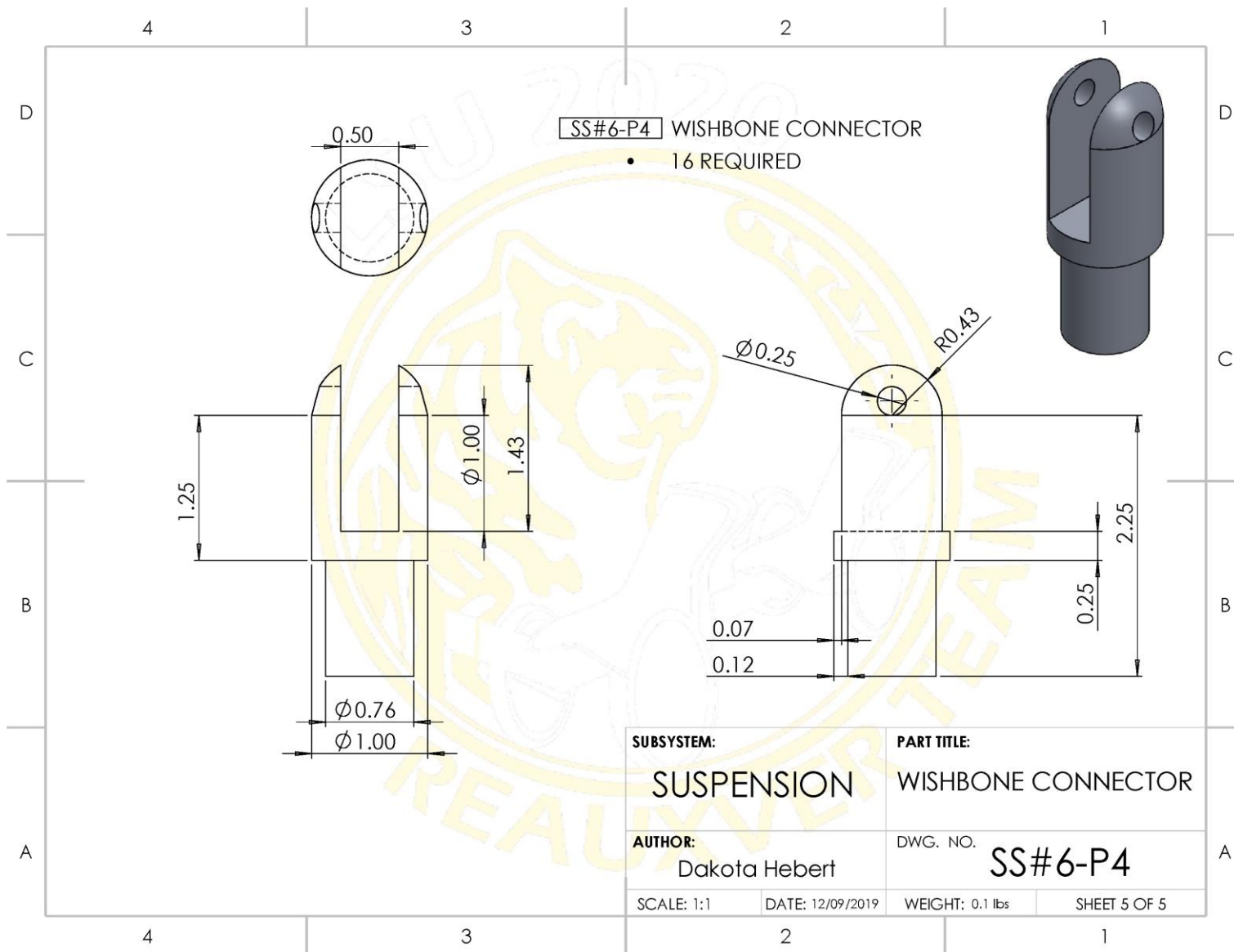


Figure XI.C.50: Manufacturing Drawing of part SS#6-P4 – Wishbone Connector

XI.C.7.Sub-System SS#7- Seats

The main functions of the seats are to provide a safe, ergonomic position. The seats each feature a safety belt, that will be attached over the rider's lap. By design recumbent seats are ergonomic, but the ability to adjust the rider's angle is also supported.

Not Shown:

SS#7-P1
SS#7-P2
SS#7-P7
SS#7-P9
SS#7-P10
SS#7-P11

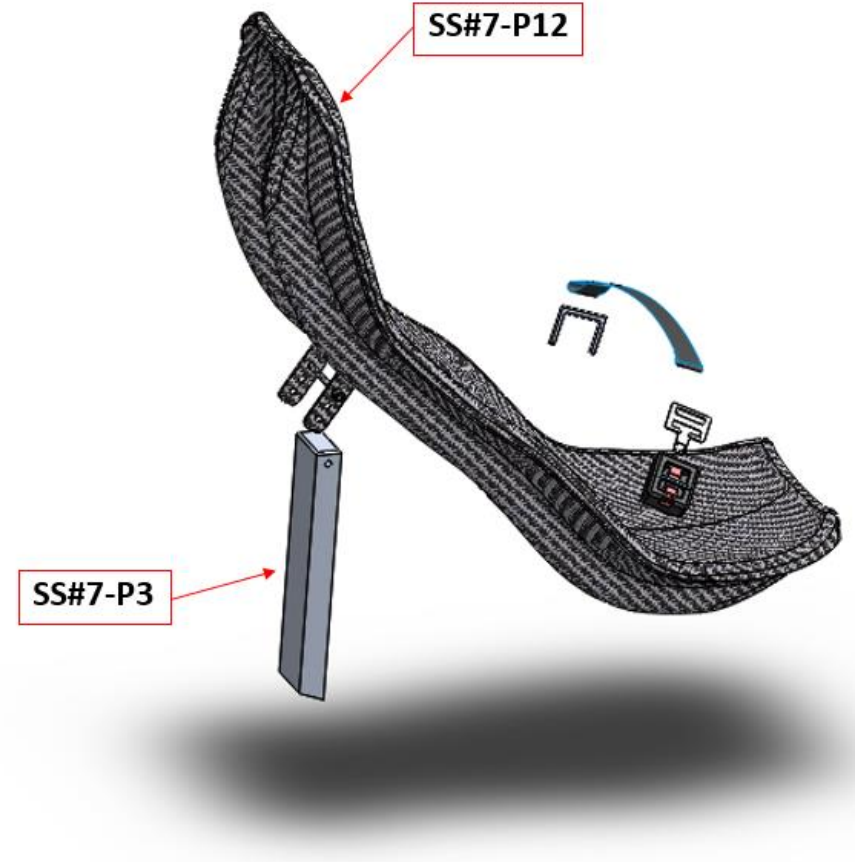
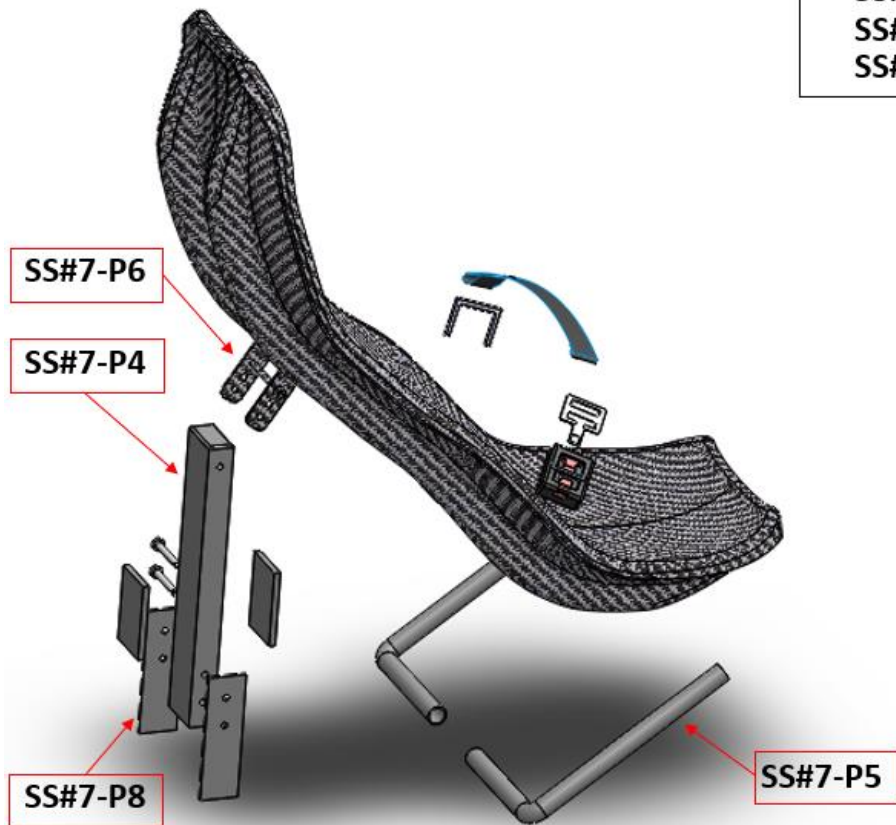


Figure XI.C.51: Exploded View Assembly Drawing of Sub-System SS#7- Seats

XI.C.7.a. Comprehensive Parts List for SS#7- Seats

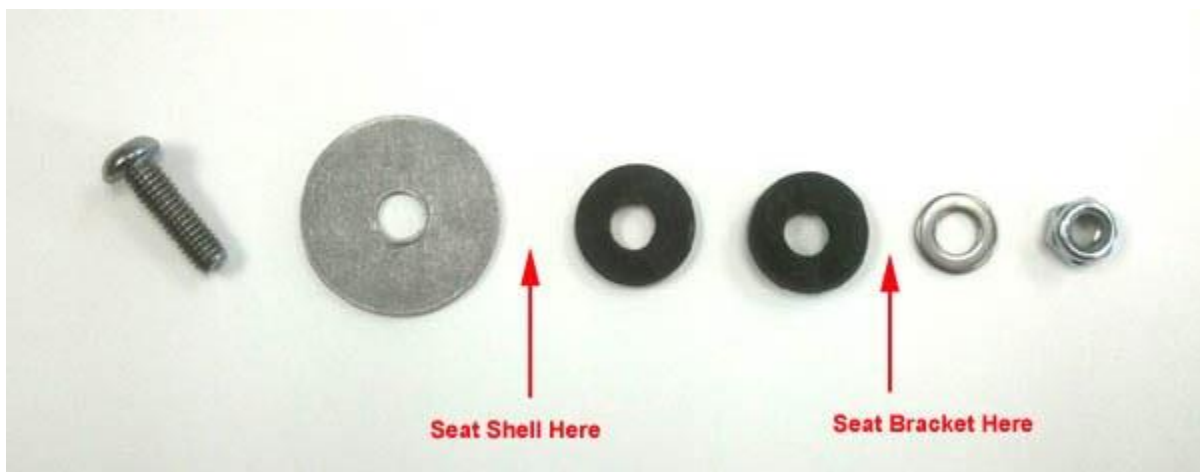
Table XI-7: List of Parts for Sub-System SS#7 – Seats

Part #	Quantity	Material	Name
SS#7-P1	2	Al 6061-T6	Bottom Seat Bracket
SS#7-P2	2	Al 6061-T6	Bottom Seat Attachment
SS#7-P3	1	Aluminum	Front Seat Bar
SS#7-P4	1	Aluminum	Rear Seat Bar
SS#7-P5	2	Aluminum	Rear Seat Handle
SS#7-P6	2	Aluminum	Seat Top Attachment
SS#7-P7	2	Aluminum	Bottom mounting bracket
SS#7-P8	1	Aluminum	Rear seat holder
SS#7-P9	4	Steel	Bolts, 2 ¾" long, Zinc-plated ¼"- 20 Thread
SS#7-P10	2	Steel	Bolts, 2 ½" long, Zinc-plated ¼"- 20 Thread
SS#7-P11	6	Steel	Medium-Strength Steel Hex Nut, Zinc-plated ¼"- 20 Thread
SS#7-P12	2	Carbon Fiber	Volae Carbon Comfort Seat Shells

XI.C.7.b. Off-The-Shelf Parts and Component Specifications for SS#7– Seats



Figure XI.C.52: Specifications for part SS#7-P1 – Volae Comfort Carbon Seat Shell (Volae, 2019) [16]



Comfort Carbon Kit includes the following parts:

- Eight M6x25 bolts
- Sixteen rubber washers
- Eight large flat washers
- Eight small flat washers
- Eight Nyloc nuts

Figure XI.C.53: Specifications for part SS#7-P2 – Volae - Comfort Carbon Hardware Kit (Volae, 2019) [16]

