MACDON ROTARY HEADER HITCH SUPPORT DESIGN

MECH 4860 – Engineering Design

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Dear Mr. Hakansson

I am pleased to inform you that our team has successfully redesigned a universal support device for MacDon's pull-type rotary header.

Enclosed is our report entitled, *MacDon Rotary Header Hitch Support Design.* The date that we will submit the report is December 7, 2016. This report provides information on the background of the project and a through explanation of the design process undertaken to achieve a final support design. In addition to the report, the team will be providing you with the final deliverables as requested:

- CAD Models
  - Step files of V-support components
  - Step file of V-support assembly
  - Step file of extended driveline guard bar
  - Step file of V-support in updated master assembly
- ANSYS FEA
  - File of V-support analysis of all three loading scenarios
  - File of Driveline Guard Bar analysis
  - Step files of both finite element models
- Detailed Design Drawings
  - SolidWorks file of drawing package
  - PDF file of drawing package

We are confident that this report satisfies all of MacDon's requirements. We would like to thank Malcom Xing and yourself for your contributions to the success of this project and report. If there are any comments or concerns please feel free to contact any member of Team 7.

Regards,

Team 7:

Jessica Tjahjadi Kyle Szaura Ray Davis Rhys Hatherly

Executive Summary

The objective of this project, as specified by MacDon Industries Ltd, was to design a universal support device for their pull-type rotary header. The V-support design provides clearance for the driveline, improves stability, reduces setup/ removal time and allows for easier connection to the tractor. This report delivers hand calculations, CAD models of the assembly, FEA modelling that validates the design and detailed design drawings. Our final design meets the target specifications outlined by the client and addresses all the required needs.

The final design successfully replaces the existing trailer jack stand and built-in stand on the drawbar and 2-point hitch style header, respectively. This final V-support design is a product of five earlier versions of the design which were each refined through a combination of client feedback and necessary design improvements. 1.50 [in] ASTM A513 Grade B tubing was used to construct the bulk of the V-support design from, enabling the final design to weigh in at 3.80 [kg]. To ensure no interference with tractor tires, the driveline guard bars were extended 12 [in] forward, with the V-support being directly mounted and welded to the front most horizontal portion of the extended driveline guards. The V-support was designed to also have the same mounting height for the trailer jack, allowing the existing trailer jack model to be used. Analytical verification confirmed this design supports the 1500 [lbs] weight of the swing tongue while maintaining a factor of safety of 3.30, deeming this fully functional design safe for use.

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1.0 Introduction

This Final Design Report outlines the design process taken to create a universal support device for MacDon's pull-type rotary header. Specifically, this report will summarize the problem definition and concept development phases of the project, as well as provide an in-depth analysis of the steps taken during the detailed design phase. These steps include a review of the concept generation and selection phase, failure mode effect and analysis, analytical analysis, numerical analysis including preliminary finite element analysis, failure analysis and preliminary engineering drawings. At the end of this report, the team will provide an overview of the final design and the steps taken to achieve the client's needs and deliverables.

1.1 Nomenclature

Before getting too far into the report, it is important for the reader to understand the nomenclature discussed and referenced throughout this document. Figure 1 below presents an image of the pull-type rotary header swing tongue, labelled with its important features, which will be mentioned throughout the report.

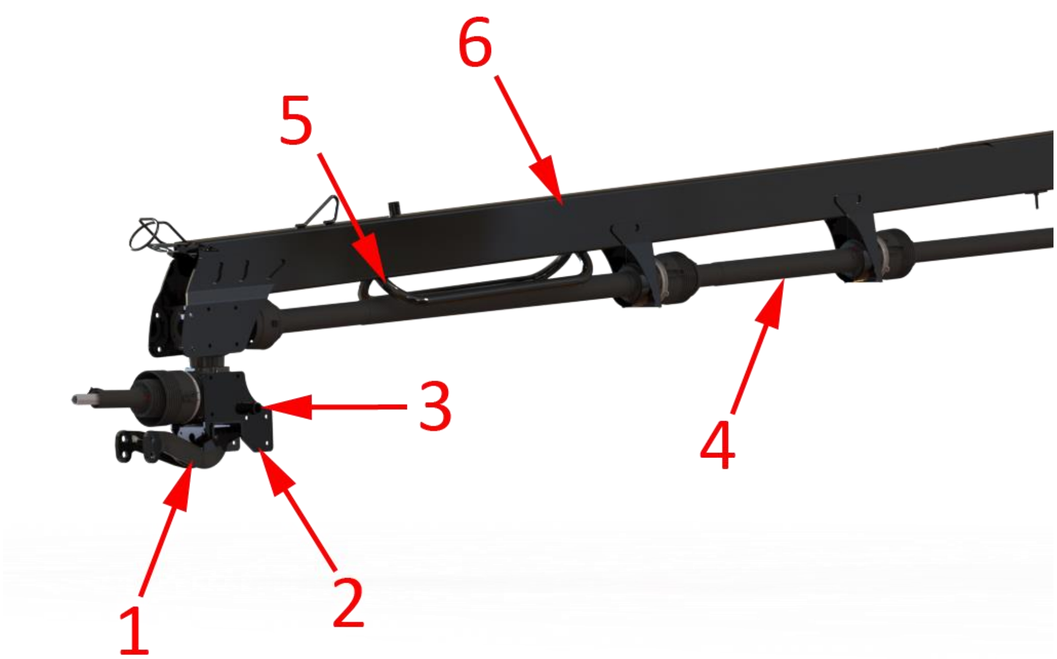

Figure 1: Pull-type header swing tongue and its important features

The numbered items are as follows:

1. **Hitch:** There are two different hitches the pull-type rotary header uses to connect to the tractor. The one currently displayed in Figure 1 is the drawbar, the other hitch variant is the 2-point.
2. **Swivel gearbox:** The gearbox is the power transmission system of the pull-type rotary header.
3. **Gearbox jack mount:** The mount that the current jack uses to support the header.
4. **Driveline:** The system that connects the gearbox (transmission) to the header.
5. **Driveline guard bars**: Bars that extend down from the swing tongue to protect the driveline from incidental contact with the tractor tires, if the tractor is to jack knife.
6. **Swing tongue:** Boom arm that extends the header from the tractor. Contains the mechanical system of the pull-type header and hydraulic lines.

1.2 Background

Agriculture and the agri-foods industry contribute over $100 billion annually to Canada's gross domestic product [1]. In order to produce this revenue, farmers require the best agricultural equipment there is to offer. MacDon Industries Ltd. manufactures and designs world-class harvesting equipment that provides quality, reliability and, most importantly, productivity to their customers. With the goal of continuous improvement, MacDon is committed to innovative and unique solutions that boost productivity, improve systems and innovate products. One such product is the MacDon pull-type rotary header, depicted in Figure 2.

Figure 2: MacDon pull type rotary header [2].

MacDon's pull-type header offers a unique design. The design trails the tractor when in operation and features MacDon's revolutionary disc cutterbar, multiple conditioning options, a responsive float system and the ability to quickly transition into transport mode [4]. These features provide a clean cut on uneven ground and allow the operator to move easily from field to field. The pull-type header also comes in two hitch styles, drawbar and 2-point, which are shown below in Figure 3 and Figure 4 respectively.