XI.C.7.c. Engineering (Manufacturing) Drawings of All Parts of SS#7- Seats

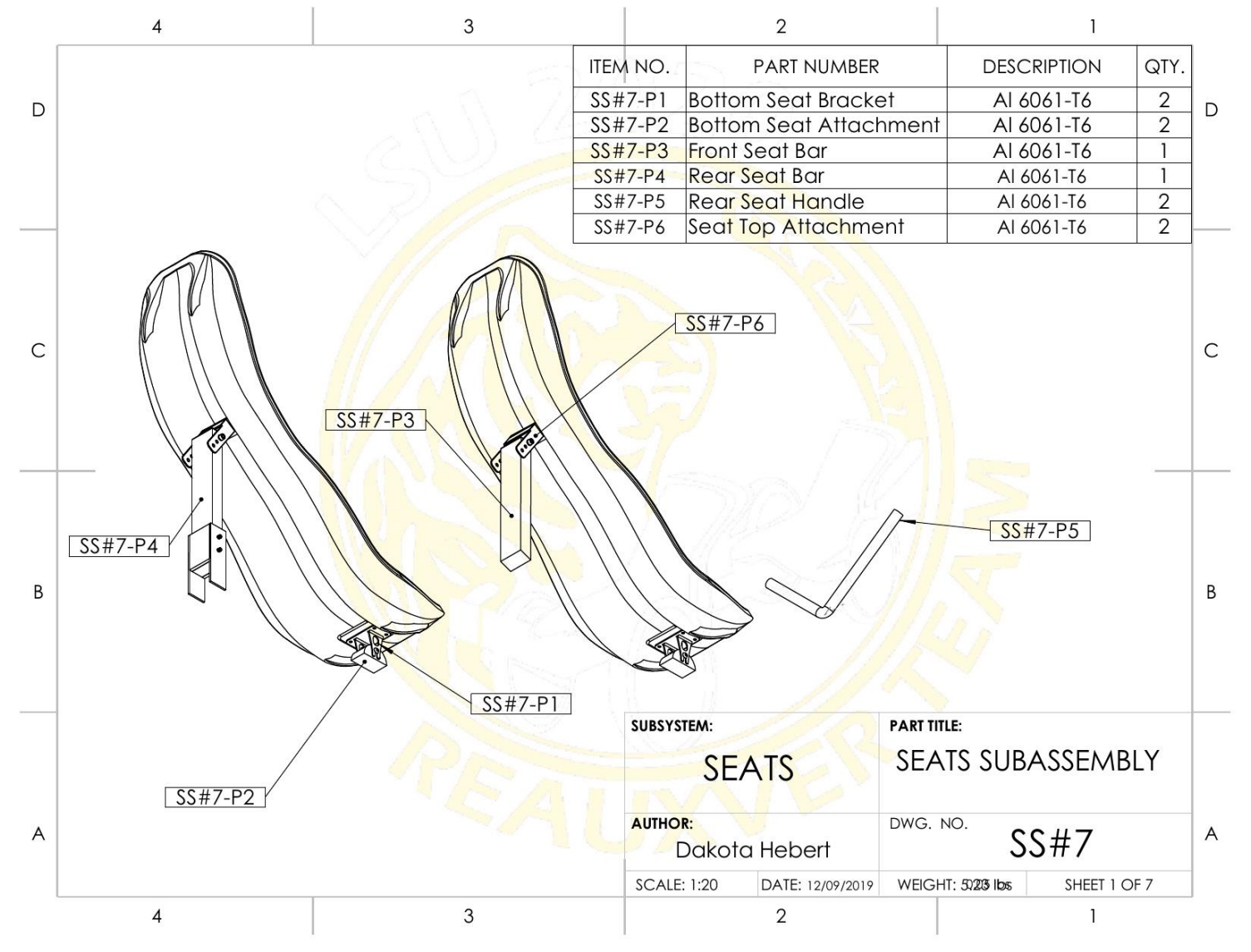


Figure XI.C.54: Manufacturing Drawing of part SS#7 – Seats Subassembly

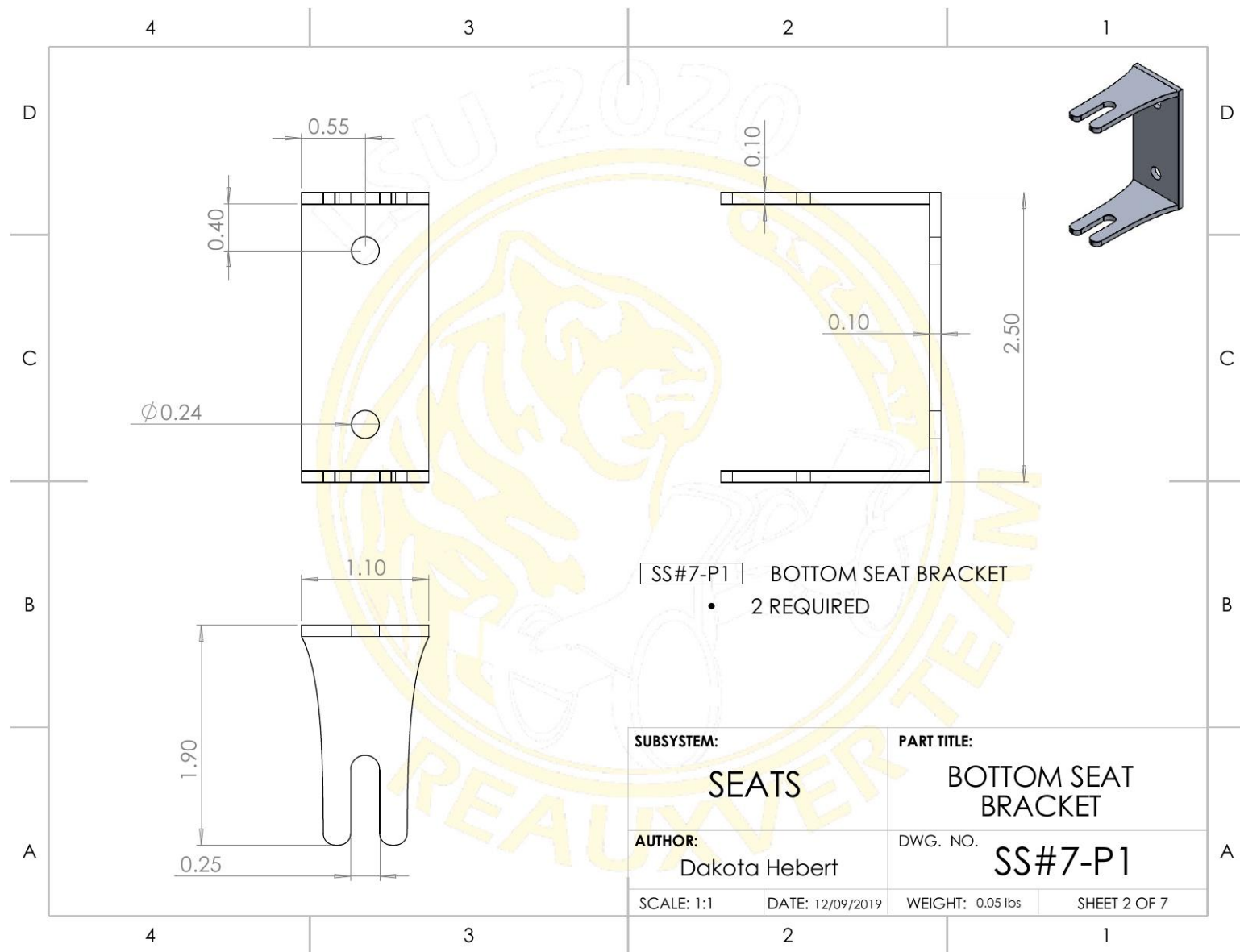


Figure XI.C.55: Manufacturing Drawing of part SS#7-P1 – Bottom Seat Bracket

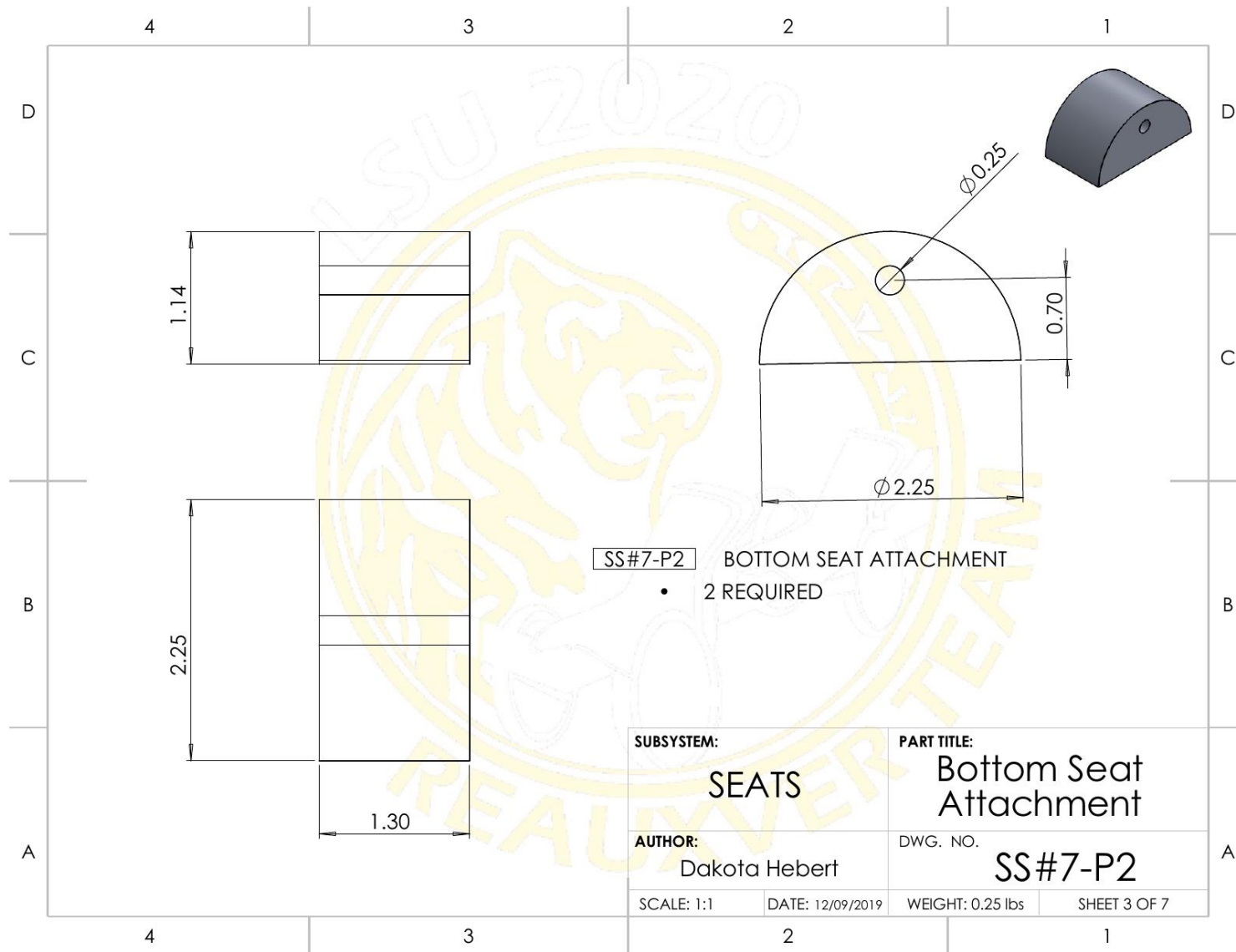


Figure XI.C.56: Manufacturing Drawing of part SS#7-P2 – Bottom Seat Attachment

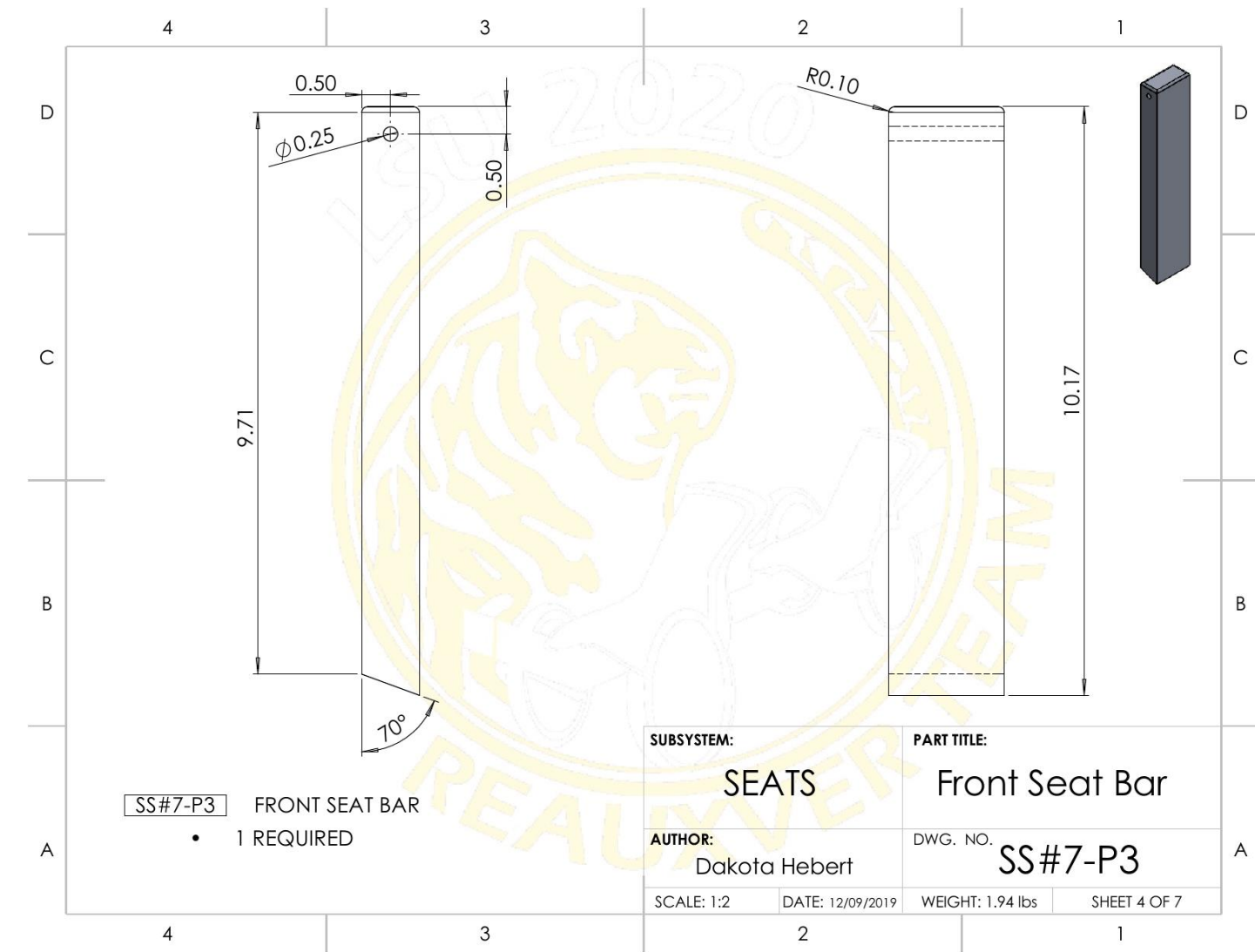


Figure XI.C.57: Manufacturing Drawing of part SS#7-P3 – Front Seat Bar

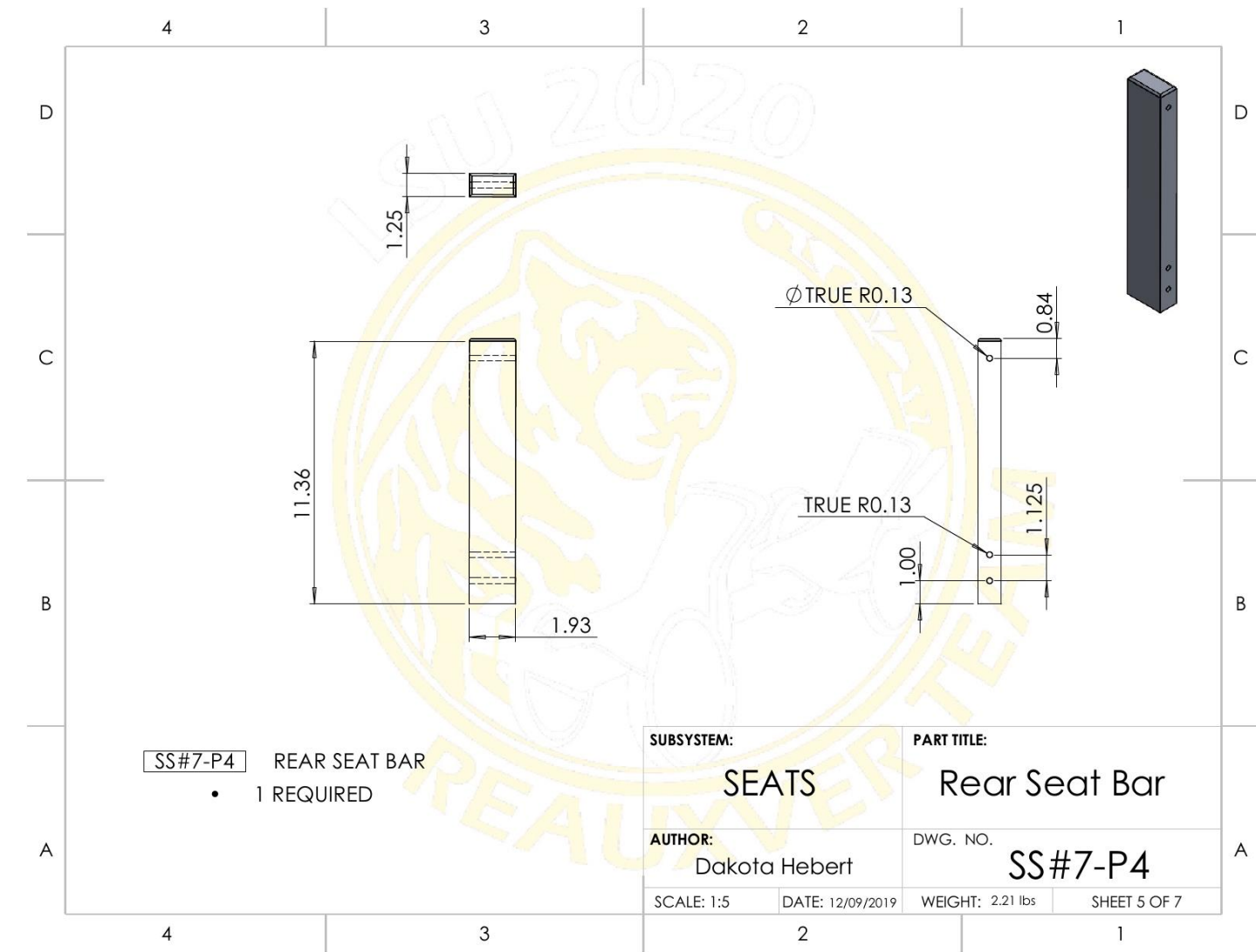


Figure XI.C.58: Manufacturing Drawing of part SS#7-P4 – Rear Seat Bar

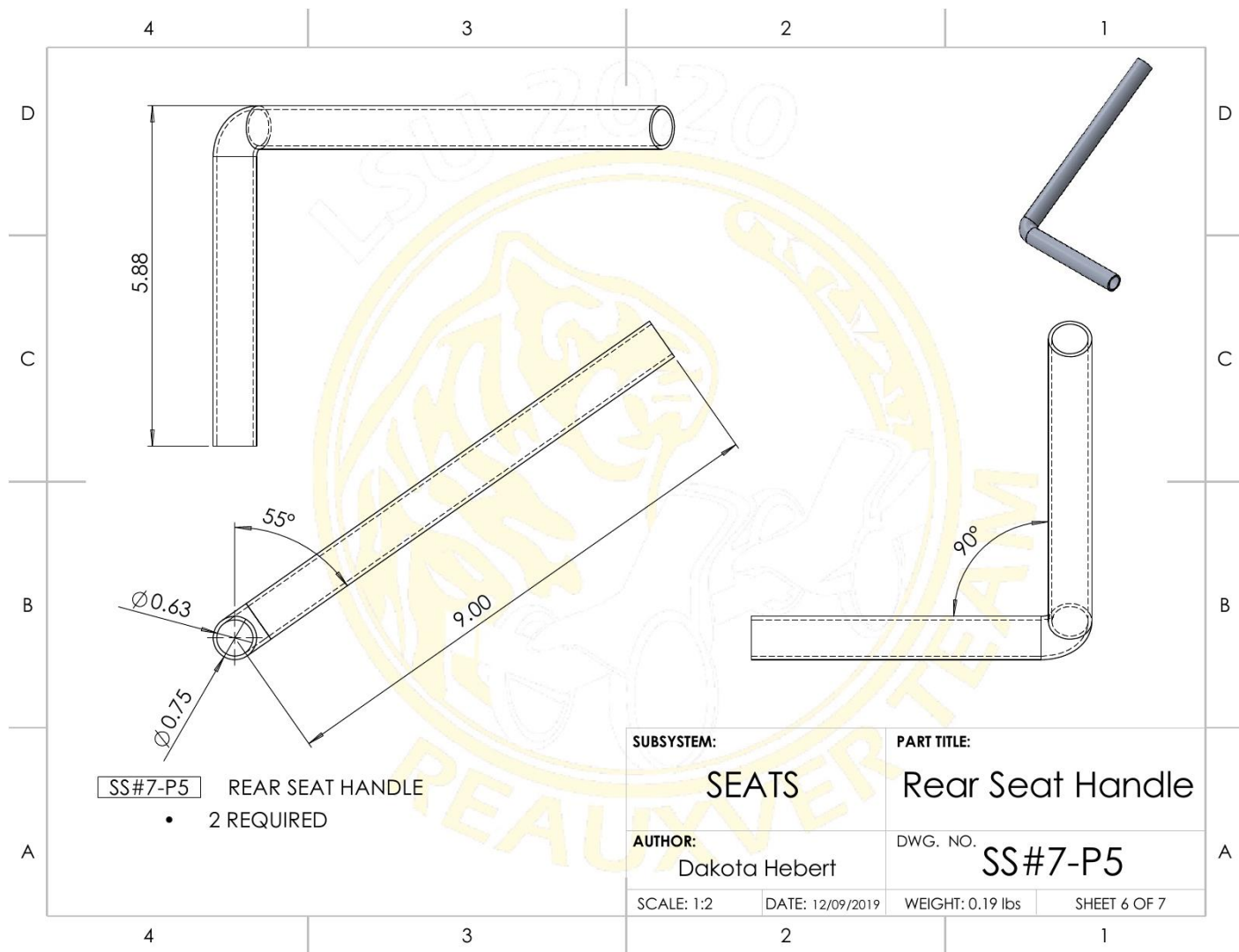


Figure XI.C.59: Manufacturing Drawing of part SS#7-P5 – Rear Seat Handle

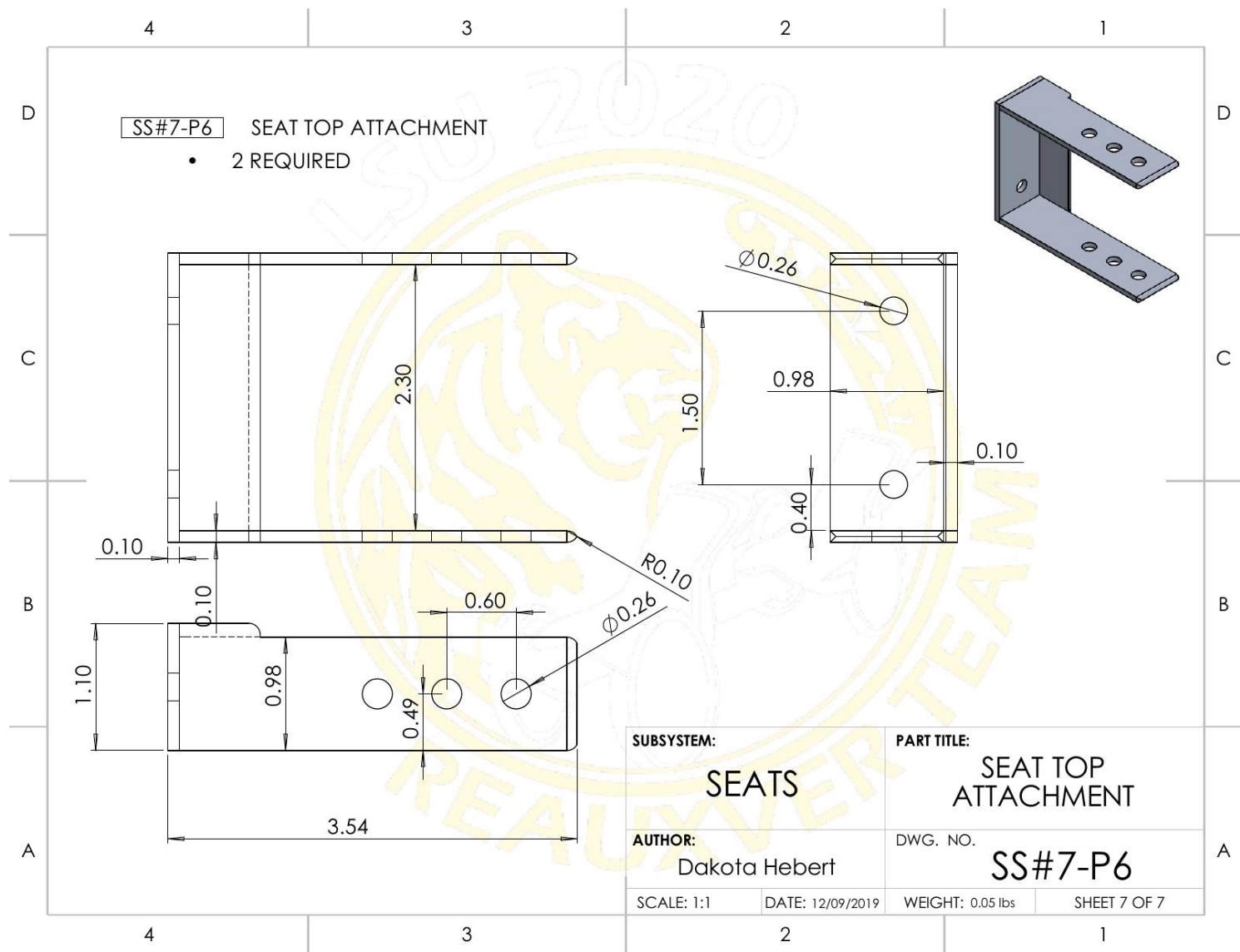


Figure XI.C.60: Manufacturing Drawing of part SS#7-P6 – Seat Top Attachment

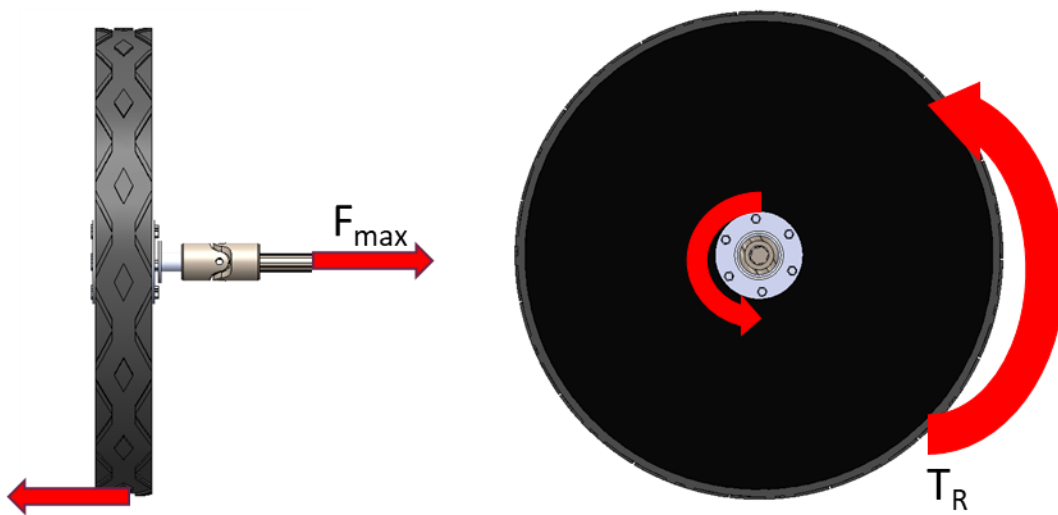
XI.D. Engineering Analysis and Materials Selection Supplement

XI.D.1. Eng. Analysis Details for SS#1- Solid Disk Wheels

XI.D.1.a. Eng. Analysis and Materials Selection Details for SS#1-P1- Solid Disk Wheels

Pham

Objective: Determine the stresses seen in the carbon fiber.



Assumptions: The weight of the rover and both the riders were held on by one wheel. The torsional stress from the drive shaft was split between two wheels. The weight of the rider and the rover was 600 lbf.

$$\sigma_b = \frac{M_{MAX}y}{I} = \frac{(600 \text{ lbf} * 12 \text{ in}) * 2.5 \text{ in}}{282.211 \text{ in}^4} = 63.78 \text{ ksi}$$

$$\frac{\tau_T}{2} = \frac{8T}{\pi d^3} = \frac{8 * 430 \text{ lbf} * (1.5 \text{ in} * .28 \text{ in})}{\pi * (1.18 \text{ in})^3} = 1.19 \text{ ksi}$$

$$\sigma_c = \frac{W_{total}}{A} = \frac{W_{total}}{(\pi * r_{hub} * t_{wheel})} = \frac{600 \text{ lbf}}{(\pi * 1.5 \text{ in} * 5 \text{ in})} = 25.5 \text{ psi}$$

Variables:

σ_b : Bending Stress on the Wheel

M_{MAX} : Maximum Bending Moment of the Wheel due to Drive Shaft Moving Horizontally

y : Distance from the Centroid of the Cross-Section of the Wheel (Half the thickness of the Wheel)

I : Moment of Inertia of the Cross-Section of the Wheel

τ_T : Torsional Shear Stress from the Drive Shaft

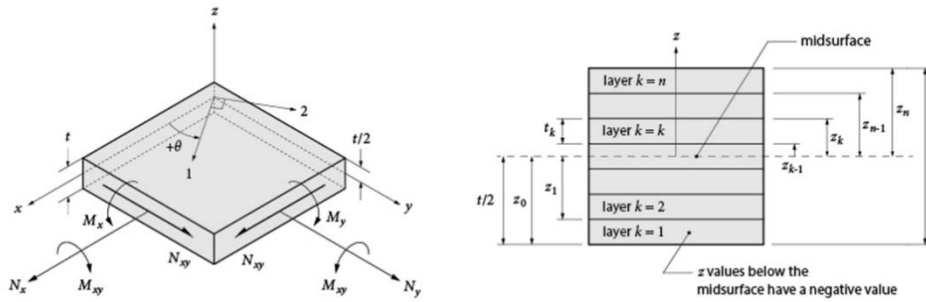
T : Torque on the Driveshaft
 d : Driveshaft Outer Diameter
 σ_c : Compressive Stress on the Wheel
 W_{total} : Total weight of the Rover and the Riders
 A : Area of Contact between the Hub and the Wheel

Results: The primary outcome of the stress analysis was that the highest stress seen in the wheel was a bending stress of **63.8 ksi**. The other two stresses caused in the wheel were a torsional stress of **1.19 ksi** and a compressive stress of **6.37 psi**. These values would then be used to calculate stress seen in different plies of the carbon fiber using classical lamination theory.

XI.D.1.b. Eng. Analysis and Materials Selection Details for SS#1-P1- Solid Disk Wheels

Pham

Objective: Determine the stress per length seen in the individual plies of the carbon fiber



[17]

Assumptions: Analysis was done based off of classical lamination theory and a perfect bond between each ply of carbon fiber [3].

$$\begin{aligned}
 N_x &= \int_{-\frac{h}{2}}^{\frac{h}{2}} \sigma_x dz = 2 \int_0^{\frac{h}{2}} -6.37 \text{ psi } dz = -0.07644 \\
 N_y &= \int_{-\frac{h}{2}}^{\frac{h}{2}} \sigma_y dz = 0 \\
 N_{xy} &= \int_{-\frac{h}{2}}^{\frac{h}{2}} \tau_{xy} dz = 2 \int_0^{0.006} 1,190 \text{ psi} * z dz = 0.08568 \frac{\text{lb} \cdot \text{f}}{\text{in}} \\
 M_x &= \int_{-\frac{h}{2}}^{\frac{h}{2}} \sigma_x z dz = 2 \int_0^{0.006} 63,780 \text{ psi} * z dz = 4.592 \frac{\text{lb} \cdot \text{f} \cdot \text{in}}{\text{in}} \\
 M_y &= \int_{-\frac{h}{2}}^{\frac{h}{2}} \sigma_y z dz = 0, \quad M_{xy} = \int_{-\frac{h}{2}}^{\frac{h}{2}} \tau_{xy} z dz = 0
 \end{aligned}$$

Variables:

N_x : Resultant Force in x-axis
 σ_x : Normal Stress in x-axis
 N_y : Resultant Force in y-axis
 σ_y : Normal Stress in y-axis
 N_{xy} : Resultant Force in xy-axis
 τ_{xy} : Shear Stress in xy-axis
 M_x : Resultant Moment in x-axis
 σ_x : Bending Stress in x-axis
 M_y : Resultant Moment in y-axis
 σ_y : Bending Stress in y-axis
 M_{xy} : Resultant Moment in xy-axis
 τ_{xy} : Shear Stress between Sheets

Stress in Each Ply of Carbon Fiber Sheet

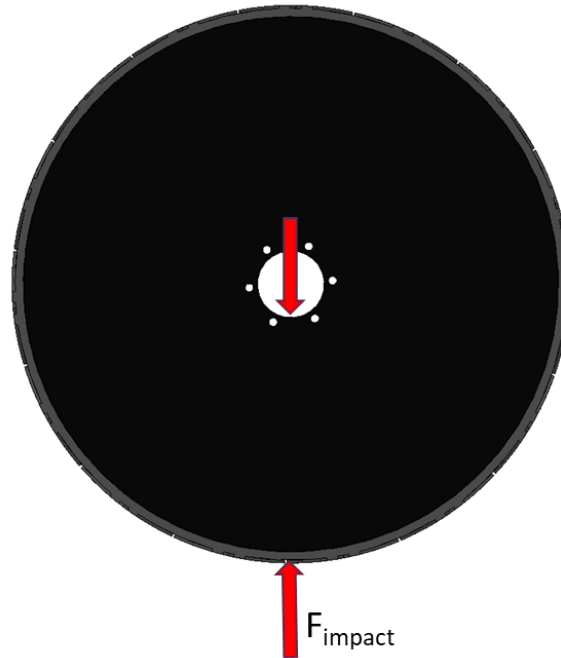
Layer	Orientation	σ_x (psi)	σ_y (psi)	σ_{xy} (psi)
1	0°	-11,657	-100	-381
2	45°	4,400	569	1,385
3	90°	7,210	-469	-997

Results: The primary outcome of the validation is that the stresses seen in different plies were below the strength of the composite in the longitudinal and transverse plane by orders of magnitude. The highest stress seen was **-11.7 ksi** in the first ply and **7.2 ksi** in the third ply compared to a longitudinal stress of carbon fiber at **1500 Msi**.

XI.D.1.c. Eng. Analysis and Materials Selection Details for SS#1-P1- Solid Disk Wheels

Pham

Objective: Determine the stresses of the core seen at impact of the 2019 rover's wheel.



Assumptions: The impact force was applied on one of the wheels radially towards the center. The fibers near the hole of the wheel were cut during manufacturing and therefore the core component of the wheel takes all of the impact force.

$$A_p = d_{\text{hub}} * t_{\text{wheel}} = 3 \text{ in} * 3 \text{ in} = 9 \text{ in}^2$$
$$\sigma_c = \frac{F_{\text{impact}}}{A_p} = \frac{1400 \text{ lbf}}{9 \text{ in}^2} = \mathbf{160 \text{ psi}}$$

Variables:

A_p : Projected Area of Contact of Hub to Wheel

t_{wheel} : Thickness of the 2019 rover's Wheels

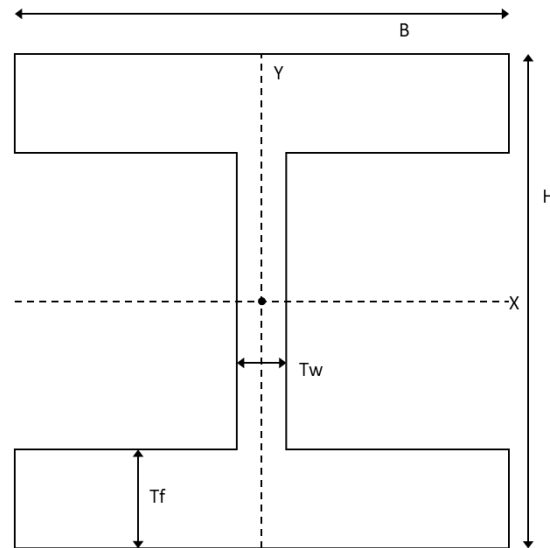
d_{hub} : Diameter of the 2019 rover's Hub

σ_c : Compressive Stress Seen in the Core Component

F_{impact} : Force of Impact

Results: The primary outcome of the stress analysis at impact on the core component was to see if the core component of the 2019 rover's wheel would yield on impact due to utilizing a low-density polyurethane foam. It was concluded that an impact force of 1400 lbf will produce a compressive stress in the core of **160 psi**; this is higher than the compressive strength of low-density polyurethane foam (5.6 lbf/ft³) of **130 psi** [2]. A balsa wood core would provide a compressive strength of **950 psi** but at a slightly higher density of 6.5 lbf/ft³.

Objective: Determine the Moment of Inertia of the Cross-Section of the 2020 Rover's Wheel



Assumptions: The cross-section of a composite material mimics that of an I-beam cross section where the flanges are the composite materials and the web is the core component. The flanges take most of the load, while the web provides thickness and stiffness.

$$n = \frac{E_{Carbon\ Fiber}}{E_{Balsa\ Wood}} = \frac{33,600\ ksi}{295.36\ ksi} = 113.76$$

$$B = n * 5\ in = 568.8\ in$$

$$H = 5\ in$$

$$T_f = 3 * 0.012\ in = 0.036\ in$$

$$T_w = 3\ in$$

$$I_x = 282.211\ in^4$$

Variables:

n : Modular Ratio

$E_{Carbon\ Fiber}$: Young's Modulus of Carbon Fiber

$E_{Balsa\ Wood}$: Young's Modulus of Balsa Wood

B : Width of the Flanges after Transformation

H : Width of the wheel

T_f : Thickness of 3-Ply Carbon Fiber

T_w : Width of the hub

I_x : Moment of Inertia of the Cross-Section of the Wheel

Results: The primary outcome of the moment of inertia calculation is a moment of inertia of **282.2 in⁴**. This moment of inertia was used in bending analysis as well as buckling analysis for the wheel.

Objective: Determine the Buckling Force of the Wheel

Assumptions: The wheel met the conditions to use Euler column buckling load.

$$P_{cr} = \frac{\pi^2 EI}{L_e^2} = \frac{\pi^2 * 10,413,300 * 282.211}{(12 * 2)^2} = 50 \text{ Msi}$$

Variables:

P_{cr} : Buckling Load

E : Effective Young's Modulus for the Sandwich Structure Composite Wheel

I : Moment of Inertia of the Cross-Section of the Wheel

L_e : Effective Length of the Column (Free-Fixed End)

Results:

XI.D.2. Eng. Analysis Details for SS#2- Single Bar Frame

Objective: Determine material for the frame and associated components

Assumptions: The design requirement is a light & stiff beam

Free Variable: A

Objective Function: $m = \rho V = \rho LA$

Constraint: $F = S\delta$

$S \geq S^$*

$$S = \frac{cEI}{L^3} = \frac{cEb^4}{12L^3} \geq S^*$$

Replace Free Variable: $A^2 \geq \frac{12SL^3}{cE}$

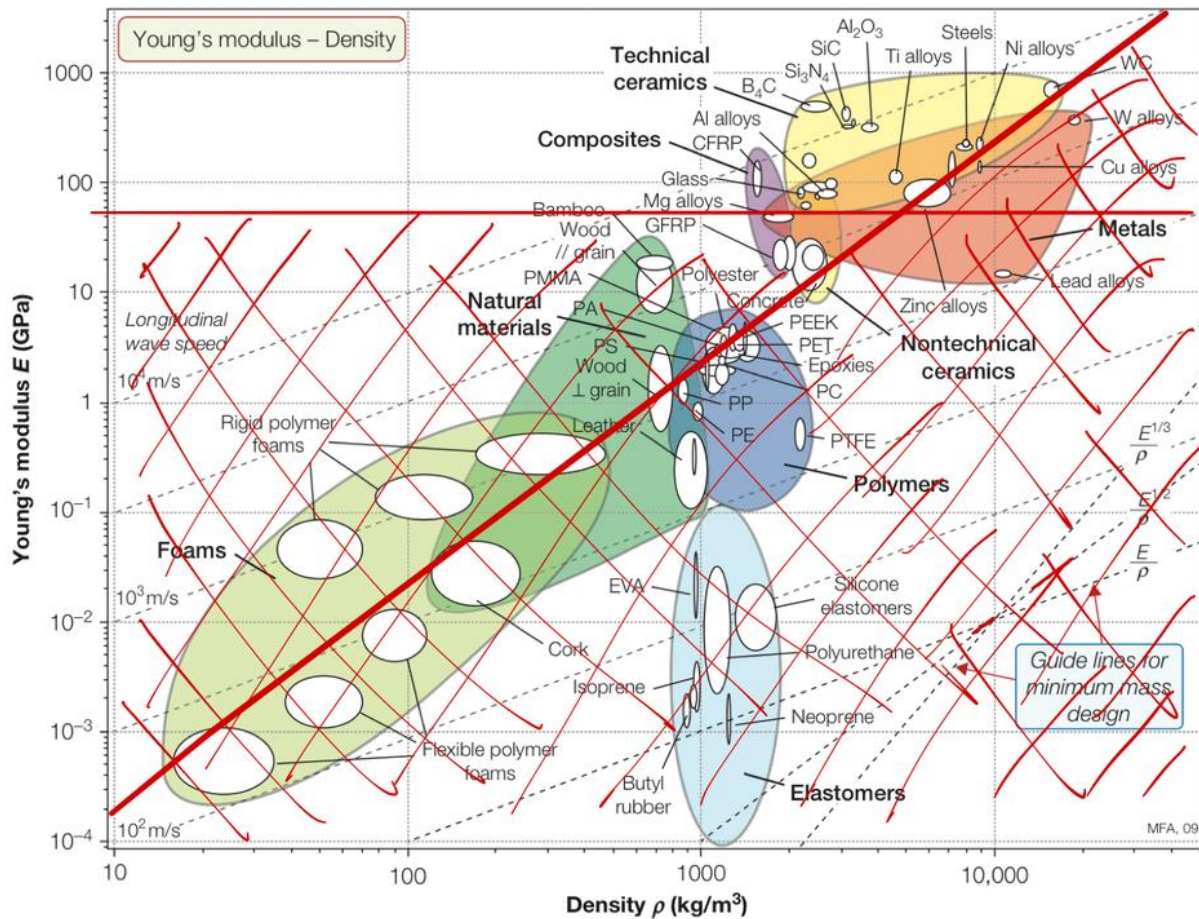
$$m = \rho L \sqrt{\frac{12SL^3}{cE}}$$

Material Index: $\frac{\sqrt{E}}{\rho}$

Want to Maximize $\frac{\sqrt{E}}{\rho}$ to minimize the mass

Variables:

- A : Cross-sectional Area
 E : Young's Modulus
 I : Moment of Inertia of the Cross-Section
 L : Effective Length
 ρ : Density
 S : Stiffness
 c : Centroid
 δ : Deflection



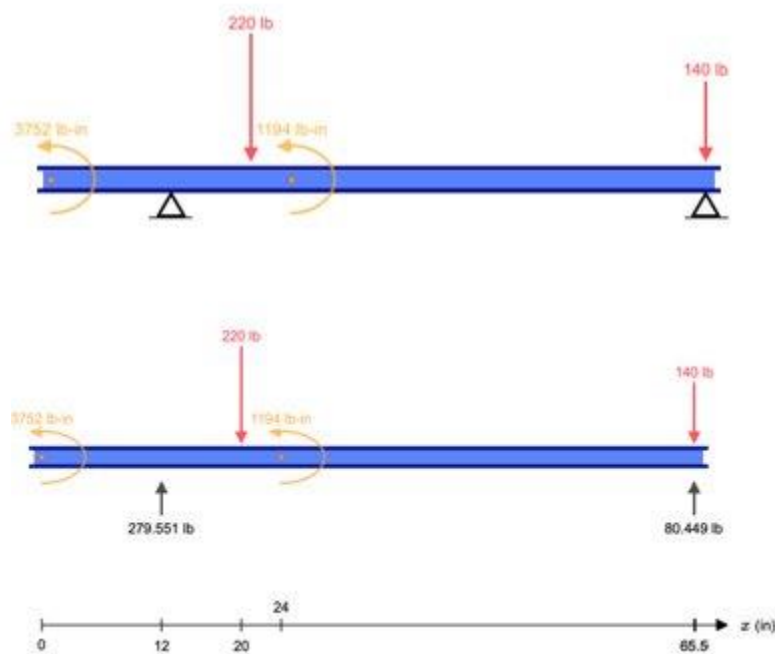
Result: The material index led to using a decision matrix between carbon fiber, steel or aluminum alloys. It is important to note that a different cross-sectional shape would not lead to a different material index as long as the shape is self-similar when area is changed.

XI.D.2.b. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the maximum stresses induced by static loading and ensure the frame will not yield

Assumptions: When the rover is not moving the frame acts as a simply supported beam (All frame loading analysis was run with SkyCiv Software. A software performance check was completed and can be found in the following analysis for impact.)



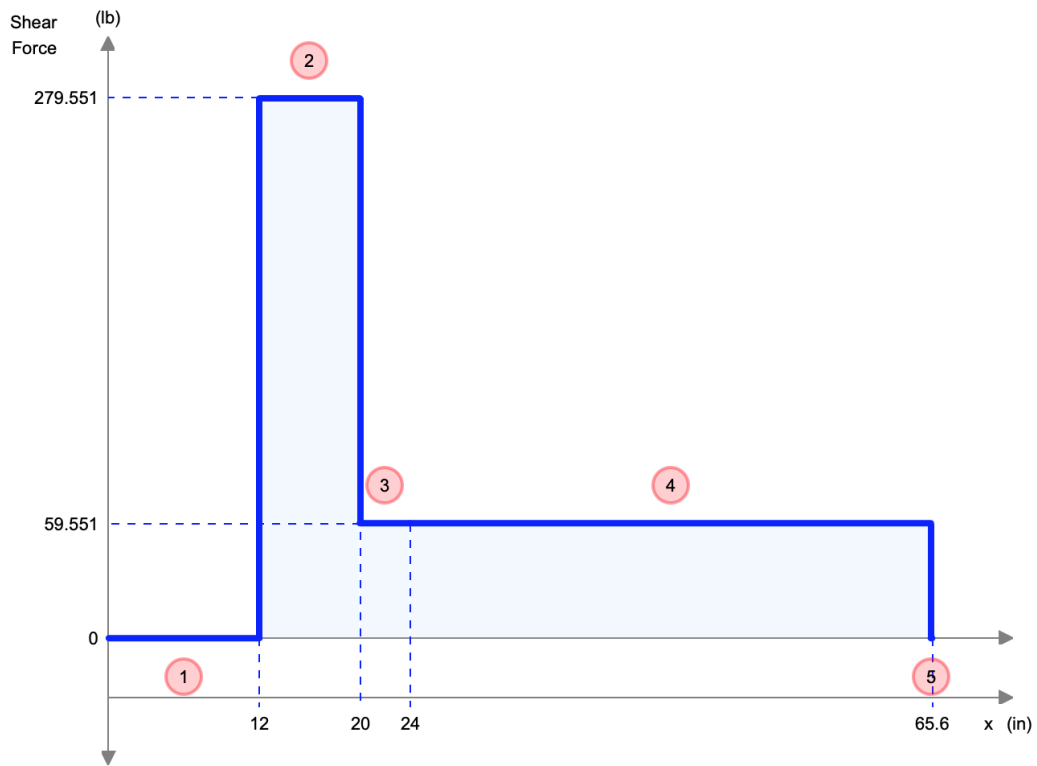
$$\begin{aligned}
 & + \uparrow \Sigma F_y = 0 \\
 & R_A + R_B - 220 - 140 = 0 \\
 & R_A + R_B - 220 - 140 = 0 \\
 & R_A + R_B = +360 \text{ lb} \quad \text{---- (1)} \\
 & + \circlearrowleft \Sigma M = 0 \\
 & R_B (65.5 - 12) + (-220) (20 - 12) + (-140) (65.5 - 12) + 3752 + 1194 = 0 \\
 & R_B (65.5 - 12) + (-220) (+8) + (-140) (+53.5) + 3752 + 1194 = 0 \\
 & R_B = +80.449 \text{ lb} \quad \text{---- (2)}
 \end{aligned}$$

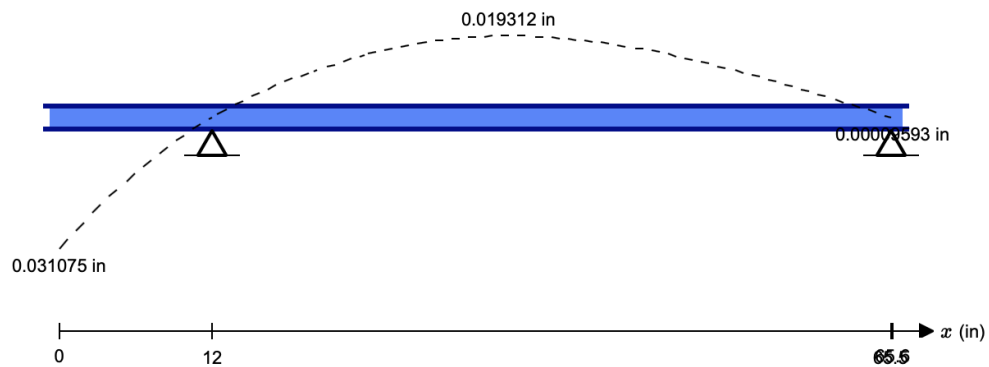
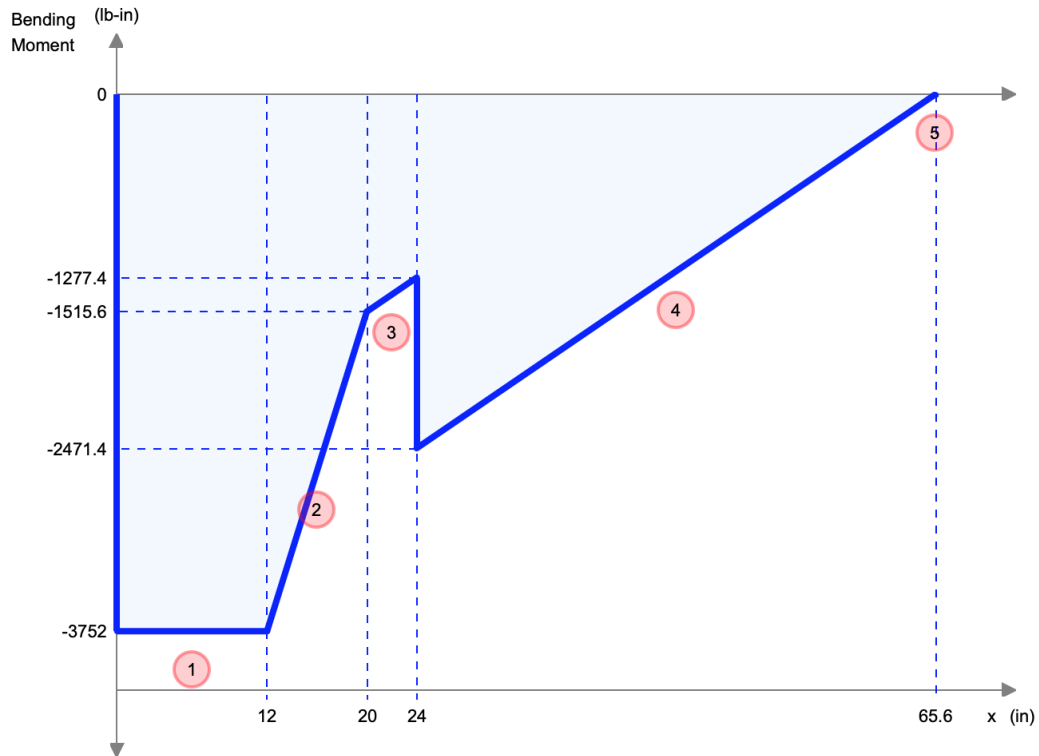
Substitute (2) into (1):

$$R_A + R_B = +360$$

$$R_A + 80.449 = +360$$

$$R_A = +279.551\text{lb}$$





Stress Units: psi	Type	Stress Value (psi)	Locations along Beam (in)	Y location on Beam Section (in)
Overall Maximum Transverse Shear Stress	Positive (Upward)	350.406	-	2
	Negative (Downward)	-	-	(Neutral Axis)
Overall Maximum Normal Bending Stress	Tension	2653.278	0	4
	Compression	2653.278	0	0
				(Top of Section)
				(Bottom of Section)

Variables:

R_A : Reaction at front of rover

R_B : Reaction at rear of rover

Result: $\sigma_{Max} = 2.65 \text{ ksi} \ll S_y = 40,000 \text{ psi}$ therefore risk of yield during static loading is not a concern.

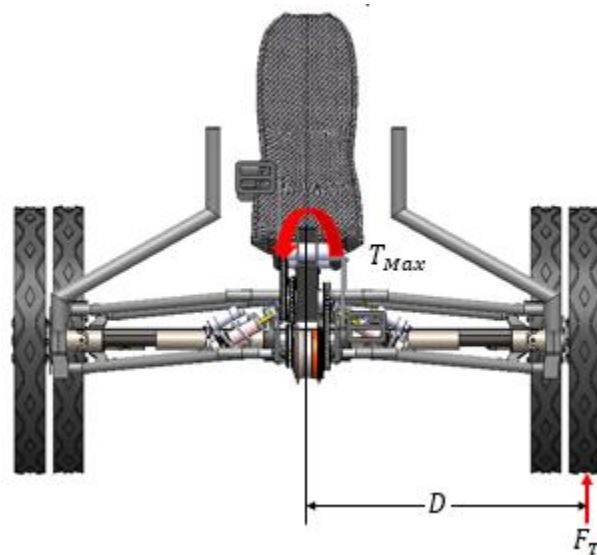
XI.D.3.

XI.D.3.a. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the maximum stresses induced by impact loading and ensure the frame will not yield

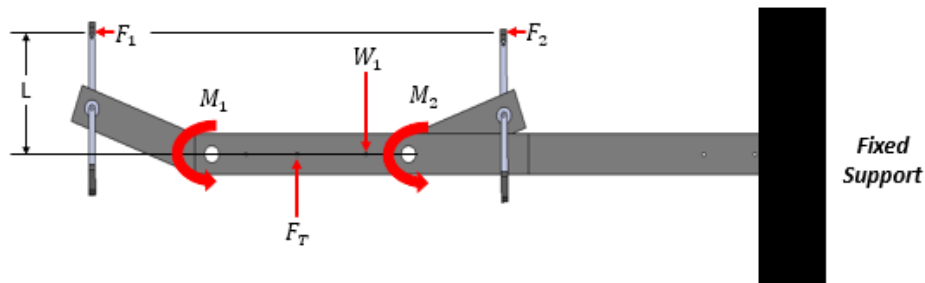
Assumptions: Assuming rear rider acts as a fixed support for worst case, most conservative scenario.



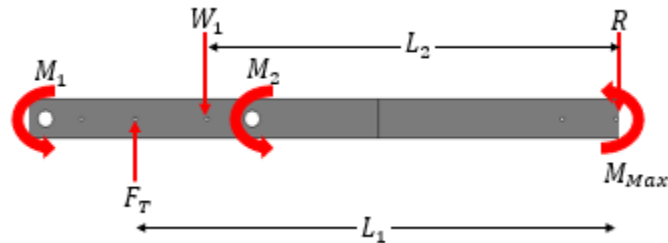
$$F_T = F_{Impact} - F_{Suspension} = 1400 \text{ lbf} - 954 \text{ lbf} = 406 \text{ lbf}$$

$$T_{Max} = DF_T = (30 \text{ in})(406 \text{ lbf}) = 12180 \text{ in} - \text{lbf}$$

$$\tau_{Max} = \frac{T_{Max}}{2t(a-t)(b-t)} = \frac{12180 \text{ in} - \text{lbf}}{2(.125 \text{ in})(4 \text{ in} - .125 \text{ in})(2 - .125)} = 6.71 \text{ ksi}$$



$$\begin{aligned}
 F_1 &= 330 \text{ lbf} \\
 F_2 &= 105 \text{ lbf} \\
 W_1 &= 220 \text{ lbf} \\
 M_1 &= LF_1 = (11.5 \text{ in})(330 \text{ lbf}) = 3795 \text{ in} - \text{lbf} \\
 M_2 &= LF_2 = (11.5 \text{ in})(105 \text{ lbf}) = 1208 \text{ in} - \text{lbf}
 \end{aligned}$$



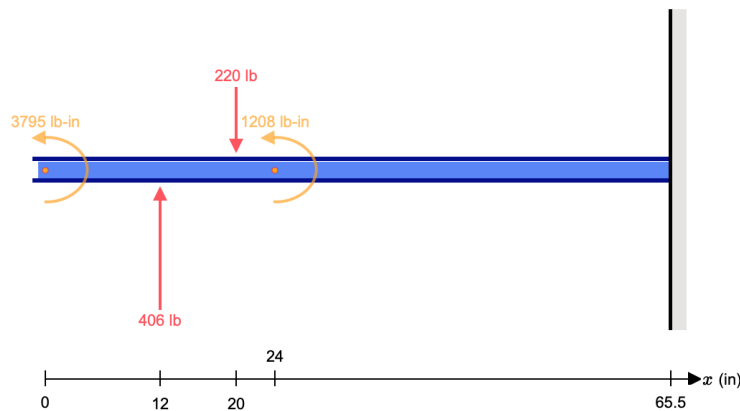
$$\begin{aligned}
 R &= F_T - W_1 = 406 \text{ lbf} - 220 \text{ lbf} = 186 \text{ lbf} \\
 M_{Max} &= -M_1 + L_1 F_T - L_2 W_1 - M_2 \\
 &= -3795 \text{ in} - \text{lbf} + (53.5 \text{ in})(406 \text{ lbf}) - (45.5 \text{ in})(220 \text{ lbf}) - 1208 \text{ in} - \text{lbf} \\
 &= 6708 \text{ in} - \text{lbf}
 \end{aligned}$$

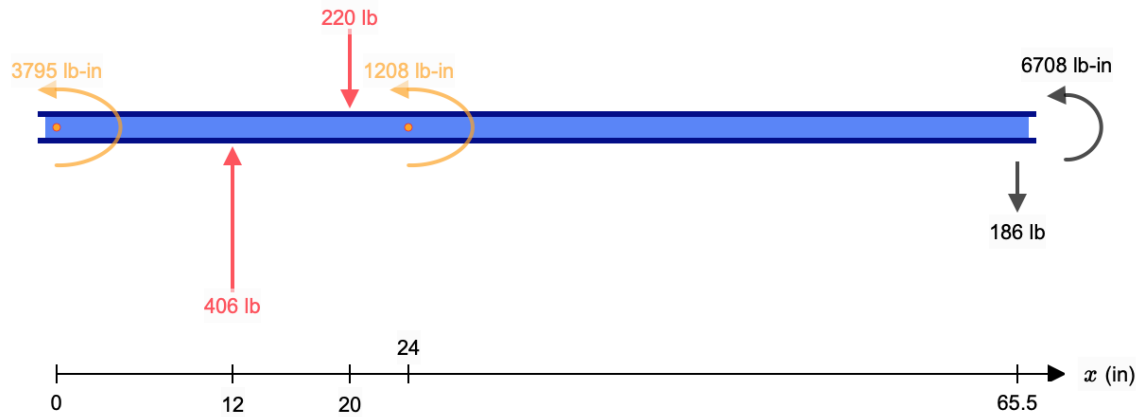
$$\sigma_B = \frac{M_{Max} y}{I} = \frac{(6708 \text{ in} - \text{lbf})(2 \text{ in})}{2.8282 \text{ in}^4} = 4.74 \text{ ksi}$$

$$\sigma_{V.M} = \sqrt{\sigma_B^2 + 3\tau_{max}^2} = \sqrt{(4.74 \text{ ksi})^2 + 3(6.71 \text{ ksi})^2} = 12.55 \text{ ksi}$$

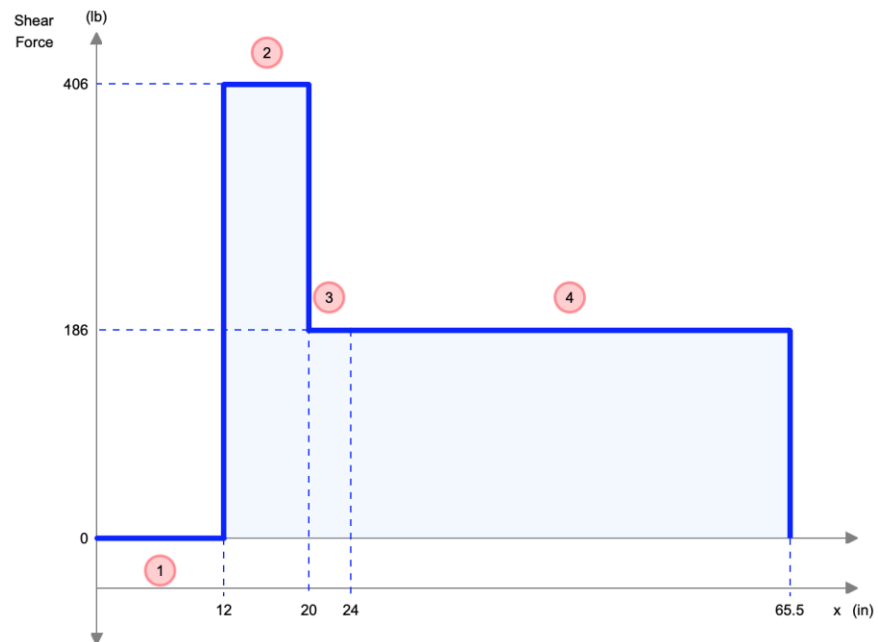
$$SF = \frac{S_Y}{\sigma_{V.M}} = \frac{40,000}{12,550} = 3.19$$

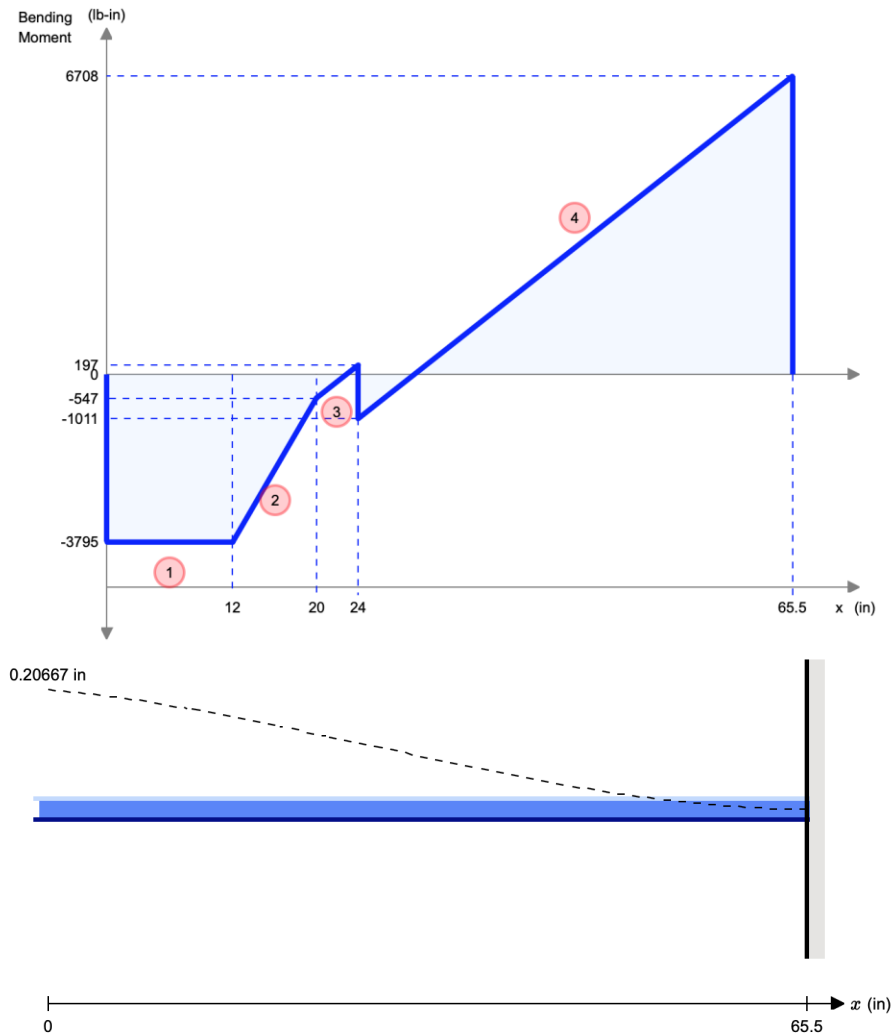
Software Answers matching theoretical values:





$$\begin{aligned}
 + \uparrow \Sigma F_y &= 0 \\
 R_y - 220 + 406 &= 0 \\
 R_y - 220 + 406 &= 0 \\
 R_y &= -186 \text{ lb} \\
 + \circlearrowleft \Sigma M &= 0 \\
 R_M + (-220)(65.5 - 20) + (-406)(65.5 - 12) + 3795 + 1208 &= 0 \\
 R_M + (+220)(+45.5) + (-406)(+53.5) + 3795 + 1208 &= 0 \\
 R_M &= +6708 \text{ lb-in}
 \end{aligned}$$





Stress Units: psi	Type	Stress Value (psi)	Locations along Beam (in)	Y location on Beam Section (in)
Overall Maximum Transverse Shear Stress	Positive (Upward)	508.905	-	2
	Negative (Downward)	-	-	(Neutral Axis)
Overall Maximum Normal Bending Stress	Tension	4743.653	65.5	0
	Compression	4743.653	65.5	4
				(Bottom of Section)
				(Top of Section)

Variables:

F_T : Total force seen by frame after suspension absorption

F_{Impact} : Maximum force from impact

T_{Max} : Max torsion seen by frame

τ_{Max} : Maximum torsional stress induced on frame

F_1 : Force input from rider one

F_2 : Force input from rider two

W_1 : Weight of rider one
 L : Length from rider force input to center of frame
 M_1 : Moment of rider one pedaling input
 M_2 : Moment of rider two pedaling input
 L_1 : Length between total force and fixed support
 L_2 : Length between weight and fixed support
 M_{Max} : Maximum moment induced on frame
 R : Maximum reaction at fixed support
 σ_B : Maximum bending stress induced on frame
 $\sigma_{V.M}$: Maximum stress induced on frame using Von Mises criteria for bending and torsion
 SF : Safety factor with respect to yield
 S_Y : Yield strength for Aluminum 6061-T6

Result: With the worst-case impact scenario the frame still has a safety factor of 3.19 with respect to yield. Team 24 believes the frame is adequately designed and capable of being durable throughout testing and competition.

XI.D.3.b. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the maximum life of the frame during worst case scenario impact

Assumptions: Assuming fully reversed bending and torsion for worst case, most conservative analysis

Factors	Bending	Torsion
C_L	1	0.58
C_G	0.8	0.8
C_S	0.7	0.7
S	4744 psi	6700 psi
S_U	45,000 psi	45,000 psi
f	0.9	0.72
S'_n	$0.5S_U$	$0.29S_U$

$$S_n = C_L C_G C_S S'_n$$

$$S_n \text{ for Bending} = 12,600 \text{ psi} \quad f S_U \text{ for Bending} = 40,500 \text{ psi}$$

$$S_n \text{ for Torsion} = 4,239 \text{ psi} \quad f S_U \text{ for Torsion} = 32,400 \text{ psi}$$

For Bending $S \leq S_n$ therefore Indefinite Life

For Torsion $S_n \leq S \leq S_U$ therefore High Life

$$\text{For Torsion } N = \left[\frac{SS_n}{(fS_U)^2} \right]^{\frac{3}{\log(\frac{S_n}{fS_U})}} = 211,228 \text{ cycles to failure}$$

Variables:

C_L : Load factor
 C_G : Gradient factor
 C_S : Surface factor
 S : Load applied
 S_U : Ultimate strength
 S_n : Fatigue strength
 f : Fraction of ultimate strength
 S'_n : Result from Moore Test

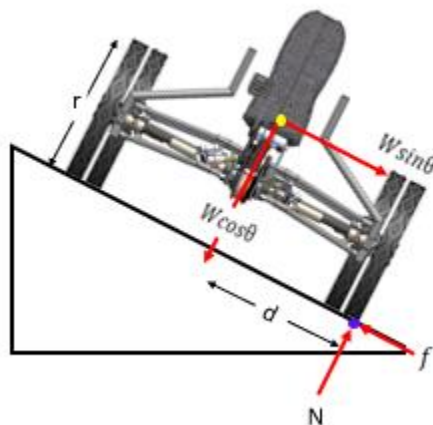
Result: Assuming a worst-case scenario of fully reversed bending and torsion during maximum impact loading. Using the maximum bending stress of $\sigma_{Max} = 4.74 \text{ ksi}$ and the maximum torsional stress of $\tau_T = 6.71 \text{ ksi}$. For bending the frame has **indefinite life**. For torsion the frame has a life of **211,228 cycles to failure**.

XI.D.3.c. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the maximum height of the center of gravity for the side to side tipping analysis on a 30° incline

Assumptions: Assuming no slip



$$\sum M_p = 0 = W \cos \theta * d - W \sin \theta * r_{max}$$

$$r_{max} = \frac{\cos \theta * d}{\sin \theta}$$

Maximum height of COG at 30° = 4.33 ft

Rover height from ground = 3.64 ft

Maximum height of COG at 30° > Rover height from ground therefore no tipping occurs

Variables:

M_p : Moment about point
 W : Rover weight

d : rover wheel base
 θ : high butte incline angle
 f : Friction
 N : Normal force
 r : rover wheel diameter

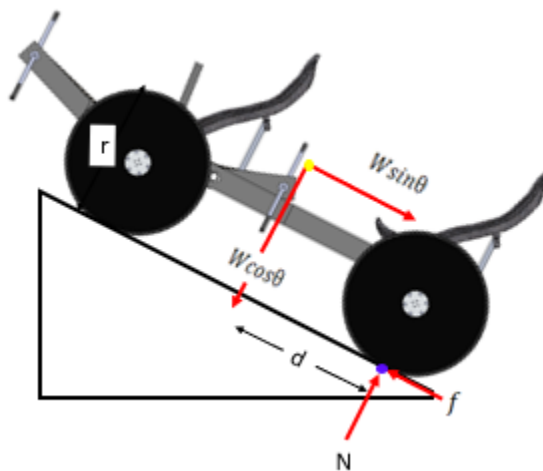
Result: The maximum allowable height of the center of gravity of the rover on a 30° incline is $H_{COG} = 4.33 \text{ ft}$ when the rover is in a side to side configuration. This exceeds the height of the rover from the ground of $H_{Rover} = 3.64 \text{ ft}$ therefore tipping in the side to side configuration is not a concern.

XI.D.3.d. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the maximum height of the center of gravity for the front to back tipping analysis on a 30° incline

Assumptions: Assuming no slip



$$\sum M_p = 0 = W \cos \theta * d - W \sin \theta * r_{max}$$

$$r_{max} = \frac{\cos \theta * d}{\sin \theta}$$

Maximum height of COG at $30^\circ = 5.20 \text{ ft}$

Rover height from ground = 3.64 ft

Maximum height of COG at $30^\circ >$ Rover height from ground therefore no tipping occurs

Variables:

M_p : Moment about point
 W : Rover weight
 d : rover wheel base
 θ : high butte incline angle
 f : Friction
 N : Normal force
 r : rover wheel diameter

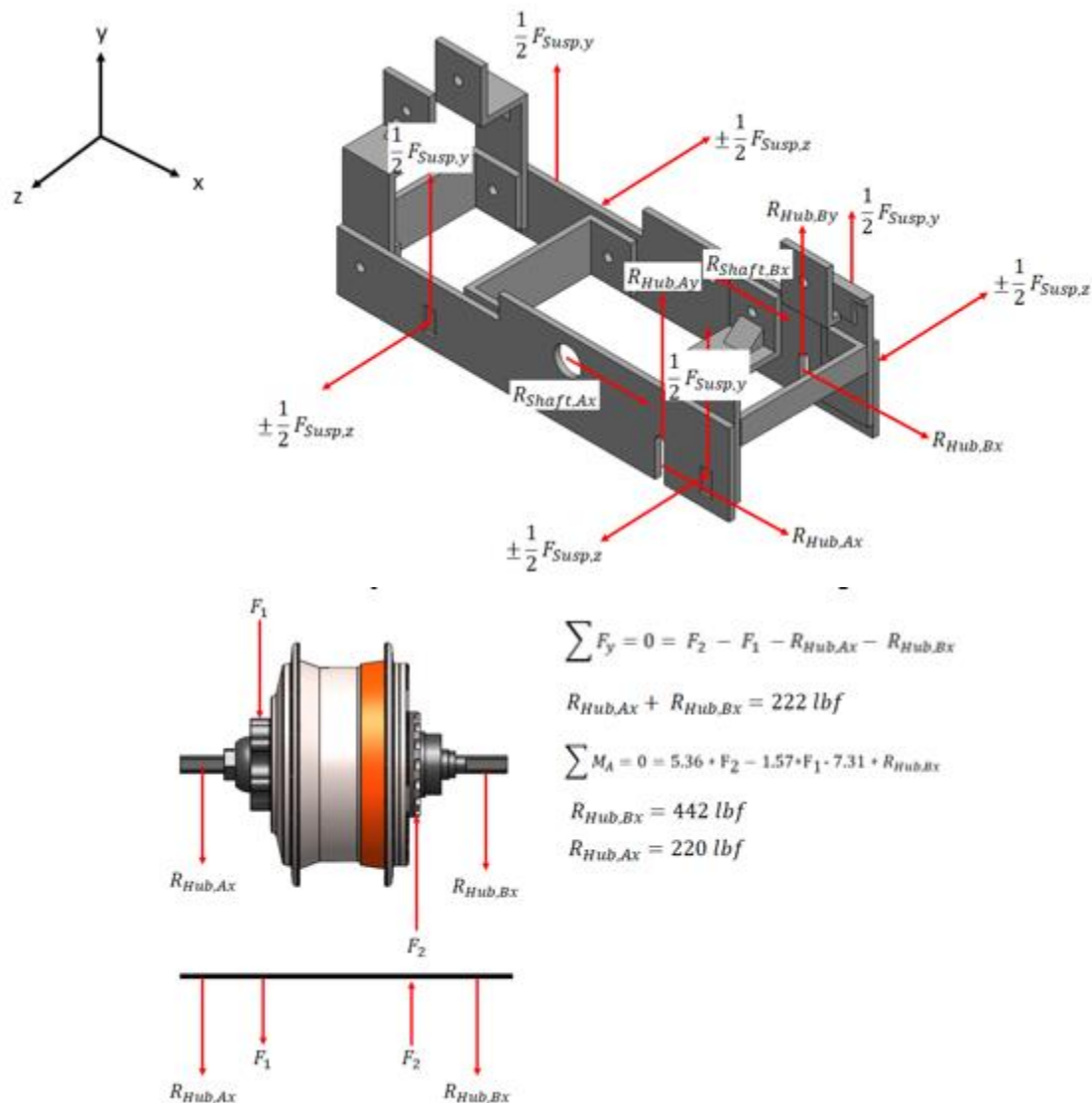
Result: The maximum allowable height of the center of gravity of the rover on a 30° incline is $H_{COG} = 5.20 \text{ ft}$ when the rover is in a front to back configuration. This exceeds the height of the rover from the ground of $H_{Rover} = 3.64 \text{ ft}$ therefore tipping in the front to back configuration is not a concern.

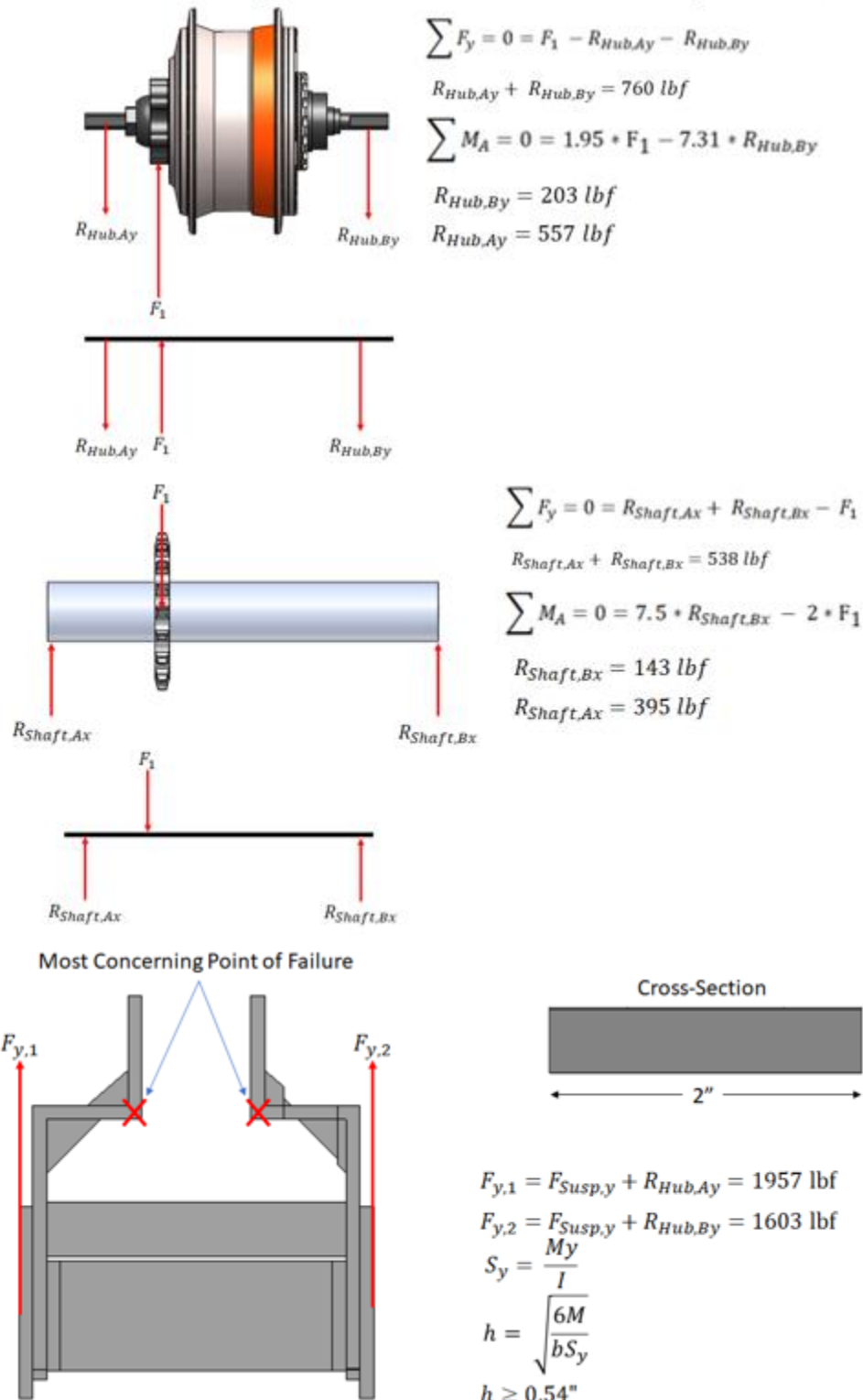
XI.D.3.e. Eng. Analysis and Materials Selection Details for SS#2-P1-12 - Frame

Smith

Objective: Determine the minimum thickness of the bracketing assemblies

Assumptions: Assuming worst case from impact load and highest chain tension





Variables:

$F_{Susp,y}$: Suspension force in the vertical direction

$F_{Susp,z}$: Suspension force in the horizontal direction

$R_{Shaft,Ax}$:	Shaft force reaction at A
$R_{Shaft,Bx}$:	Shaft force reaction at B
$R_{Hub,Ax}$:	Hub force parallel to ground at A
$R_{Hub,Bx}$:	Hub force parallel to ground at B
$R_{Hub,Ay}$:	Hub force vertical at A
$R_{Hub,By}$:	Hub force vertical at B
h :	Minimum height required at failure point

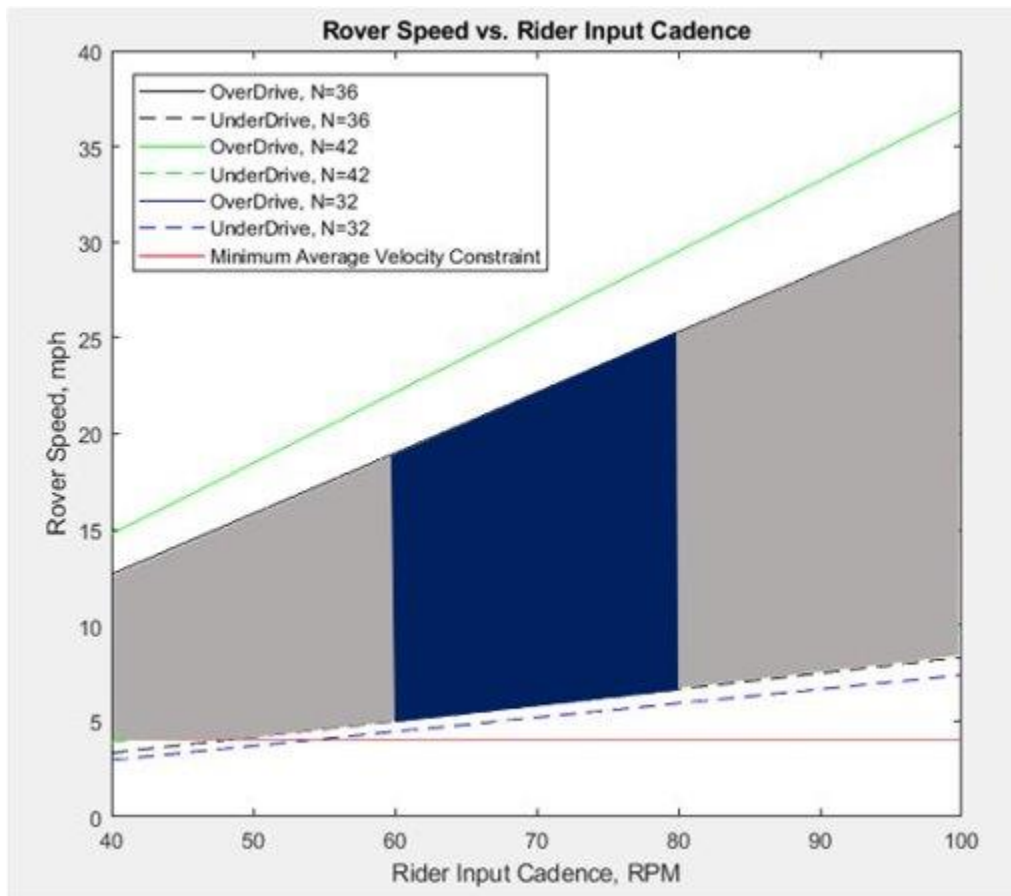
Result: The worst-case scenario is found during impact with the force of the chain pulling up on the internally geared hub. The maximum force seen by the bracketing is in the front drivetrain area and was found to be $F_{Max} = 957 \text{ lbf}$ in the vertical direction. This creates a maximum moment of $M_{Max} = 1.91 \text{ in} - \text{kip}$. The minimum allowable thickness of the bracketing is $t_{min} = 0.387 \text{ in}$ at the most concerning point of failure.

XI.D.4. Eng. Analysis Details for SS#3- Front Tandem Drivetrain

XI.D.4.a. Eng. Analysis and Materials Selection Details for SS#3-P1, P2, P9 – Surly 36T/18T Cog & NuVinci N380 Internally-Geared Hub

Prejean

Objective: Determine the relationship between rider input cadence and variable cog sizes to output rover speed.



Assumptions: Negligible energy loss through the drivetrain system.

Mathematical Model:

$$\text{Output Rover Speed (mph)} = \text{Gear Multiplier} * \text{Upstream Gear Ratio} * \text{Downstream Gear Ratio} * \text{Input Cadence (RPM)} * 1 \left(\frac{\text{rev}}{\text{min}} \right) * 2\pi \left(\frac{\text{rad}}{\text{rev}} \right) * \frac{1}{60} \left(\frac{\text{min}}{\text{sec}} \right) * r * 0.0254 \left(\frac{\text{meters}}{\text{inch}} \right) * 2.23694 \left(\frac{\text{mph}}{\frac{\text{m}}{\text{s}}} \right)$$

Variables:

Gear Multiplier:
1.9

Upstream Gear Ratio:

Downstream Gear Ratio:

Input Cadence (RPM):

r:

Range of gear ratios from internally geared hub, range: 0.5-

$N_{driving}/N_{driven}$ upstream of N380 Internally-Geared hub

$N_{driving}/N_{driven}$ downstream of N380 Internally-Geared hub

Rider Input Cadence to Rover

Rover Wheel Radius (14")

```

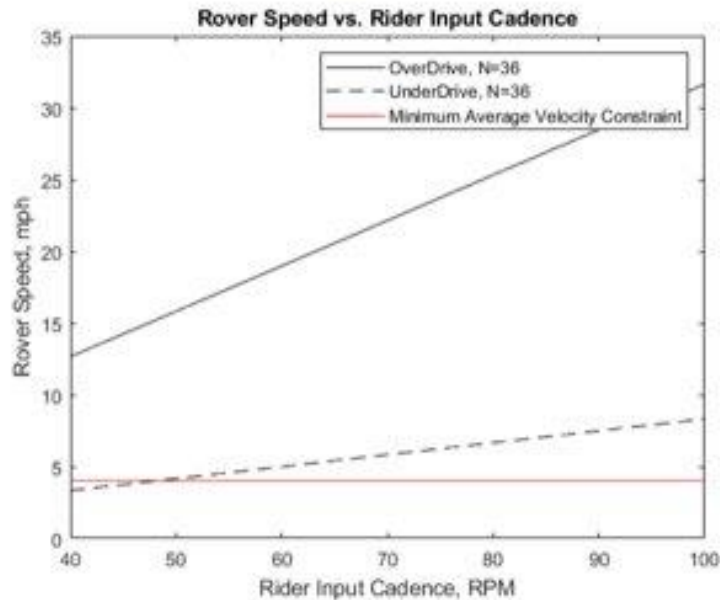
clear:clc;
%Range of Input Cadences
x = {40:1:100};
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Driving Gear # of teeth (relative to N380)
driving = 36;
%Driven Gear # of teeth (relative to N380)
driven = 18;
%Driving Gear # of teeth (relative to Axle)
driving2 = 18;
%Driven Gear # of teeth (relative to Axle)
driven2 = 18;
y1 = x.*2*3.14/60*.3556*2.23694*1.9*driving/driven*driving2/driven2;
y2 = x.*2*3.14/60*.3556*2.23694*0.5*driving/driven*driving2/driven2;
plot(x,y1,'k');
hold on
plot(x,y2, 'k--');
hold on
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Driving Gear # of teeth (relative to N380)
driving3 = 42;
%Driven Gear # of teeth (relative to N380)
driven3 = 18;
%Driving Gear # of teeth (relative to Axle)
driving4 = 18;
%Driven Gear # of teeth (relative to Axle)
driven4 = 18;
y6 = x.*2*3.14/60*.3556*2.23694*1.9*driving3/driven3*driving4/driven4;
y7 = x.*2*3.14/60*.3556*2.23694*0.5*driving3/driven3*driving4/driven4;
%plot(x,y6,'r');
hold on
%plot(x,y7,'r--');
hold on
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Driving Gear # of teeth (relative to N380)
driving5 = 32;
%Driven Gear # of teeth (relative to N380)
driven5 = 18;
%Driving Gear # of teeth (relative to Axle)
driving6 = 18;
%Driven Gear # of teeth (relative to Axle)
driven6 = 18;
%y8 = x.*2*3.14/60*.3556*2.23694*1.9*driving5/driven5*driving6/driven6;
%y9 = x.*2*3.14/60*.3556*2.23694*0.5*driving5/driven5*driving6/driven6;
%plot(x,y8,'b');
%hold on
%plot(x,y9,'b');
%hold on

```

```

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Mike's Average Cadence based on Testing
xmike(n) = 76.1;
%Lucy's Average Cadence based on Testing
xlucy(n) = 70.0;
end
for n = [1:61]
    zz(n) = 4;
end
plot(x,zz,'r')
%plot(xmike,y4)
%plot(xlucy,y4)
xlabel("Rider Input Cadence, RPM")
ylabel("Rover Speed, mph")
%legend("OverDrive, N=36", "UnderDrive, N=36", "OverDrive, N=42", "UnderDrive, N=42", "Minimum Average Velocity Constraint")
legend("OverDrive, N=36", "UnderDrive, N=36", "Minimum Average Velocity Constraint")
title('Rover Speed vs. Rider Input Cadence')

```



Published with MATLAB® R2019a

Result: By selecting a 36T cog size, the rover's speed range was constrained to the grey region of the plot. The model was further constrained by integrating the average cadence of the team's riders over an eight-minute interval (60-80 RPM). The blue region corresponds to the range of speeds with both constraints applied. The rover's average output speed will be between **5-17 mph**.

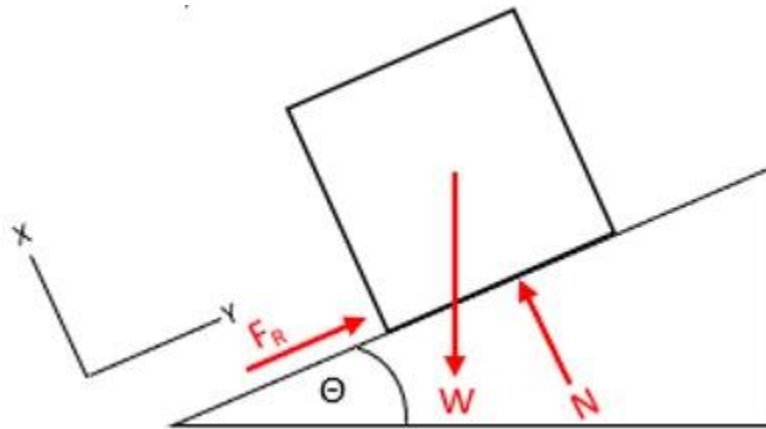
XI.D.5.

XI.D.5.a. Eng. Analysis and Materials Selection Details for SS#3-P1, P2 Surly 36T/18T Cogs

Prejean

Objective: Determine the required low gear ratio to traverse the High Butte Obstacle.

High-Butte Obstacle FBD



Assumptions: Negligible frictional loss due to surface material. Uniform incline of 20° across the entirety of the obstacle's incline.

$$W = (m_{\text{rover}} + m_{\text{riders}})g = 230 \text{ lbf} + 370 \text{ lbf} = 600 \text{ lbf}$$

$$F_R - W \sin \theta = 0 \Rightarrow F_R = W \sin \theta = 600 \text{ lbf} * \sin(20^\circ) = 205 \text{ lbf}$$

$$T_R = F_R r = 205 \text{ lbf} * (14 \text{ in} * \frac{1 \text{ ft}}{12 \text{ in}}) = \mathbf{239 \text{ ft} - \text{lbf}}$$

$$R_{\text{low}} = \frac{T_R}{T_{\text{in}}} = \frac{T_R}{F_{\text{in}} L} = \frac{239 \text{ ft} - \text{lbf}}{(430 \text{ lbf})(\frac{7 \text{ in}}{12 \text{ in}} \text{ ft})} = \mathbf{0.95}$$

Variables:

W : Weight of the rover plus the two riders

F_R : Minimum required force to overcome gravity in the -y direction

T_R : Minimum torque required of the rover to climb the incline

R_{low} : low gear ratio of the rover's drivetrain

F_{in} : Combined input rider forces based on testing

L : crankarm length

g : gravitational acceleration

Θ : angle of incline

N : Rover's normal force

r : rover wheel radius

Result: The required output torque of the rover to traverse the High Butte is **239 ft-lbf**. The low gear ratio corresponding to this required torque is **0.95**. Now, the drivetrain gear sizes can be selected accordingly, and must satisfy at least a low gear ratio of 0.95 (can be lower).

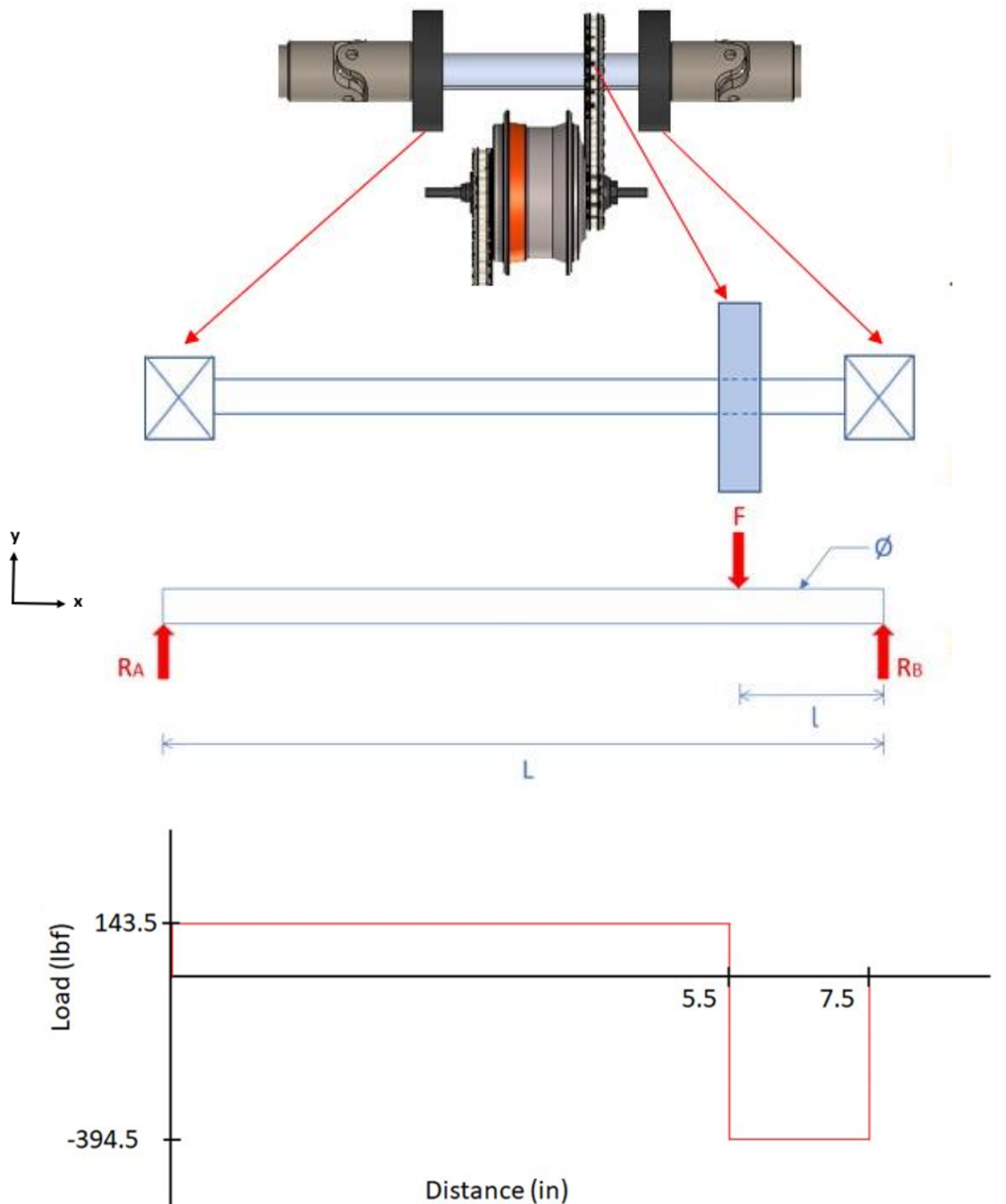
XI.D.6.

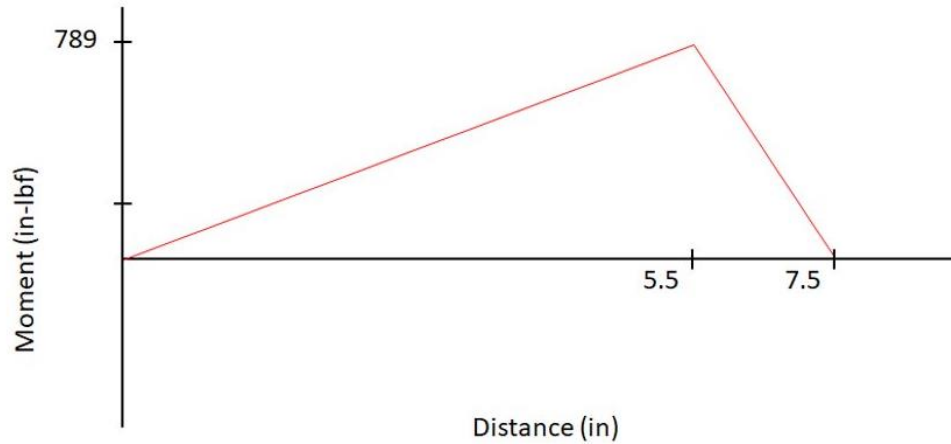
XI.D.6.a. Eng. Analysis and Materials Selection Details for SS#3-P4, P11 – Aluminum Driveshaft and Telescopic U-Joints

Gremillion

Objective: Determine the stress induced on the driveshaft

Free Body Diagram





Assumptions: Friction and power loss across the system was negligible to simulate the worst-case loading scenario experienced by the driveshaft. The tension imposed onto the 18T cog is 538 lbf during the worst-case loading conditions. The reaction forces (R_A and R_B) are located at the center of each bearing.

$$\sigma_b = -\frac{32M_{MAX}}{\pi d^3} = -\frac{(32)(789 \text{ in} - \text{lbf})}{(\pi)(1.18 \text{ in})^3} = -4.90 \text{ ksi}$$

$$\tau_T = \frac{16T}{\pi d^3} = \frac{(16)(538 \text{ lbf})(1.5 \text{ in} + .28 \text{ in})}{(\pi)(1.18 \text{ in})^3} = 2.97 \text{ ksi}$$

$$\sigma_{MAX} = \frac{\sigma_b}{2} + \sqrt{\frac{\sigma_b^2}{4} + \tau_T^2} = -6.30 \text{ ksi}$$

$$SF = \frac{\sigma_Y}{\sigma_{MAX}} = \frac{40 \text{ ksi}}{6.3 \text{ ksi}} = 6.3$$

Variables:

M_{MAX} : Maximum Bending Moment of the Driveshaft

F: Force Applied to Cog on the Driveshaft from the Chain Tension

T: Torque on the Driveshaft

σ_b : Bending Stress on the Driveshaft

d: Driveshaft Outer Diameter

τ_T : Torsional Shear Stress on the beam

SF: Safety Factor

σ_{MAX} : maximum stress on the beam

Result: The maximum stress induced on the driveshaft from the tension of the chain during the worst loading condition is **6.30 ksi**. The relationship between the yield stress of the weakest drive shaft material, 6061 T6 aluminum, and the maximum stress results in a safety factor of **6.3**. Thus, the geometry selected for the driveshaft and U-Joints are more than adequate.

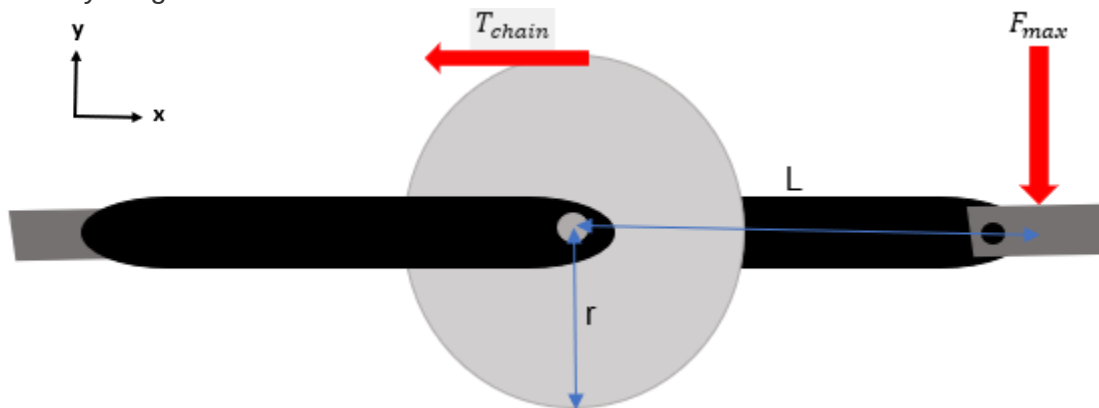
XI.D.7.

XI.D.7.a. Eng. Analysis and Materials Selection Details for SS#3-P5, P7, P10 – ASTM #40 Standard Roller Chain, Crankset, and Surly Singleator Chain Tensioner

Prejean

Objective: Determine the maximum tension on the ASTM #40 roller chain and its resultant safety factor.

Free Body Diagram



Assumptions: Chain tension vector acts perfectly orthogonal to the cog.

$$F_{max}L = T_{chain}r \Rightarrow T_{chain} = \frac{F_{max}L}{r} = \sim 1,050 \text{ lbf}$$

$$\sigma_{ult} = 3125 \text{ lbf (ANSI Standard \#40 Roller Chain)}$$

$$FS = \frac{\sigma_{ult}}{T_{chain}} = 2.98$$

Variables:

F_{max} : Maximum Force Output by Both Riders

L : Length of Crankarm

T_{chain} : Chain Tension

r : Radius of Cog

σ_{ult} : Ultimate Stress of an ANSI Standard #40 Roller Chain

FS : Safety Factor

Result: Because the safety factor yields a satisfactory value of **2.98**, the ANSI #40 Roller Chain is adequate for the design.

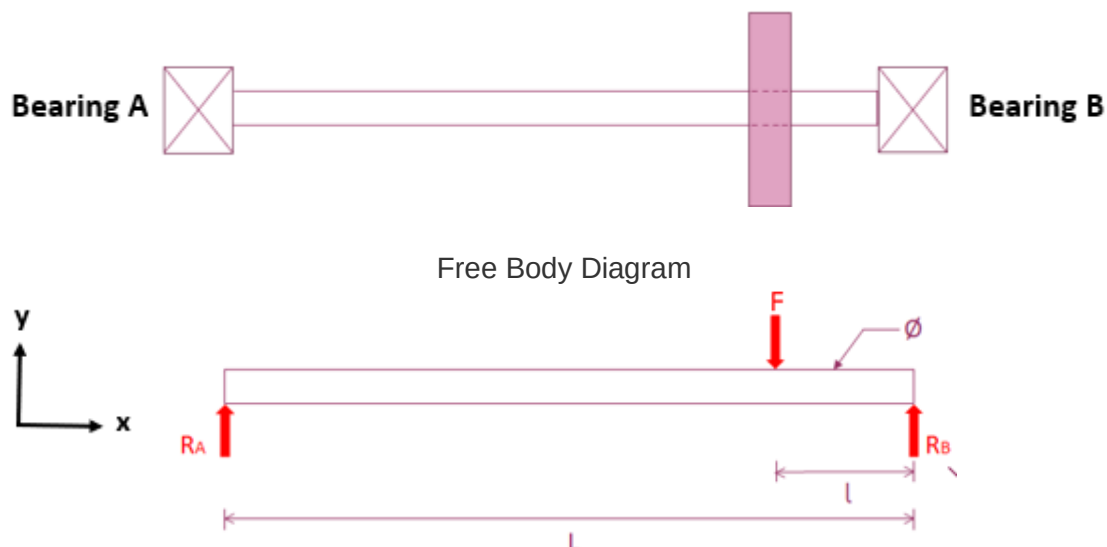
XI.D.8.

XI.D.8.a. Eng. Analysis and Materials Selection Details for SS#3-P12 – Pillow Block Bearing

Gremillion

Objective: Ensure that the bearings used on last year's rover model can support the loads of this year's design.

System



Assumptions: The system is designed for a bearing Life of 3 Years with 3% probability of failure. The target driveshaft speed is 100 rpm, which is a conservative estimate based on data collected during the preliminary driver selection testing. There is a sprocket gearing application factor (K_a) of 1.1. Reaction forces (R_A and R_B) are located at the center of the flanged pillow block bearings.

$$F_e = R_B = 394.5 \text{ lbf (Determined in Shaft Stress Analysis)}$$

$$L = 26280 \text{ hrs} \left(\frac{60 \text{ min}}{1 \text{ hr}} \right) \left(\frac{100 \text{ rev}}{\text{min}} \right) = 157.86 \times 10^6 \text{ rev}$$

$$L_R = 9 \times 10^6 \text{ rev}$$

$$K_R = 0.45$$

$$C_{req} = F_e K_a \left(\frac{L}{K_R L_R} \right)^{1/3} = 682.9 \text{ lbf}$$

Variables:

C_{Req} : Bearing Rated Capacity

F_e : Radial Load

K_a : Application Factor

L : Expected Lifetime Required

K_r : Reliability Factor
 L_R : Revolution Life with 90% Reliability (9×10^7 Rev)

Result: L06 bearings were determined to be adequate for the most severe loading conditions, **683 lbf**. Since the driveshaft is being reused from the previous year's rover, the bearing bore size is constrained to 1.18 inches (30 mm). Therefore, Team 24 will reuse the 206 flanged pillow block bearings from the 2019 Rover. The 206 series flanged pillow block bearings have a rated capacity of **1,214 lbf**; therefore, the bearings are more than adequate for the expected loading conditions.

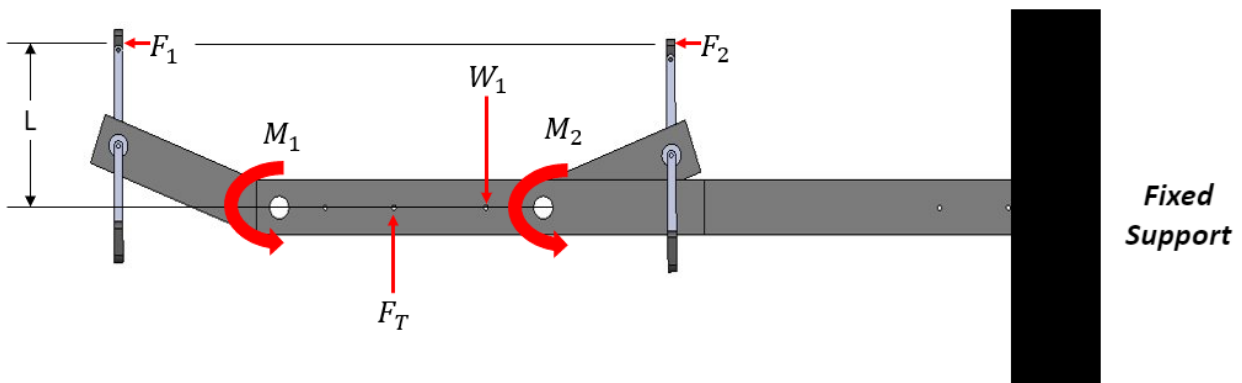
XI.D.9.

XI.D.9.a. Eng. Analysis and Materials Selection Details for SS#3-P7, P8 – Square-Tapered Bottom Bracket and Bottom Bracket Shells

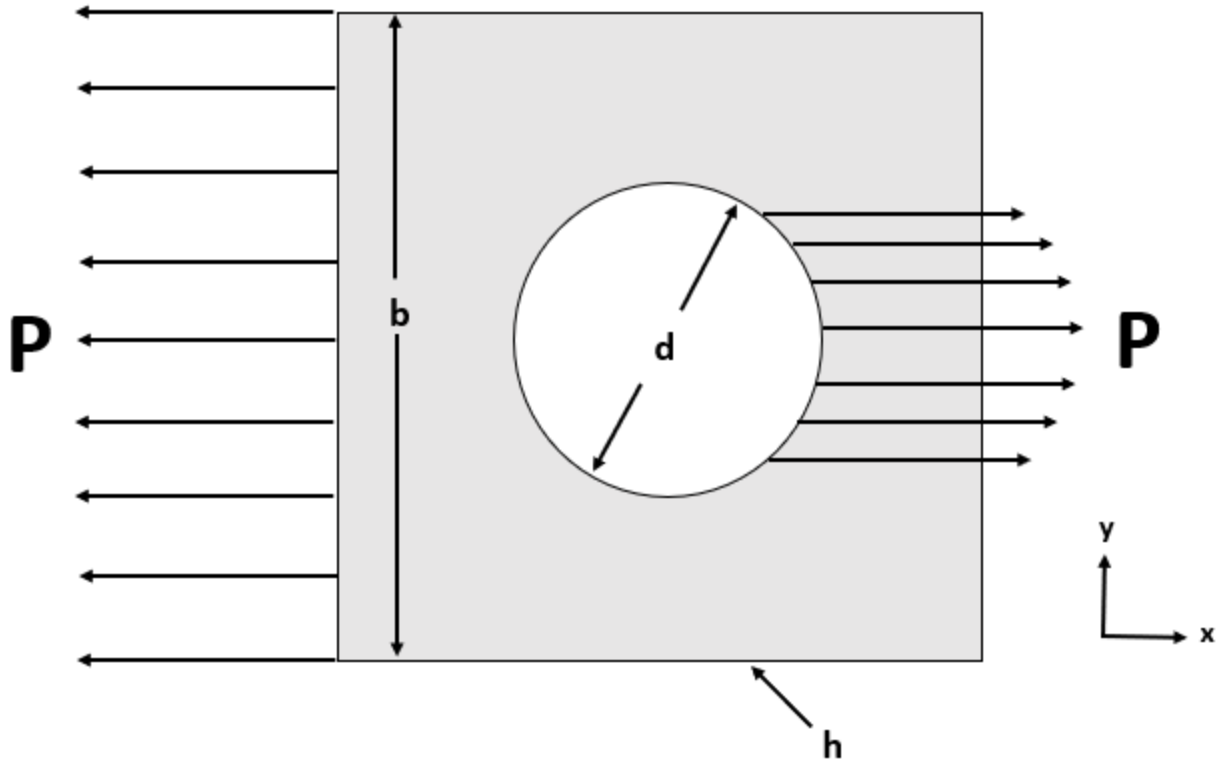
Gremillion

Objective: Determine if the hole required to fit the crank set's bottom bracket shell into the frame will induce a stress on the frame greater than the yield strength due to a stress concentration factor.

Schematic



Free Body Diagram



Assumptions: The force experienced on the plate, P , is equal to the maximum force imparted on the crankset by the driver. This maximum force was collected during driver testing and was found to be 330 lbf. Frame analysis will be treated as a plate with a pin loaded hole. The material of the frame is aluminum 6061-T6, which has a yield strength of 40 ksi. The frame geometry used is a rectangular frame with the dimensions of 4inx2inx0.125in. The hole in the frame is 1.67 inches in diameter, which is the outer diameter of the bottom bracket shell for a UN 55 Shimano bottom bracket.

$$\sigma_{nom} = \frac{P}{A} = \frac{P}{(b-d)h} = \frac{330 \text{ lbf}}{(4 \text{ in} - 1.67 \text{ in})(0.125 \text{ in})} = \mathbf{1.13 \text{ ksi}}$$

$$\sigma_{MAX} = K_t \frac{P}{A} = K_t \frac{P}{(b-d)h} = (2.7) \frac{330 \text{ lbf}}{(4 \text{ in} - 1.67 \text{ in})(0.125 \text{ in})} = \mathbf{3.05 \text{ ksi}}$$

Variables:

P : Applied Load from Driver to Crankset

b : Height of the Plate

d : Diameter of Hole

h : Thickness of Plate

K_t : Stress Concentration Factor (2.7, Found Graphically)

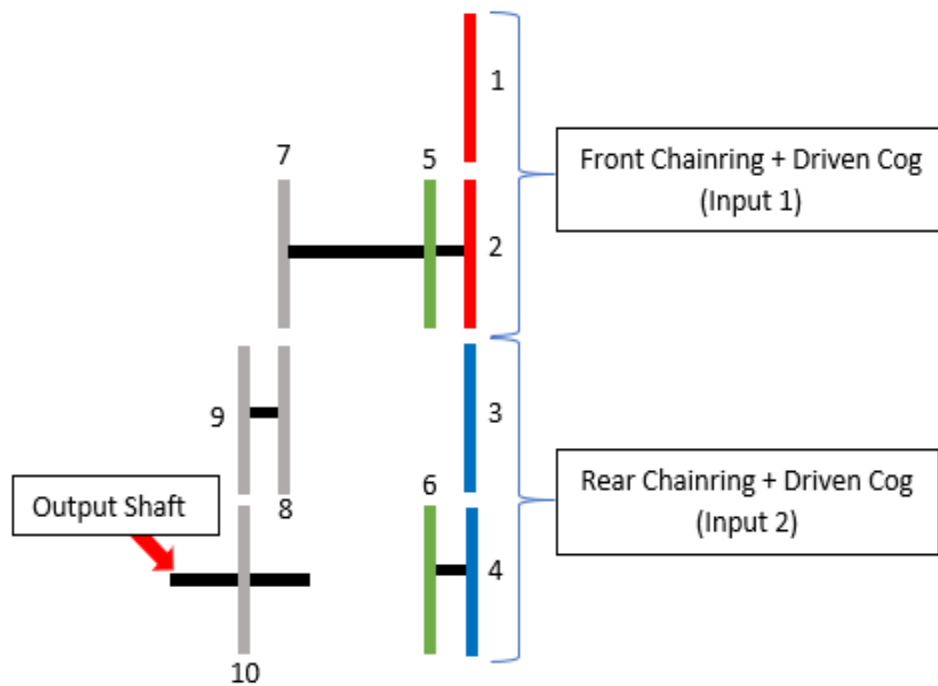
Result: Using the geometry and material properties of the single beam frame, a stress concentration factor was found to be **2.7**. The maximum stress experienced in the frame with a hole is **3.05 ksi** which is well below the yield strength of aluminum 6061-T6.

XI.D.10.

XI.D.10.a. Eng. Analysis and Materials Selection Details for SS#3-P1, P2, P9 – Surly 36T/18T Cog & NuVinci N380 Internally-Geared Hub

Prejean

Objective: Create a drivetrain transmission diagram to communicate cog size relationships (gear ratios) and prove feasibility of drivetrain concepts.



Assumptions: n/a

$$\begin{aligned}
 N1 - N7 &= 36T \\
 N8 - N10 &= 18T \\
 R_{1,2} &= R_{3,4} = R_{5,6} = 1 \\
 R_{7,8} &= 0.5 \\
 R_{8,9} &= 0.5 - 1.9^* \\
 R_{9,10} &= 1
 \end{aligned}$$

Variables:

$N\#$: Number of Teeth on Cog #

$R_{x,y}$: Gear Ratio between cogs x, y

*: Gear Ratios Available by Internally-Geared Hub (Underdrive ,Overdrive)

Result: Effective communication was conducted because of the creation of the drivetrain transmission diagram. Gear sizing was made easier because of the direct visualization of the cog relationships. Transmission diagram represents gear ratios of **0.5-1.0** and additional multipliers of **0.5-1.9** based on the internally-geared hub.

XI.D.11.

XI.D.11.a. Eng. Analysis and Materials Selection Details for SS#3-P6 – Hub Anchor

Prejean

Objective: Describe how the hub anchor will be an effective off-the-shelf part used in our design.

Reasoning of its Choice: One of the main issues noticed when testing last year's rover model was that the internally-geared hub did not stay in a static position relative to its support brackets. This was because the force acting on the hub had a direction component that fell within the bracket's slot which allowed the hub to travel in the slot. The 'hub anchor' will prevent any motion of the internally-geared hub on the 2020 Rover.

How it Satisfies the Requirements of its Purpose: The support bracket for the internally-geared hub was designed to incorporate the hub anchor. As the internally-geared hub tries to translate along the line of direction of the bracket's slot, its axle will contact the ring on the hub anchor. This will put an external force onto the hub anchor, but the anchor/nut portion of the hub anchor will contact the bottom of the support bracket, which will prevent translation of the hub.



[18]

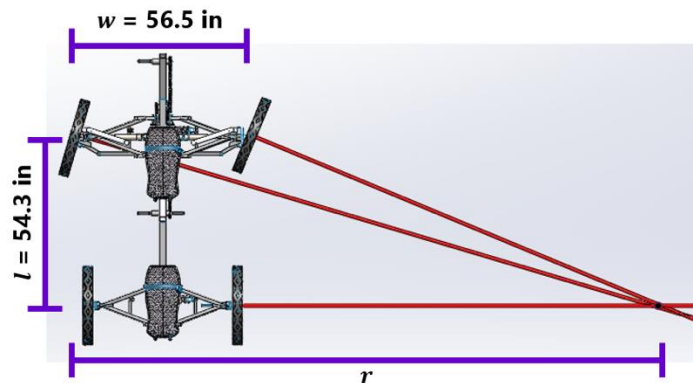
Result: The hub anchor will be integrated onto the 2020 Rover during the manufacturing phase in order to prevent translation of the internally-gearred hub in its support bracket.

XI.D.12. Eng. Analysis Details for SS#4- Push-Pull Steering

XI.D.12.a. Eng. Analysis and Materials Selection Details for SS#4P1-P11- Steering Assembly

Donner

Objective: Calculate the required wheel angles to satisfy turning radius requirement.



Assumptions: Using the given rover dimensions and a team imposed 13 foot turning radius constraint, the following equations, based off of simple geometry, yield the following:

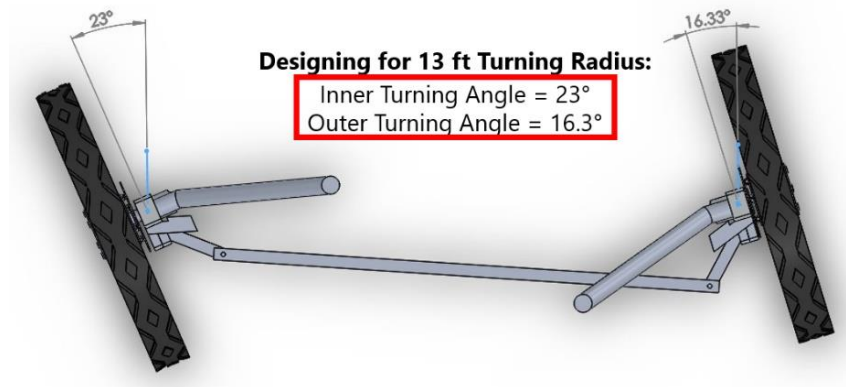
$$\tan \varphi_i = \frac{l}{r - \frac{w}{2}} = 23^\circ$$

$$\tan \varphi_o = \frac{l}{r + \frac{w}{2}} = 16.3^\circ$$

Variables:

- φ_i : Inner Turning Angle
- φ_o : Outer Turning Angle
- r : Turning Radius
- l : Rover Wheelbase
- w : Rover Track Length

Results: These angles are guaranteed by the tie rod linkage, as shown below:

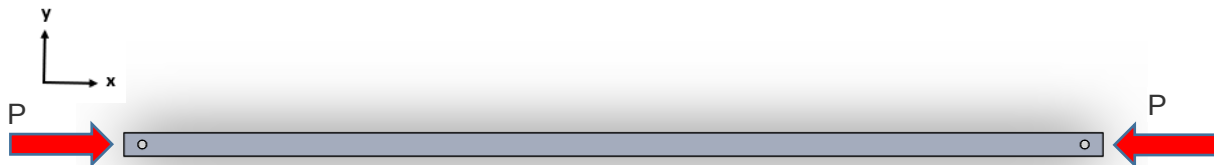


XI.D.12.b. Eng. Analysis and Materials Selection Details for SS#4P2 – Tie Rod

Donner

Objective: Calculate Tie Rod Buckling Load

Assumptions: Compressive Force experienced by Tie Rod will not exceed 1000 lbs.



$$P = 1000 \text{ lbf}$$

$$l = 40 \text{ in}$$

$$b = .75 \text{ in}$$

$$h = 1 \text{ in}$$

$$A = bh = .75 \text{ in}^2$$

$$I_{min} = \frac{bh^3}{12} = 0.03516 \text{ in}^4$$

$$k_{min} = \sqrt{\frac{I_{min}}{A}} = 0.2165 \text{ in}$$

$$\left(\frac{l}{k_{min}}\right) = 184.74$$

$$\left(\frac{l}{k_{min}}\right)_{cr} = \sqrt{\frac{2\pi^2 E}{\sigma_y}} = 70.25$$

where

$$E = 10,000 \text{ ksi}$$

$$\sigma_y = 40000 \text{ psi}$$

since $\left(\frac{l}{k_{min}}\right) > \left(\frac{l}{k_{min}}\right)_{cr}$, use Euler

$$P_{cr} = \frac{\pi^2 EI_{min}}{(KL)^2} = \mathbf{2168.8 \text{ lbf}}$$

where

$$K = 1$$

$$\text{Safety Factor, } SF = \frac{P_{cr}}{P} = \mathbf{2.17}$$

Variables:

P : Compressive Force on Tie Rod

l : Length of Tie Rod (from pin to pin)

b : Base of Cross Section

h : Height of Cross Section

A : Cross-Sectional Area

I_{min} : Minimum Cross-Sectional Moment of Inertia

k_{min} : Minimum Cross-Sectional Radius of Gyration

I_{min} : Minimum Cross-Sectional Moment of Inertia

$\left(\frac{l}{k_{min}}\right)$: Slenderness Ratio

$\left(\frac{l}{k_{min}}\right)_{cr}$: Critical Slenderness Ratio

E : Modulus of Elasticity

σ_y : Yield Strength

P_{cr} : Critical Buckling Load

K : Effective Length Factor

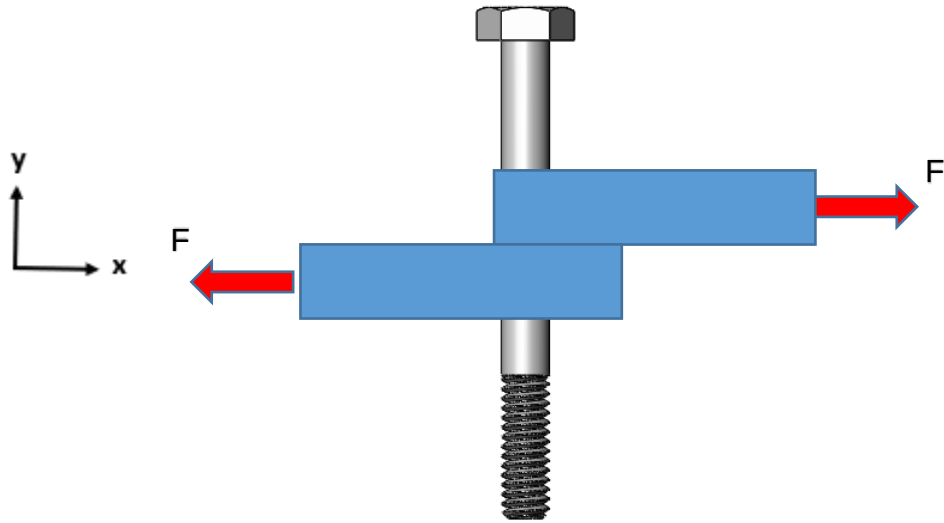
Results: The given Tie Rod dimensions and assumptions yield a safety factor of **2.17**.

Xi.D.12.c.

Eng. Analysis and Materials Selection Details for SS#4P9- Tie Rod Bolt

Donner

Objective: Calculate Shear Stress on Tie Rod Bolts



Assumptions: Shear force, F won't exceed 1,000 pounds. Also, the shear strength of a bolt is roughly 60% of its tensile strength [19].

$$\begin{aligned}
 d &= .375 \text{ in.} \\
 A &= \frac{\pi d^2}{4} = 0.1104 \text{ in.}^2 \\
 \tau &= \frac{F}{A} = \mathbf{9,057.97 \text{ psi}} \\
 TS &= 150,000 \text{ psi} \\
 \tau_a &= .6TS = \mathbf{90,000 \text{ psi}} \\
 \text{Safety Factor, } SF &= \frac{\tau_a}{\tau} = \mathbf{9.94}
 \end{aligned}$$

Variables:

F : Shear Force on Bolt

d : Diameter of Bolt

A : Cross-Sectional Area of Bolt

τ : Shear Stress

TS : Tensile Strength of Bolt

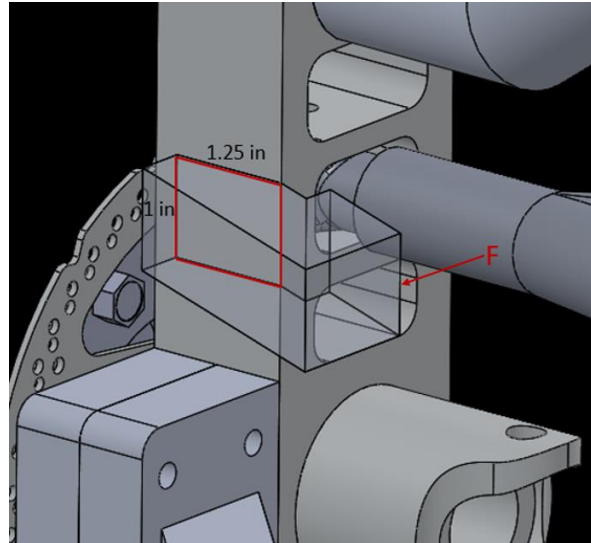
τ_a : Maximum Allowable Shear Stress

Results: The bolt's dimensions under the given loading circumstance yields a safety factor of **9.94**.

XI.D.12.d. Eng. Analysis and Materials Selection Details for SS#4P1,P5 – Steering Knuckle, ROM Limiter

Donner

Objective: Calculate the stresses found in the fillet welds connecting the Range of Motion Limiter and the Steering Knuckle



Assumptions: Force, F , is an impact force that acts at the end of the range of motion limiter. Designing with this assumptions is a worst-case scenario. Assume Impact Force will not exceed 2000 lbs.

$$F = 2000 \text{ lbf}$$

$$H = 1 \text{ in}$$

$$B = 1.25 \text{ in}$$

$$t_{min} = 1 \text{ in}$$

$$a = \sqrt{t_{min}} = 1$$

$$r_y = .75 \text{ in}$$

$$A_w = 2a(H + B) = 4.5 \text{ in}^2$$

$$I_w = \frac{aH^3}{6} + \frac{B((B + 2a)^3 - H^3)}{12} = 2.875 \text{ in}^2$$

$$M = Fd = 4500 \text{ lb} \cdot \text{in}$$

$$\sigma_M = \frac{M \cdot r_y}{I_w} = \mathbf{1173.9 \text{ psi}}$$

$$\tau = \frac{F}{A_w} = \mathbf{628.93 \text{ psi}}$$

$$S_e = \sqrt{\sigma_M^2 + \tau^2} = \mathbf{1331.76 \text{ psi}}$$

Variables:

F : Impact Force

H : Height of Weld Group

B : Base of Weld Group

t_{min} : Thickness of Thinnest Connected Material

a : Fillet Weld Throat Thickness

r_y : Distance from Outer Weld to Centroid

A_w : Weld Throat Area

I_w : Moment of Inertia of the Weld

σ_M : Stress due to Moment
 M : Moment due to Impact Force
 d : Moment Arm
 τ : Shear Stress due to Impact Force
 S_e : Equivalent Stress due to Combined Load

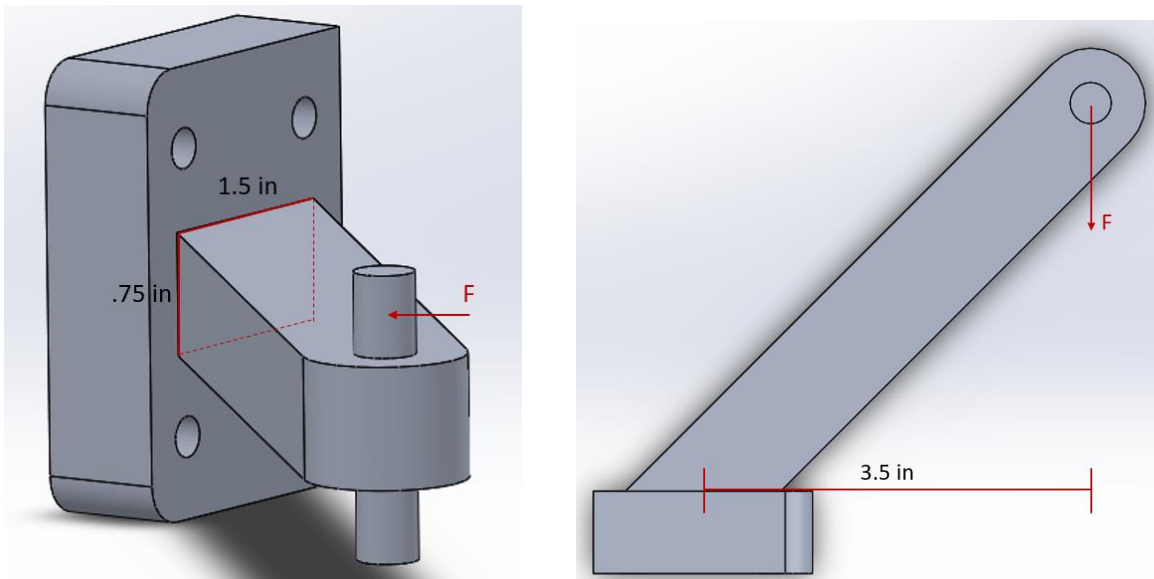
Results: Due to the assumed weld geometry and loading conditions, the fillet welds will be subjected to a maximum stress of **1,331.76 psi**.

Eng. Analysis and Materials Selection Details for SS#4P3,P4 – Steering Arm, Bolt Plate

XI.D.12.e. Eng. Analysis and Materials Selection Details for SS#4P3,P4 – Steering Arm, Bolt Plate

Donner

Objective: Calculate the stresses found in the welds connecting the Steering Arm and Bolt Plate



Assumptions: Force, F , is an impact force that acts at the end of the range of motion limiter. Designing with this assumptions is a worst-case scenario. Assume Impact Force will not exceed 1000 lbs.

$$\begin{aligned}
 F &= 1000 \text{ lbf} \\
 H &= 0.75 \text{ in} \\
 B &= 1.5 \text{ in} \\
 t_{min} &= 0.75 \text{ in} \\
 a &= \sqrt{t_{min}} = 0.866 \\
 r_y &= 0.75 \text{ in} \\
 A_w &= 2a(H + B) = 3.897 \text{ in}^2
 \end{aligned}$$

$$I_w = \frac{aH^3}{6} + \frac{B((B + 2a)^3 - H^3)}{12} = 1.919 \text{ in}^2$$

$$M = Fd = 3500 \text{ lb} \cdot \text{in}$$

$$\sigma_M = \frac{M \cdot r_y}{I_w} = 1,367.6 \text{ psi}$$

$$\tau = \frac{F}{A_w} = 256.6 \text{ psi}$$

$$S_e = \sqrt{\sigma_M^2 + \tau^2} = 1,391.47 \text{ psi}$$

Variables:

- F : Impact Force
- H : Height of Weld Group
- B : Base of Weld Group
- t_{min} : Thickness of Thinnest Connected Material
- a : Fillet Weld Throat Thickness
- r_y : Distance from Outer Weld to Centroid
- A_w : Weld Throat Area
- I_w : Moment of Inertia of the Weld
- σ_M : Stress due to Moment
- M : Moment due to Impact Force
- d : Moment Arm
- τ : Shear Stress to due Impact Force
- S_e : Equivalent Stress due to Combined Load

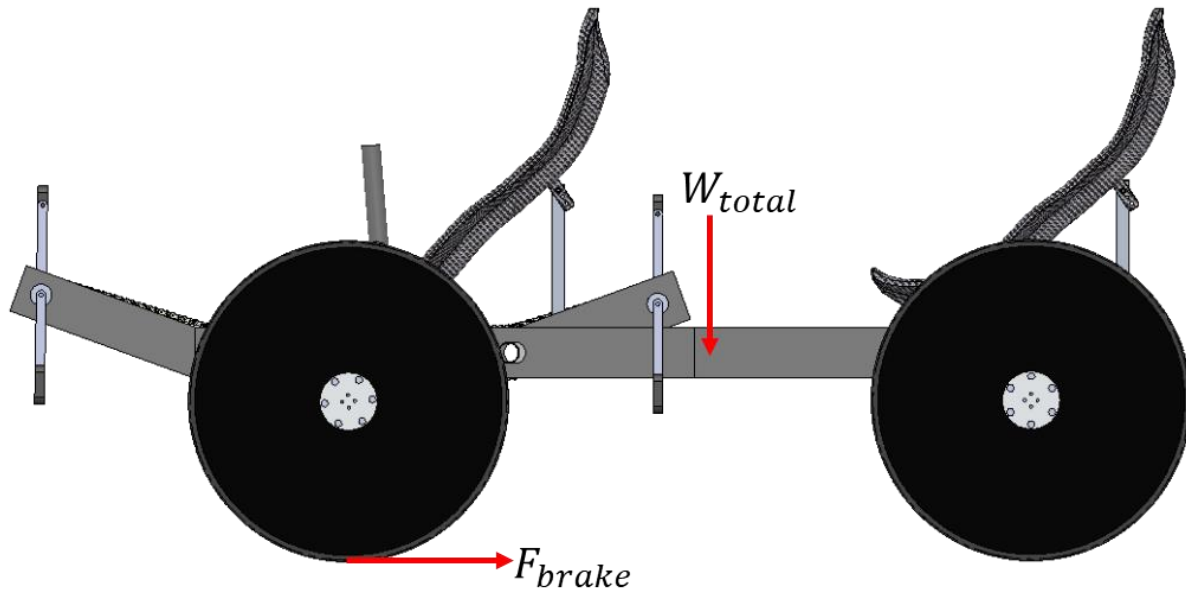
Results: Due to the assumed weld geometry and loading conditions, the fillet welds will be subjected to a maximum stress of **1,391.47 psi**.

XI.D.13. Eng. Analysis Details for SS#5- Hydraulic Brakes

XI.D.13.a. Eng. Analysis and Materials Selection Details for SS#5P1- Hydraulic Brakes

Guo

Objective: Calculate the expected braking distance and time for an average rover speed.



Assumptions: Average rover speed was set at 10 mph from testing conducted on the old rover as well as drivetrain gear ratio analysis. The combined weight of the rover and drivers was 431.1 lbs. the coefficient of friction was assumed to be concrete at 0.6 for the worse case scenario. No slipping was assumed.

$$\begin{aligned}
 v_{rover} &= 10 \text{ mph} \\
 W_{total} &= 431.1 \text{ lbs.} \\
 \mu_{concrete} &= 0.6 \\
 KE &= \frac{1}{2}mv_{rover}^2 = 1,440.9 \text{ ft-lbf} \\
 F_{wheel} &= \mu mg = 258.7 \text{ lbf} \\
 d_{stopping} &= \frac{KE}{F_{wheel}} = 5.6 \text{ ft} \\
 a_{rover} &= \frac{v_{rover}^2}{2d_{stopping}} = 19.3 \text{ ft/s}^2 \\
 t_{stopping} &= \frac{v_{rover}}{a_{rover}} = 0.8 \text{ s}
 \end{aligned}$$

Variables:

v_{rover} : Average speed of rover

W_{total} : weight of rover and drivers

$\mu_{concrete}$: Coefficient of friction of concrete

KE : Kinetic Energy of rover

m : mass of rover and drivers

F_{wheel} : Force of wheel from gravity

g : gravitational acceleration

$d_{stopping}$: stopping distance of rover

a_{rover} : deacceleration of the rover

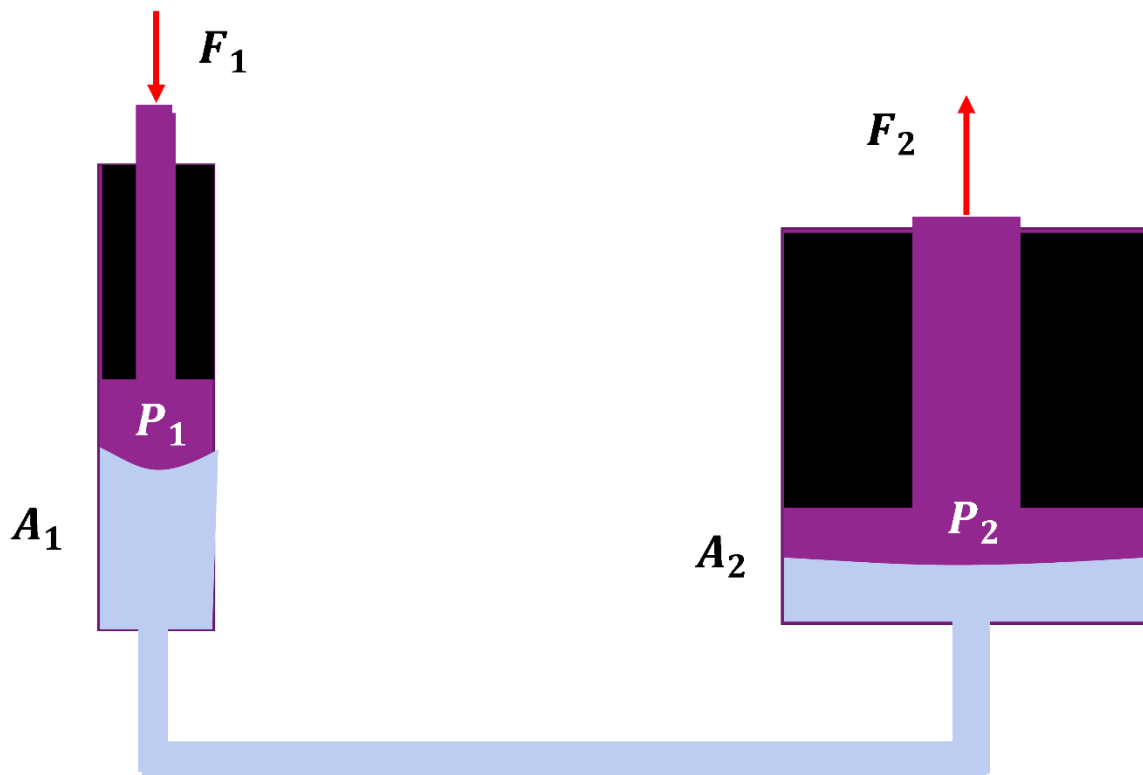
$t_{stopping}$: stopping time of rover

Results: The stopping distance of a rover going at an average speed of 10 mph is **5.6 ft** at a stopping time of **0.8 s**.

XI.D.13.a. Eng. Analysis and Materials Selection Details for SS#5P1- Hydraulic Brakes

Guo

Objective: Calculate the maximum forces applied to the caliper piston.



Assumptions: Gripping force for the right and left hand for a male of age 20-24 based on grip strength study. The combined weight of the rover and drivers was 431.1 lbs. Minimum and Maximum coefficients of friction for hydraulic disc brakes were assumed to be 0.35 and 0.42 respectively.

$$\begin{aligned} F_{right} &= 121.0 \text{ lbf} \\ F_{left} &= 104.5 \text{ lbf} \\ F_{hand} &= \frac{F_{right} + F_{left}}{2} = 112.75 \text{ lbf} \\ A_1 &= \frac{\pi}{4} D_1^2, \quad A_2 = \frac{\pi}{4} D_2^2, \end{aligned}$$

$$P_1 = \frac{F_{hand}}{A_1} = \frac{F_2}{A_2} \rightarrow F_2 = \frac{A_2}{A_1} F_1 = 1297.5 \text{ lbf}$$

$$2F_2 = F_{Brake\ Pedal} = 2595.0 \text{ lbf}$$

$$F_{Brake\ Pedal} \times \mu_{min} = F_{R,min} = 908.2 \text{ lbf}$$

$$F_{Brake\ Pedal} \times \mu_{max} = F_{R,max} = 1,089.9 \text{ lbf}$$

$$F_{R,min} \times r_R = T_{B,min} = 2,860.6 \text{ lb-in}$$

$$F_{R,max} \times r_R = T_{B,max} = 3,432.7 \text{ lb-in}$$

$$\frac{T_{B,min}}{R_w} = F_{T,min} = 204.3 \text{ lbf}$$

$$\frac{T_{B,max}}{R_w} = F_{T,max} = 245.2 \text{ lbf}$$

$$2F_{T,min} = F_{total,min} = \mathbf{408.7 \text{ lbf}}$$

$$2F_{T,max} = F_{total,max} = \mathbf{490.4 \text{ lbf}}$$

Variables:

F_{right} : Force of right hand

F_{left} : Force of left hand

F_{hand} : Force from hands

A_1 : Area of 1

A_2 : Area of 2

D_1 : Diameter of 1

D_2 : Diameter of 2

P_1 : Pressure in hydraulic brake

a_{rover} : deacceleration of the rover

$F_{Brake\ Pedal}$: Force on Brake Pedal

F_R : Reaction Force on Brakes

μ : Coefficient of friction for brakes

r_R : radius of rotor

$T_{B,min}$: Torque on brake

R_w : Radius of wheel

Results: The maximum force applied to the caliper piston is **490.4 lbf**.

Table 2: Performance of All Subjects an Grip Strength (pounds)

	Hand	Men					Women				
		Mean	SID	SE	Low	mm	Mean	SID	SE	Low	High
20-24	R	121.0	20.6	3.8	91	167	70.4	14.5	2.8	46	95
	L	104.5	21.8	4.0	71	150	61.0	13.1	2.6	33	88
25-29	R	120.8	23.0	4.4	78	158	74.5	13.9	2.7	48	97
	L	110.5	16.2	3.1	77	139	63.5	12.2	2.4	48	97
30-34	R	121.8	22.4	4.3	70	170	78.7	19.2	3.8	46	137
	L	110.4	21.7	4.2	64	145	68.0	17.7	3.5	36	115
35-39	R	119.7	24.0	4.8	76	176	74.1	10.8	2.2	50	99
	L	112.9	21.7	4.4	73	157	66.3	11.7	2.3	49	91
40-44	R	116.8	20.7	4.1	84	165	70.4	13.5	2.4	38	103
	L	112.8	18.7	3.7	73	157	62.3	13.8	2.5	35	94
45-49	R	109.9	23.0	4.3	65	155	62.2	15.1	3.0	39	100
	L	100.8	22.8	4.3	58	160	56.0	12.7	2.5	37	83
50-54	R	113.6	18.1	3.6	79	151	65.8	11.6	2.3	38	87
	L	101.9	17.0	3.4	70	143	57.3	10.7	2.1	35	76
55-59	R	101.1	26.7	5.8	59	154	57.3	12.5	2.5	33	86
	L	83.2	23.4	5.1	43	128	47.3	11.9	2.4	31	76
60-64	R	89.7	20.4	4.2	51	137	55.1	10.1	2.0	37	77
	L	76.8	20.3	4.1	27	116	45.7	10.1	2.0	29	66
65-69	R	91.1	20.6	4.0	56	131	49.6	9.7	1.8	35	74
	L	76.8	19.8	3.8	43	117	41.0	8.2	1.5	29	63
70-74	R	75.3	21.5	4.2	32	108	49.6	11.7	2.2	33	78
	L	64.8	18.1	3.7	32	93	41.5	10.2	1.9	23	67
75+	R	65.7	21.0	4.2	40	135	42.6	11.0	2.2	25	65
	L	55.0	17.0	3.4	31	119	37.6	8.9	1.7	24	61
All	R	104.3	28.3	1.6	32	176	62.8	17.0	0.96	25	137
Subjects	L	93.1	27.6	1.6	27	160	53.9	15.7	0.88	23	115

XI.D.14. Eng. Analysis Details for SS#6- Double Wishbone Suspension

XI.D.14.a. Eng. Analysis and Materials Selection Details for SS#6

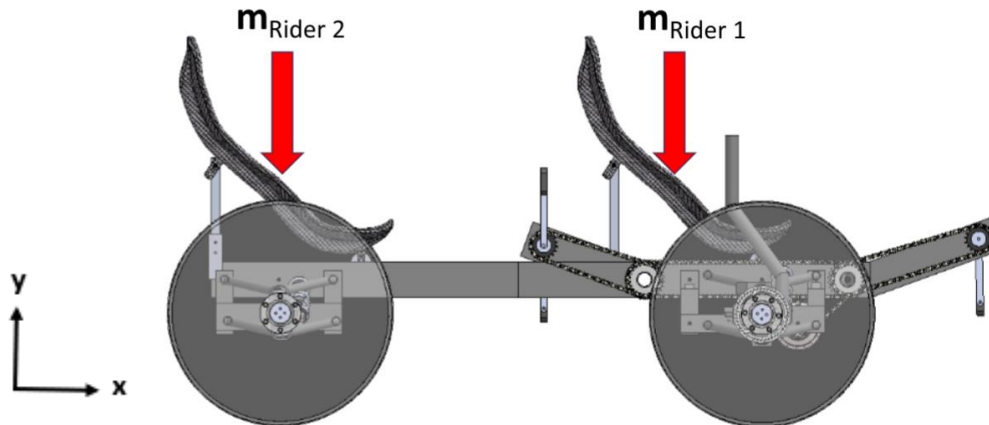
Hebert

Objective: Determine the Aluminum alloy best suited for the suspension.

Metric	Weight	2024	6061-T6	7075
Yield Strength	0.3	1	2	3
Weight	0.25	2	3	1
Manufacturability	0.2	1	3	2
Tensile Strength	0.15	1	2	3
Cost	0.1	2	3	1
Total	1	1.35	2.55	2.1

Outcome & Significance: Of the three alloys considered **Aluminum 6061-T6** was seen at the best fit because of its weight, ease of manufacturability, and it was the most cost-efficient option.

Objective: Determine the maximum impact force applied to the rover. The impact force will likely be the highest force the rover encounters so this force will determine the thickness of various members and other quantities throughout the design.



Assumptions: Fully elastic reaction, which is not actually the case, but this assumption also works as a safety factor. Reusing LSU's 2019 Rover Team springs, a spring constant of **685.13 slug/s²**, the mass from the reaction forces of the front rider (Rider 1), **431.1 lbs.**, and a max top velocity of **10 mph** during impact will be used to calculate the force.

$$\frac{1}{2}mv^2 = \frac{1}{2}k\delta \quad (1)$$

$$F_{Impact} = v_{top}\sqrt{mk} = 10\sqrt{\left(685.13\frac{\text{slugs}}{\text{s}^2}\right)(431.1\text{ lbs})} = \mathbf{1,341.38\text{ lbf}} \quad (2)$$

Variables:

m : Mass

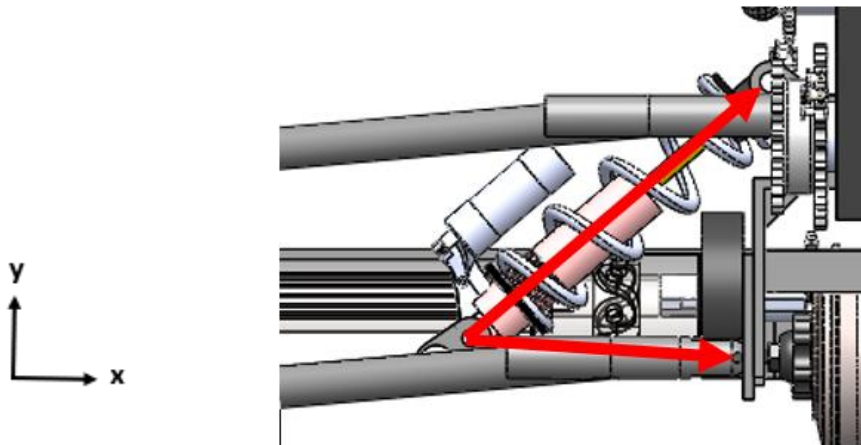
F_{Impact} : Impact force

k : Spring Constant

v_{top} : Top impact velocity

Outcome & Significance: The max impact force at the front of the rover was calculated to be **1,341.31 lbf.**

Objective: Determine the location of the shock mounting arm and the angle of the shock. The Rockshox Vivid R2C was chosen because it can be removed from LSU's 2019 Team's rover. The Rockshox Vivid Air R2C was purchased because it offered similar properties to Rockshox Vivid R2C, but without a coil, thus simplifying the manufacturing process.



Assumptions: Assuming a completely vertical position would be the optimal position for the shock. Evaluating the component with the control arm having a length of **16 in.**, the fully extended shock length being **9.5 in.**, the height between the mounting points being **5.5 in.**, and the shock mount will extend **4 in.** from the frame the shock angle and bending arm length can be determined.

- The steering knuckle mounting point is **13.85 in.** horizontally from the frame.

$$\text{shock angle} = \sin^{-1}(h_{mp}/l_{shock}) = \sin^{-1}(5.5/9.5) = \mathbf{34.6^\circ} \quad (2)$$

$$9.5^2 = 6.5^2 + x^2 \quad (2)$$

$$x = 7.8 \text{ in}$$

$$13.85" - 7.8" - 4" = \mathbf{2.95"} \quad (3)$$

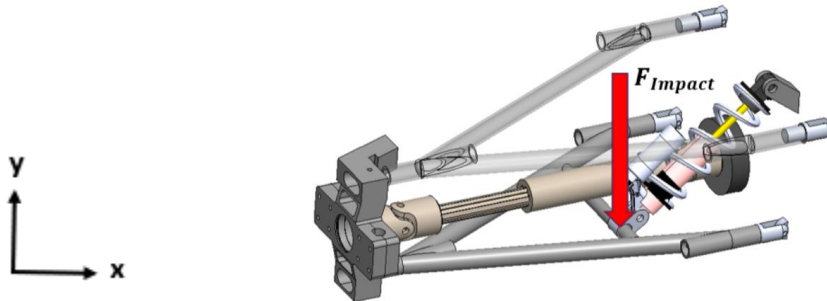
Variables:

h_{mp} : Height of the mounting point

L_{Shock} : Shock Length

Outcome & Significance: The shock will be mounted at **34.6°** and the mounting arm on the control arm will be **2.95"** horizontal from the steering knuckle towards the frame.

Objective: Determine the minimum thickness of the suspension bending arms.



Assumptions: Using **Aluminum 6061 T6** properties, an outer diameter of $\frac{3}{4}$ in., a safety factor of **2.7**, and assuming the maximum impact force calculated in XI.D.14.a is completely absorbed here.

$$P = F \sin(\theta)$$

$$M = Pab(L_{Ctrl})^{-1}$$

$$M = \mathbf{2,143 \text{ lbf in}}$$

$$\sigma_b = \frac{Mr_o}{I}$$

$$I = \frac{\pi}{4} (r_o^4 - r_i^4)$$

$$t = r_o - r_i = \mathbf{0.1 \text{ in.}}$$

Variables:

M : Moment

L_{Ctrl} : Control arm Length

σ_b : Height of the mounting point

I : Area moment of inertia

t : Thickness

r_o : Outer radius

r_i : Inner radius

F : Force

θ : Shock angle

Outcome & Significance: The thickness of the suspension arms must be **0.1 in.** when the OD is $\frac{3}{4}$ in. wide.

Objective: This Python script is used to calculate the necessary analysis from XI.D.14.a, XI.D.14.c, and XI.D.14.d. The script was verified with hand calculations before giving it significant meaning in the analysis process.

```

1. # -*- coding: utf-8 -*-
2.
3. """
4.     __author__ = Dakota Hebert
5.     __date__ = 10-31-2019
6. """
7.
8.
9. def suspension_model(front_mass, rear_mass, front_spring_constant, rear_spring_constant
    , max_velocity, mounting_point_height, shock_length, steering_knuckle_distance, shock_m
    ount_extension, a, b, outer_radius, control_arm_length, factor_of_safety):
10.     # Conversions
11.     max_velocity = max_velocity * 5280 / 3600
12.     yield_strength = 63.1 * 1000 # ksi->psi
13.     yield_strength = yield_strength / factor_of_safety
14.
15.     # Calculations
16.
17.     # Impact force
18.     force = lambda v, m, k: v * np.sqrt(m * k) / 32.1740
19.     front_impact_force = force(max_velocity, front_mass, front_spring_constant)
20.     rear_impact_force = force(max_velocity, rear_mass, rear_spring_constant)
21.
22.     # Shock Angle
23.     shock_angle = np.arcsin(mounting_point_height / shock_length) * 180 / np.pi
24.
25.     # Mounting Arm Location
26.     x = np.sqrt(shock_length ** 2 - mounting_point_height ** 2)
27.     mounting_arm_distance = steering_knuckle_distance - x - shock_mount_extension
28.
29.     # Suspension arm bending
30.
31.     # Bending force
32.     bending_force = lambda f, theta: f * np.sin(theta * np.pi / 180)
33.     front_bending_force = bending_force(front_impact_force, shock_angle)
34.     rear_bending_force = bending_force(rear_impact_force, shock_angle)
35.
36.     # Moment
37.     rear_moment = rear_bending_force * a * b * (1 / control_arm_length)
38.     front_moment = front_bending_force * a * b * (1 / control_arm_length)
39.     old_moment = 682 * a * b * (1 / control_arm_length)
40.
41.     # Inner Radius Calculations
42.     inertia = lambda M: M * outer_radius / yield_strength
43.     inner_r = lambda i: (-(i * (4 / np.pi)) + outer_radius ** 4) ** (1 / 4)
44.     front_inner_radius = inner_r(inertia(front_moment))
45.     rear_inner_radius = inner_r(inertia(rear_moment))
46.
47.     # Thickness
48.     front_thickness = outer_radius - front_inner_radius
49.     rear_thickness = outer_radius - rear_inner_radius
50.

```

```

51.     # Outputs
52.
53.     print("The front impact force is: %.2f lbf" % front_impact_force)
54.     print("The rear impact force is: %.2f lbf\n" % rear_impact_force)
55.     print("The shock angle is: %.1f\u00b0 \n" % shock_angle)
56.     print("The mounting arm on the control arm will be %.2f inches horizontally from th
    e steering knuckle towards the "
57.         "frame.\n" % mounting_arm_distance)
58.     print("The front bending force is: %.2f lbf" % front_bending_force)
59.     print("The rear bending force is: %.2f lbf\n" % rear_bending_force)
60.     print("The front moment is: %.2f lb in" % front_moment)
61.     print("The rear moment is: %.2f lb in\n" % rear_moment)
62.     print("The front inner radius is: %.4f in" % front_inner_radius)
63.     print("The rear inner radius is: %.4f in\n" % rear_inner_radius)
64.     print("The front thickness is: %.4f in" % front_thickness)
65.     print("The rear thickness is: %.4f in" % rear_thickness)

```

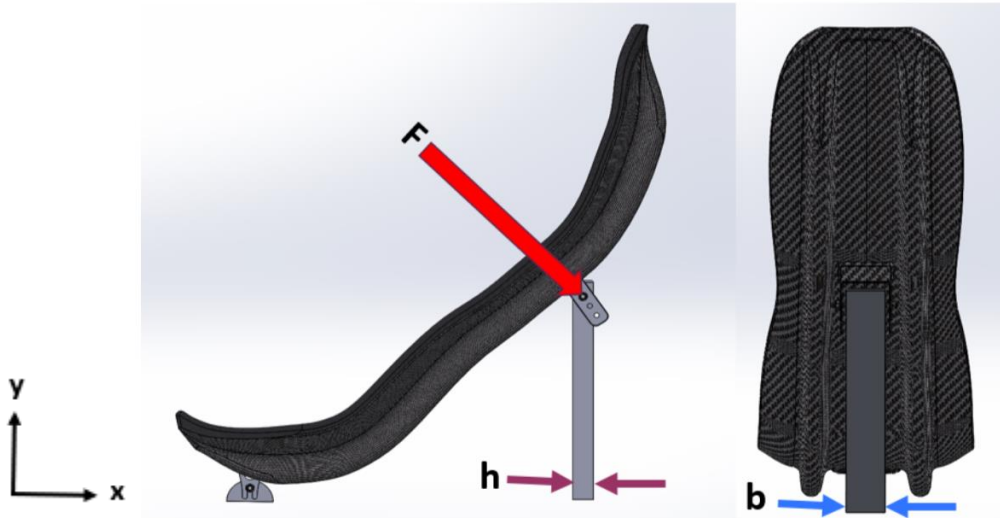
Outcome & Significance: This code was used to generate results for different variations of the mass, springs, velocity, geometry, and factors of safety throughout the design process.

XI.D.15. Eng. Analysis Details for SS#7- Seats

XI.D.15.a. Eng. Analysis and Materials Selection Details for SS#7-P3, P4 – Front seat bar, rear seat bar

Hebert

Objective: Determine the minimum width of the bar used to support the seats.



Assumptions: Assuming **Aluminum 6061 T6** properties, a thickness of **1 1/8 in.**, the length of the bar is **11.36 in.**, and that the impact force calculated in XI.D.14.a is completely absorbed here. This is a conservative assumption that will act as a safety factor. The seat was modeled as a beam with one end fixed with the other end rounded so **n=2**.

$$F = \frac{\pi^2 EI}{kL} = 1,400 \text{ lbf} \quad (3)$$

$$k = \sqrt{1/n} = 0.707 \quad (2)$$

$$I = \frac{bh^3}{12} \quad (3)$$

$$F = \frac{\pi^2 Ebh^3}{12 \left[L \left(\frac{1}{n} \right)^{\frac{1}{2}} \right]^2} \quad (4)$$

$$\Rightarrow b = \frac{12F \left[L \left(\frac{1}{n} \right)^{\frac{1}{2}} \right]^2}{\pi^2 E h^3} \quad (5)$$

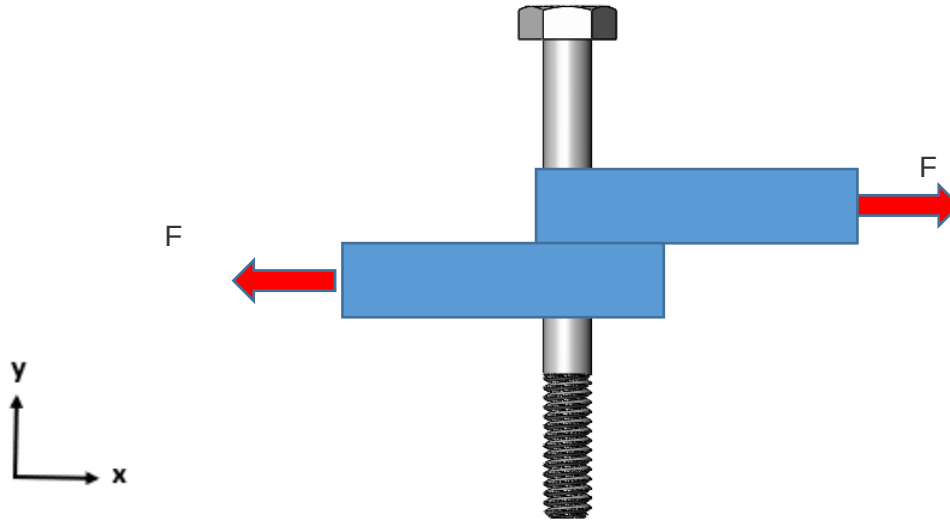
$$b = 1.93 \text{ in}$$

Variables:

F : Force
 b : Width
 I : Moment of Inertia
 h : Thickness
 L : Length of column
 n : Factor accounting for the end conditions
 E : Modulus of Elasticity

Outcome & Significance: A minimum width for the seat post has been calculated to be **1.93 in.** This means using the thickness and length specified will be enough to allow for the bar to fit on the frame without any overhang.

Objective: Determine if the bolts selected can withstand the impact.



Assumptions: 85% of the impact force calculated in XI.D.14.a is absorbed here. This is a conservative assumption that will act as an additional safety factor. The length of the bolt is **2.00 in**. The tensile strength of the bolt is **150,000 psi**. assume that the shear strength is 60% of that, which is **90,000 psi**. The bolt has a **¼ in.** diameter

$$\tau = \frac{F}{\pi r^2} = \frac{1200 \text{ lbf}}{\pi (0.125 \text{ in})^2} = \mathbf{24,490 \text{ psi}}$$
$$n = \frac{\tau_{\text{allowable}}}{\tau_{\text{actual}}} = \frac{90,000 \text{ psi}}{24,490 \text{ psi}} = \mathbf{3.7}$$

Variables:

F : Force

r : Radius of the bolt

τ : Shear stress

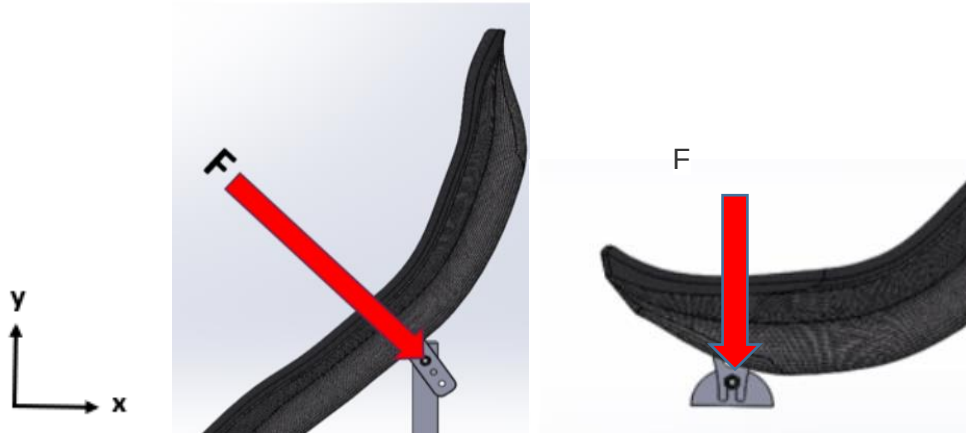
n : Factor of safety

Outcome & Significance: The bolts selected offer will have a max shear force of **24,2490 psi** which offers a factor of safety of **6.1**.

XI.D.15.c. Eng. Analysis and Materials Selection Details for SS#7-P6, P7 – Top mounting bracket, bottom mounting bracket

Hebert

Objective: Determine if the seat attachments can withstand the impact force.



Assumptions: 80% of the impact force calculated in XI.D.14.a is completely absorbed here. The attachments use properties of **Aluminum 6061 T6** and a thickness of **0.1 in.**

Top Bracket:

$$\sigma_t = \frac{F}{A} = \frac{1,200 \text{ lbf}}{(0.92 \text{ in} - 0.26 \text{ in})(0.1 \text{ in})} = \mathbf{18,182 \text{ psi}}$$
$$n = \frac{\sigma_{\text{allowable}}}{\sigma_{\text{actual}}} = \frac{40,000 \text{ psi}}{18,182 \text{ psi}} = \mathbf{2.2}$$

Bottom Bracket

$$\sigma_t = \frac{F}{A} = \frac{1,200 \text{ lbf}}{(1.44 \text{ in})(0.1 \text{ in})} = \mathbf{8,333 \text{ psi}}$$
$$n = \frac{\sigma_{\text{allowable}}}{\sigma_{\text{actual}}} = \frac{40,000 \text{ psi}}{8,333 \text{ psi}} = \mathbf{4.8}$$

Variables:

F : Force

A : Area

σ_t : Tensile stress

n : Factor of safety

Outcome & Significance: The top seat bracket faces a max tensile stress of **18,182 psi** which means they have a safety factor of **2.2**. While the bottom seat bracket will see a max tensile stress of **8,333 psi** which means they have a safety factor of **4.8**.

Objective: Determine if the seat handles will fail.



Assumptions: The max force applied is 40% of the max impact force calculated in XI.D.14.a, **560 lbs**. This assumption is made because the rear rider will produce a much smaller impact force than the front rider, and a small percentage of that impact force will be absorbed by the handlebars. The force is acting perpendicular to the handles and directly on at the end. The attachment use properties of **Aluminum 6061T6** and tubing with a radius of **1/16 in.** is being used.

$$\sigma = \frac{F_{Rear\ Rider}}{A_{handle}} = \frac{F_{Rear\ Rider}}{L_{Rod}\pi r^2} = 11,200\ psi$$

$$n = \frac{\sigma_{allowable}}{\sigma_{actual}} = \frac{35,000\ psi}{11,200\ psi} = 3.125$$

Variables:

$F_{Rear\ Rider}$: Force from the rear rider

A_{handle} : Area of the handles

σ_t : Tensile stress

n : Factor of safety

Outcome & Significance: The maximum stress seen by the handles is **11,200 psi**, which gives the component a factor of safety of **3.125**.

XII. Manufacturing & Assembly Supplement

XII.A. Manufacturing Processes

Table XII-1: Manufacturing Processes for Parts of Sub-System SS#1

Part #	Manufacturing Process
SS#1-P1	Wood-cutting, Vacuum Bagging
SS#1-P2	Lath to desired geometry, Mill 6 3/8" diameter through holes on rim
SS#1-P3	Lath to desired geometry, Mill 6 1/4" diameter holes on disk, Mill 4 3/8" diameter holes at end
SS#1-P4	Purchased
SS#1-P5	Purchased
SS#1-P6	Purchased
SS#1-P7	3-D Print Mold, Liquid Rubber Casting
SS#1-P8	Lath to desired geometry, Mill 6 3/8" diameter through holes on rim, Mill 4 3/8" diameter holes at end

Table XII-2: Manufacturing Processes for Parts of Sub-System SS#2

Part #	Manufacturing Process
SS#2-P1	Purchased
SS#2-P2	Purchased
SS#2-P3	Welding & Water Jetting
SS#2-P4	Welding & Water Jetting
SS#2-P5	Welding
SS#2-P6	CNC Milling
SS#2-P7	CNC Milling
SS#2-P8	CNC Milling
SS#2-P9	CNC Milling
SS#2-P10	CNC Milling
SS#2-P11	CNC Milling
SS#2-P12	CNC Milling
SS#2-P13	CNC Milling
SS#2-P14	CNC Milling

Table XII-3: Manufacturing Processes for Parts of Sub-System SS#3

Part #	Manufacturing Process
SS#3-P1	Purchased
SS#3-P2	Purchased
SS#3-P3	Purchased
SS#3-P4	Reused from previous model
SS#3-P5	Purchased

SS#3-P6	Purchased
SS#3-P7	Purchased
SS#3-P8	Purchased
SS#3-P9	Donated
SS#3-P10	Purchased
SS#3-P11	Reused from previous model
SS#3-P12	Reused from previous model

Table XII-4: Manufacturing Processes for Parts of Sub-System SS#4

Part #	Manufacturing Process
SS#4-P1	CNC Milling
SS#4-P2	Mill to correct length. Bore 3/8-inch holes at ends
SS#4-P3	Mill end to the correct angle. Bore 3/8-inch hole at end
SS#4-P4	Mill Raw Aluminum into 4 equal pieces. Bore and thread 4 1/4-inch holes in corners
SS#4-P5	Mill to desired geometry.
SS#4-P6	Mill aluminum rod into sections of desired length. Mill handle's knuckle connection end. Bore 1/4-inch hole at knuckle connection end. Mill ends of sections to desired angles to be welded together.
SS#4-P7	Mill aluminum rod into sections of desired length. Mill handle's knuckle connection end. Bore 1/4-inch hole at knuckle connection end. Mill ends of sections to desired angles to be welded together.
SS#4-P8	Purchased
SS#4-P9	Purchased
SS#4-P10	Purchased
SS#4-P11	Purchased

Table XII-5: Manufacturing Processes for Parts of Sub-System SS#5

Part #	Manufacturing Process
SS#5-P1	Reused

Table XII-6: Manufacturing Processes for Parts of Sub-System SS#6

Part #	Manufacturing Process
SS#6-P1	Purchased
SS#6-P2	Purchased
SS#6-P3	Water jet and mill cuts to correct geometry and weld to combine the tubing.
SS#6-P4	Water jet and mill cuts to correct geometry and weld to combine the tubing.
SS#6-P5	Purchased
SS#6-P6	Purchased

Table XII-7: Manufacturing Processes for Parts of Sub-System SS#7

Part #	Manufacturing Process
SS#7-P1	Purchased
SS#7-P2	Purchased
SS#7-P3	Cut with a combination of a saw, mill, or waterjet.
SS#7-P4	Cut with a combination of a saw, mill, or waterjet.
SS#7-P5	Cut with a combination of a saw, mill, or waterjet, then welded together.
SS#7-P6	CNC to create geometry then welded to form bracket.
SS#7-P7	CNC to create geometry then welded to form bracket.
SS#7-P8	Cut with a combination of a saw, mill, or waterjet. Drilled for bolt holes then welded.
SS#7-P9	Purchased
SS#7-P10	Purchased
SS#7-P11	Purchased

]

XII.B. Assembly Processes

XII.B.1. Assembly Process for Sub-System SS#1 – Solid Disk Wheels

The first step in the assembly process for the wheels will consist of: cutting disks out of balsa wood equal to the overall diameter of the wheel (SS#1-P1). Once cut, multiple sheets of balsa wood will then be lightly glued together, using only dollops to keep them in place. When the desired thickness is reached, holes will be bored to specified locations on (SS#1-P1) where the bolts and hubs will be. Carbon fiber will then be put across the face of the wheel with some overhang on the side of the wheel where epoxy resin will then be painted across the carbon fiber to bond with the balsa wood. This process will be repeat for 3 plies of carbon fiber. After, the team will set up the vacuum bagging process and use it to ensure a good bond between the carbon fiber and balsa wood. The aluminum hubs (SS#1-P2) will start off as 4 ½" diameter round stock and then lathed. Holes will be bored using a mill after the lathing process. The hub connector (SS#1-P3) will start off as 2 ¼" diameter round stock, lathed, and then bored using a mill for the holes; four holes on the face of the connector will need to be threaded.

The outer wheel hub (SS#1-P8) will be held on the outer side of the wheel (SS#1-P1) while the inner wheel hub (SS#1-P2) will be held on the side opposite from the outer wheel hub. Making sure that the holes are all aligned, the three pieces will be mated together using bolts and nuts (SS#1 P5 &6) in the 6 outer diameter holes. The brake disc (SS#1-P4) will slide onto the hub connector (SS#1-P3). The hub connector (SS#1-P3) will then be mated to the outer wheel hub (SS#1-P8) using a hex screw bolt and the thread holes in the hub connector. The hub connector will connect with the u-joint of the drivetrain via a key and keyway connection.

XII.B.2. Assembly Process for Sub-System SS#2- Single Bar Frame

The first step in the assembly process for the frame will consist of cutting the purchased rectangular tubing (SS#2 P1-P4) into four sections. Next, the angled frame pieces (SS#2 P3,P4) that will support the drivetrain cranksets will be cut at a 30° angle. After cutting, they will be welded to the appropriate positions on top of the single beam frame. Once mounted and the weld is cured, holes will be bored through the angled tubes to support the drivetrain bottom brackets. SS#2 P6-12 will be CNC milled and water-jetted as appropriate to fit the specifications on the engineering drawings.

XII.B.3. Assembly Process for Sub-System SS#3- Front Tandem Drivetrain

The first step in the assembly process for the drivetrain will consist of boring holes into the frame equal to the outer diameter of the bottom bracket shells (SS#3-P8). Once cut, the bottom bracket shells will be inserted into the holes and welded circumferentially to fix them to the frame. The square-tapered bottom brackets (SS#3-P7) will then be threaded into the bottom bracket shells. Once threaded into place, the 7" cranksets (SS#3-P3) will mate to the bottom bracket via a male-to-female tapered square fitting. The 7" cranksets will consist of a Surly 36T cog (SS#3-

P1) on the crankset's spider. Next, the crankarms on the two cranksets that compose the timing (tandem) feature are to be cut off as they will not be used as input cranksets.

The first timing crankset (closer to the front of the rover) will consist of three Surly 36T cogs (SS#3-P1). Two of these 36T cogs will be on the right of the frame, and the third will be on the left. These cogs will be bolted to the corresponding crankset's spider. The farther of the two on the right side will be the driven cog from the front rider's input. The neighboring 36T cog will be the timing cog, and the left-hand side 36T cog will serve as the input cog to the NuVinci N380 Internally-Geared hub (SS#3-P9).

The input side of the NuVinci N380 Internally-Geared hub will consist of a Surly 18T cog (SS#3-2). This cog will mate to the pre-fabricated mating region on the hub. The output side of the internally-geared hub will also consist of a Surly 18T cog (SS#3-P2) that will mate to an integrated six-bolt pattern brake disc adapter. All cogs will be connected externally via an ASTM #40 Standard Roller Chain (SS#3-P5). The chain will be cut to the appropriate length with a chain cutting tool.

Driveshaft assembly will consist mostly of reuse and modification of the previous year's rover model. The driveshaft will consist of an aluminum driveshaft (SS#3-P11) supported by two 206 Series Pillow-Block Bearings (SS#3-P12). The 206 Series Pillow-Block Bearings (SS#3-P12) will be mounted to the frame with two bolts 180° apart and parallel to the frame. The aluminum driveshaft will couple to the telescopic u-joints (SS#3-P4) via both set screws and keyway/key mates. The opposite end of the telescopic u-joints will mate to the shaft portion of the wheel hub in a similar fashion. A Surly 18T cog (SS#3-P2) will be keyed to the driveshaft and aligned with the output cog of the internally-geared hub for the final power transmission from the drivetrain to the output of the system.

XII.B.4. Assembly Process for Sub-System SS#4- Push-Pull Steering

The steering knuckle (SS#4-P1) will be CNC Milled by the LSU Machine Shop. This steering knuckle has cut outs to accommodate the steering handles (SS#4-P6,P7), the drivetrain u-joints (SS#3-P4), and the suspension control arms (SS#6-P3,P4).

Four bolt plates (SS#4-P4) will be milled into their rectangular shapes from a ¾ inch thick sheet of raw aluminum. Four ¼-inch holes will be bored and threaded into the four corners of the bolt plates, as is specified in the Bolt Plate manufacturing drawing. Two bolt plates will be welded directly onto the steering knuckle, in line with the knuckle's u-joint cut out. The Range of Motion Limiters (SS#4-P5) will be welded onto the rear side of the steering knuckle above the bolt plate. The tie rod linkage will be assembled as follows: The tie rod (SS#4-P2) and two steering arms (SS#4-P3) must be milled to the correct geometry. Next, 3/8-inch holes will be bored at the ends of the tie rod and steering arms, as is specified in their respective manufacturing drawings. The angled edge of each steering arm must be welded onto the center of a bolt plate. The steering arms will then be bolted to their companion bolt plates. The companion bolt plate will be welded directly onto the steering knuckle. This design allows the tie rod linkage to be removed and reattached as the team pleases. This will be useful if the team decides to change the geometry of the tie rod linkage to accommodate different turning behavior. A 3/8-inch bolt will join the tie rod to the steering arms. The holes must not be threaded as to allow for unhindered motion.

Finally, the steering handles (SS#4-P6,7) will be assembled. First a raw aluminum rod must be cut into sections of appropriate length, as specified by their manufacturing drawings. Next the steering knuckle connection end must be milled to its required shape, as specified by the manufacturing drawings. This shape will allow the steering handles to slide into their steering knuckle cut outs. Next, a 1/4 -inch hole must be bored at the steering knuckle connection end, as specified by their manufacturing drawings. Next, the ends of the handle sections must be cut at their appropriate angles. At this point, the sections can be welded together to form a complete handle. Finally, the handle can slide into the steering knuckle. The handles will be secured by a 1/4 bolt that goes through the top of the steering knuckle and through the hole bored at the steering handle's connection end.

XII.B.5. Assembly Process for Sub-System SS#5- Hydraulic Brakes

The hydraulic disc brake caliper and physical disc will be mounted to the wheel hub. The caliper will be attached to the brake fluid hose, which will attach to the master cylinder. The master cylinder will have an input hose from the brake lever, which will be positioned on the front rider's left steering arm. The entire subsystem will be reused directly from the 2019 rover.

XII.B.6. Assembly Process for Sub-System SS#6- Double Wishbone Suspension

The double wishbone suspension will have one assembly method that will be replicated four times. The top and bottom A-frame sections (SS#6-P3, P4) will be cut and shaped using a mill and water-jet combination, before being welded together. Ball joints ends (SS#6-P6) will be used on to attach the front A-frames to the steering knuckle (SS4-P1) to allow for not only vertical, but rotational movement as well. Clevis rod ends (SS#6-P5) will be used for the rear suspension systems to restrict the motion to the vertical plane only. The shock absorbers (SS#6-P1, P2) will be bolted at both ends, the bottom to the suspension's A-frame (SS#6-P4) and the top to the frame's pad-eye attachments (SS2-P1, P2).

XII.B.7. Assembly Process for Sub-System SS#7- Seats

The assembly of the seats is very straight forward. The bracket attachments (SS#7-P6, P7) will be cut the proper shapes using a CNC, then welded to together to form the bracket. These brackets will then be connected to the seats (SS#7-P1) using the hardware kit provided by the seat manufacturer (SS#7-P2).

Once the brackets are in place the seat bars (SS#7-P3, P4) will be cut to the proper lengths. Once the bars are cut, they will be drilled for bolts. The front seat bar (SS#7-P3) will have one hole drilled at the top of it to place a bolt (SS#7-P9) to connect it to the top bracket (SS#7-P6). Before the front seat bar is connected though it will need to be welded to (SS2-P1).

The rear seat bar (SS#7-P4) will have a bolt hole at the top of it (similar to SS#7-P3) but will be bolted at two points to SS#7-P8. The rear seat holder (SS#7-P8) will be cut to the correct dimensions using a saw and mill combination then the two holes will be drilled for the bolts (SS#7-P9). The seat holder will be welded together to hold its form, then welded to (SS2-P2).

The bottom seat to frame attachments (SS#7-P5) will be manufacture, by cutting a rod in half to create two separate pieces. Each of these pieces will be drilled to fit bolts (SS#7-P10) then they will be welded to the frame (SS2-P1, P2). Once all the pieces are cut and welded together the assembly can be put together using 6 bolts (SS#7-P9, P10) in the previously specified through holes.

XII.C. Testing & Validation Supplement

XII.C.1. Design Phase Testing Methods and Details

All testing methods utilized in the testing of previous LSU rovers were created on an as-needed basis. Hence, no professionally-recognized standards were used in the design phase testing stage. Instead, Team 24 first determined the desired outcome of the test being performed (i.e. average braking distance, average rover output speed, average cadence over an 8-minute interval, etc.) and then designed the test accordingly. Design-phase testing was completed on the Louisiana State University campus at varying locations within. Most of the tests which required a concrete surface were conducted south of LSU's Tiger Stadium. Tests which required surfaces other than concrete were completed outside of LSU's Patrick F. Taylor Hall. The results of these design-phase tests are illustrated in XII.C.2 below.

XII.C.2. Design Phase Testing Results

	Max Speed			
Test	20 ft	40 ft	60 ft	80 ft
Mike 1	4.15	1.93	1.52	1.46
Mike 2	3.49	1.83	1.47	1.27
Peter 1	3.24	2.61	1.79	1.69
Matt 1	3.62	2.33	1.5	1.75
Matt 2	3.24	1.75	1.56	1.3
Garrett 1	3.39	2.26	1.8	2.57
Dakota	3.1	2.24	1.45	1.31
Evan	2.96	1.98	1.54	1.33
Lucy	4.09	2	1.66	1.23

	Braking		
Test	Braking Distance (ft)	Braking Time	Deacceleration (ft/s^2)
Mike 1	6	1.27	7.44
Mike 2	1	1.07	1.75
Peter 1	10	1.69	7.00
Matt 1	4	1.37	4.26
Matt 2	5	1.24	6.50
Garrett 1	3	0.63	15.12
Dakota	2.75	1.17	4.02
Evan	2.75	1.27	3.41
Lucy	3	1.69	2.10
Avg	4.17	1.27	5.73

	Distance (ft)					
Test	20	40	60	80	Velocity (ft/s)	Acceleration (ft/s^2)
Mike 1	4.15	6.08	7.6	9.06	12.3	1.95
Mike 2	3.49	5.32	6.79	8.06	13.1	2.46
Peter 1	3.24	5.85	7.64	9.33	9.9	1.84
Matt 1	3.62	5.95	7.45	9.2	10.9	1.89
Matt 2	3.24	4.99	6.55	7.85	12.9	2.60
Garrett 1	3.39	5.65	7.45	10.02	9.2	1.59
Dakota	3.1	5.34	6.79	8.1	11.9	2.44
Evan	2.96	4.94	6.48	7.81	12.3	2.62
Lucy	4.09	6.09	7.75	8.98	12.1	1.98
				Avg (ft/s)	11.6	2.15
				Avg (mph)	7.92	

Team Members		Strength Test Leg Press		
Name	Weight	Max Weight	Reps (Goal 10 Reps)	
Matthew Donner	190 lbs	568 lbs	10 / 10	
Garrett Gremillion	195 lbs	388 lbs	10 / 10	
Lucy Guo	140 lbs	208 lbs	10 / 10	
Dakota Hebert	145 lbs	388 lbs	10 / 10	
Peter Pham	178 lbs	478 lbs	10 / 10	
Evan Prejean	145 lbs	348 lbs	10 / 10	
Mike Smith	220 lbs	658 lbs	10 / 10	

Endurance Test Bicycle		
Resistance	Time (Goal: 8 Minutes)	Distance
15 / 25	8 Min	2.44
15 / 25	8 Min	2.47
15 / 25	8 Min	2.21
15 / 25	8 Min	2.58
15 / 25	8 Min	1.81
15 / 25	8 Min	2.41
15 / 25	8 Min	2.45

Strength Factor (SF)	Endurance Factor (EF)	
Max Weight/Weight	Distance/Time	
2.989	18.30	mph
1.990	18.53	mph
1.486	16.58	mph
2.676	19.35	mph
2.685	13.58	mph
2.400	18.08	mph
2.991	18.38	mph

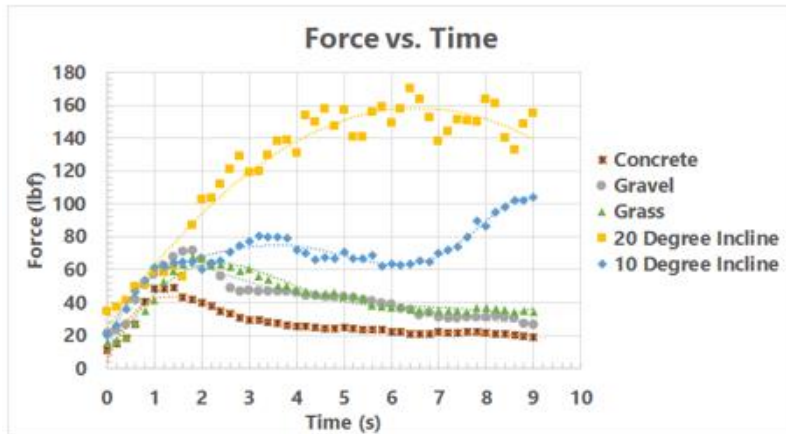
Strength Factor Scaled (SFS)	Endurance Factor Scaled (EFS)
9.9952	9.4574
6.6526	9.5736
4.9674	8.5659
8.9467	10.0000
8.9785	7.0155
8.0243	9.3411
10.0000	9.4961

$$SFS = \frac{SF_{INDIVIDUAL}}{SF_{MAX}} * 100$$

$$EFS = \frac{EF_{INDIVIDUAL}}{EF_{MAX}} * 100$$

$$Score_{OVERALL} = SFS * 0.35 + EFS * 0.65$$

Overall Score		
Name	Overall pts (Max = 10 pts)	Rank
Matthew Donner	9.65	2
Garrett Gremillion	8.55	5
Lucy Guo	7.31	7
Dakota Hebert	9.63	3
Peter Pham	7.70	6
Evan Prejean	8.88	4
Mike Smith	9.67	1



	Max Force (lbf)	Average Force (lbf)
Concrete	49.1	25.6
Gravel	71.7	42.4
Grass	68.0	43.4
20 Degree Incline	170.5	123.3
10 Degree Incline	103.8	69.7



XII.C.3. Preliminary Validation Testing Methods and Details



Figure XII-XII.C.1: Liquid Penetrant Non-Destructive Testing [6]

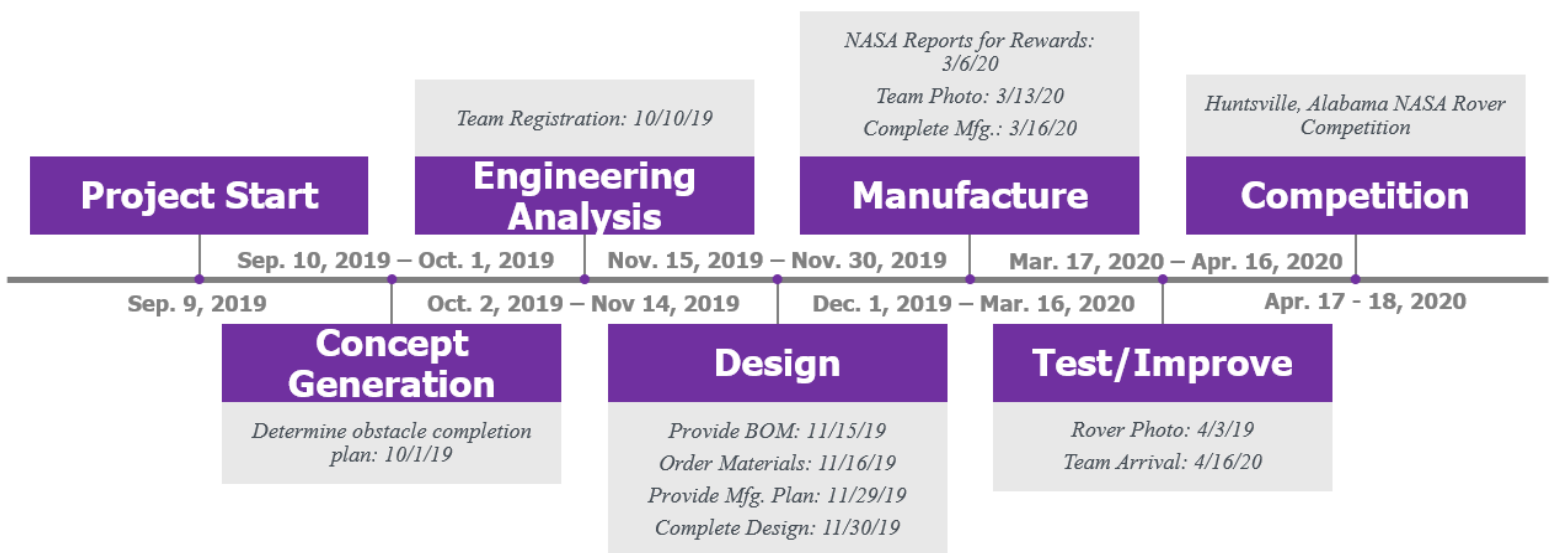


Figure XII-XII.C.2: LSU's Phased Array Ultrasonic Testing [20]

XII.D. Project Management Supplement

XII.D.1. Schedule and Milestones Details

The most significant milestone for Team 24 is the NASA Human Exploration Rover Challenge. The Challenge takes place on April 17 and 18, 2020 in Huntsville Alabama. To be successful during the competition, the Team imposed a milestone to complete manufacturing by March 16, 2020. This would allow Team 24 a month to adequately test and improve the rover for competition. In order to complete manufacturing by March 16, 2020 Team 24 provided a Bill of Materials (BOM) on November 15, 2019. Additionally, Team 24 has provided a manufacturing plan on December 11, 2019. If the Team had not provided a BOM or manufacturing plan by the specified dates, the manufacturing completion date would need to be pushed back. Therefore, the BOM and manufacturing plan were the critical path for the project to be completed successfully. The current critical path is completion of manufacturing by March 16, 2020. The most significant point of failure for the project is inadequate time to test and improve the newly designed rover.



Man Powered Rover

Team 24

Project Lead: Garrett
Gremillion

Project Start:

Mon, 9/9/2019

Display Week:

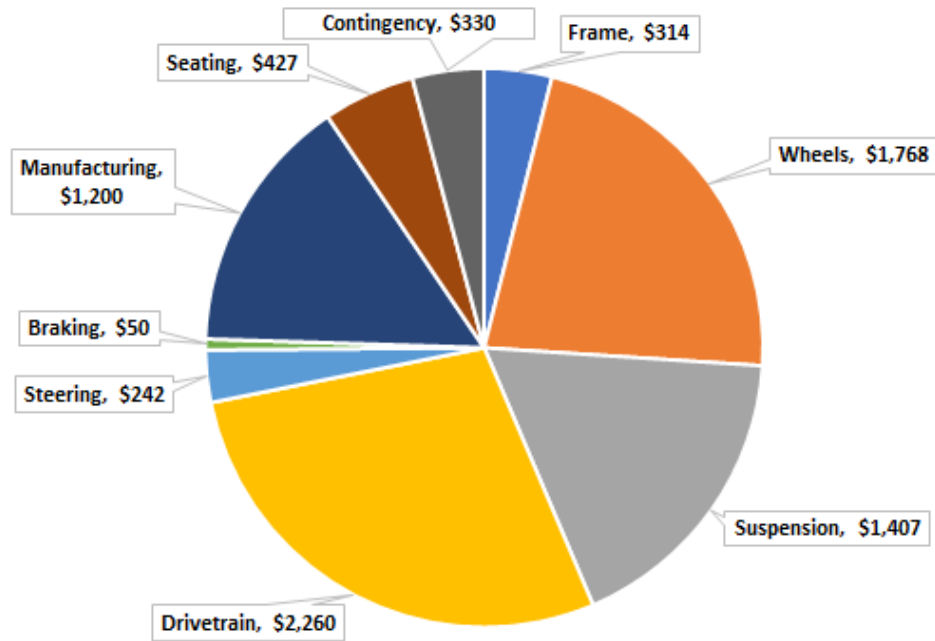
1

TASK	ASSIGNED TO	PROGRESS	START	END
Research				
Perception Presentation	Team	100%	9/9/19	9/12/19
E. Spec. Presentation	Team	100%	9/12/19	9/16/19
ESIR	Team	100%	9/12/19	9/23/19
Concept Generation	Team	100%	9/23/19	9/28/19
Concept Generation				
Concept Evaluation	Team	100%	9/28/19	10/2/19
Concept Selection	Team	100%	10/2/19	10/5/19
Midterm Presentation	Team	100%	9/23/19	10/2/19
Obstacle Completion Plan	Team	100%	9/23/19	10/1/19
Embodiment Report	Team	95%	10/2/19	10/15/19
Midterm Peer Eval	Team	0%	10/14/19	10/23/19
Engineering Analysis				
Team Registration	Team	100%	10/10/19	1/16/20
Drivetrain Analysis	GG & EP	20%	10/4/19	10/28/19
Wheel Analysis	LG & PP	20%	10/4/19	10/28/19
Frame Analysis	MS	20%	10/4/19	10/28/19
Steering Analysis	MD & DH	20%	10/4/19	10/28/19
Suspension & Braking Analysis	MD & DH	20%	10/4/19	10/28/19
Eng. Analysis Tech. Brief	Team	0%	10/14/19	10/28/19
Design				
Final Proj. Presentation	Team	0%	10/28/19	12/1/19
Project Poster	Team	0%	10/28/19	12/1/19
Final Design Proposal Report	Team	0%	10/28/19	12/6/19
Bill of Materials	Team	0%	10/28/19	11/15/19
Order Materials	Team	0%	11/15/19	12/1/19
Provide MFG plan	Team	0%	10/28/19	11/29/19
Complete Design	Team	0%	10/28/19	11/30/19
Manufacturing				
Nasa reports for awards	Team	0%	2/6/20	3/6/20
Team Photo	Team	0%	2/13/20	3/13/20
Manufacture Rover	Team	0%	12/6/19	3/16/20
Testing/Improvement				
Test & Improve Rover	Team	0%	3/17/20	4/15/20
Rover Photo	Team	0%	3/17/20	4/3/20
Team Travel	Team	0%	4/15/20	4/16/20
Competition				
Rover Competition	Team	0%	4/17/20	4/18/20

Figure XII.D.1: Gantt Chart with Detailed Project Time-Line

XII.D.2. Budget Details

2020 Preliminary Budget, \$8,000



2020 Final Budget, \$8,000

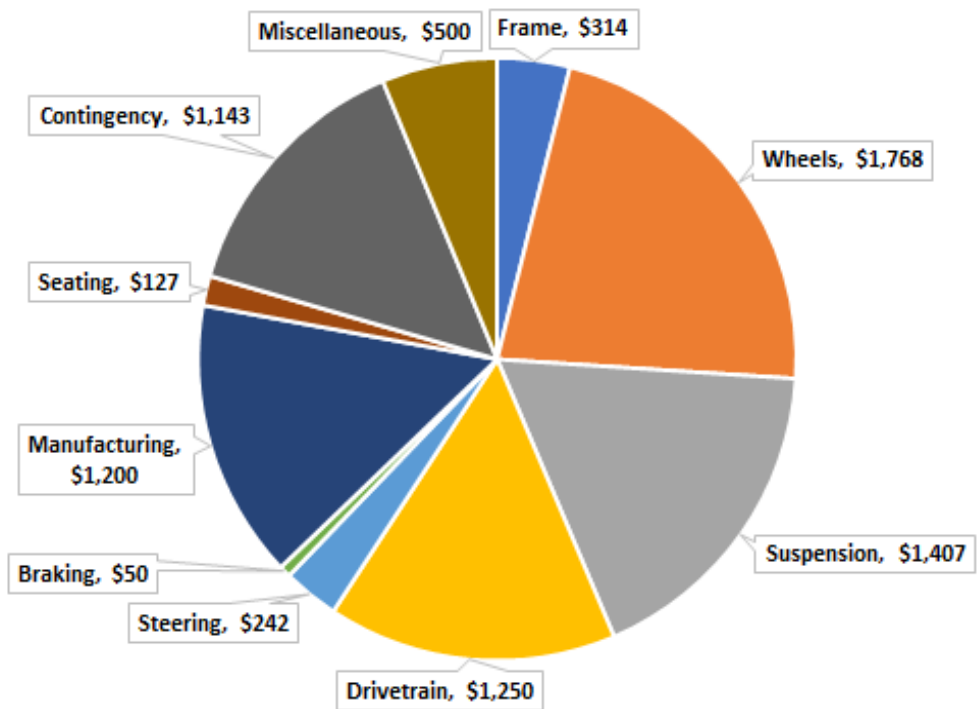


Table XII-8: Bill of Materials/Parts

Part No. (SS#-P#)	Part Name	Quantity	Price (per unit)	Cost Total (no shipping)	Vendor and Ordering Info.
SS#1-P4	Braking Disc	2	\$15.99	\$0	Reused from previous LSU Teams
SS#1-P5	3/8 in. nut	24	\$0.08	\$1.92 \$7.04	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#1-P6	3/8 in. bolt	40	\$0.98	\$39.20	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#3-P1	Surly 36T Cog	7	\$26.25	\$183.75	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P2	Surly 18T Cog	2	\$26.25	\$52.50	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P3	7" Crankset	5	\$41.25	\$206.25	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P5	ASTM #40 Standard Roller Chain	5	\$15.00	\$75.00	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P6	Hub Anchor	2	\$9.75	\$19.50	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P7/P8	Square-tapered bottom bracket	4	\$22.50	\$90.00	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#3-P10	Surly Singelator Chain Tensioner	3	\$37.50	\$112.50	The Bicycle Shop 3315 Highland Rd., Baton Rouge, LA, 70802
SS#4-P1	Steering Knuckle	2	\$32.90	\$65.80	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P2	Tie Rod	1	\$28.08	\$28.08	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P3	Steering Arm	2	\$2.81	\$5.62	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P4	Bolt Plate	4	\$12.43	\$49.71	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P5	R.O.M.Limiter	2	\$9.67	\$19.33	McMaster-Carr

					200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P6	Left Steering Handle	1	\$30.87	\$30.87	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P7	Right Steering Handle	1	\$30.87	\$30.87	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P8	Bolt Plate bolt	10	\$0.98	\$9.82	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P9	Tie rod bolt	2	\$1.26	\$2.52	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P10	Bolt plate nut	10	\$0.08	\$0.80	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#4-P11	Tie rod nut	2	\$0.15	\$0.30	McMaster-Carr 200 Aurora Industrial Pkwy. Aurora, OH 44202-8087
SS#6-P7	Rockshox Vivid R2C	2	\$0	\$0	Reused from previous LSU Teams
SS#6-P8	Rockshox Vivid Air R2C	2	\$499.99	\$0	Finish Strong Endurance
SS#6-P1,P2	A-Frame Suspension Tubing	7	\$27.42	\$191.94	McMaster-Carr
SS#6-P3	A-Frame Connector, Aluminum Bar	1	\$53.26	\$0	McMaster-Car
SS#6-P5	Clevis Rod ends	4	\$5.35	\$21.40	McMaster-Carr
SS#6-P6	Ball joint rod ends	4	\$16.50	\$0	McMaster-Carr
SS#7-P12	Volae Comfort Carbon Seat Shell	2	\$0	\$45	Hostel Shoppe
SS#7-P2	Volae Comfort Carbon Hardware kit	2	\$28.00	\$0	Hostel Shoppe
SS#7-P6, P7	Aluminum sheet	3	\$16.02	\$48.06	McMaster-Carr
SS#7-P9	Bolts, 2 3/4" long, zinc-	4	\$10.41	\$0	McMaster-Carr

	plated 1/4" – 20 thread				
SS#7- P10	Bolts, 2 1/2" long, zinc- plated 1/4" – 20 thread	2	\$12.53	\$0	McMaster-Carr
SS#7- P11	Medium- strength steel hex nut, zinc plated 1/4" – 20 thread	6	\$3.10	\$0	McMaster-Carr

XII.E. Reference Materials

XII.E.1. References and Bibliography

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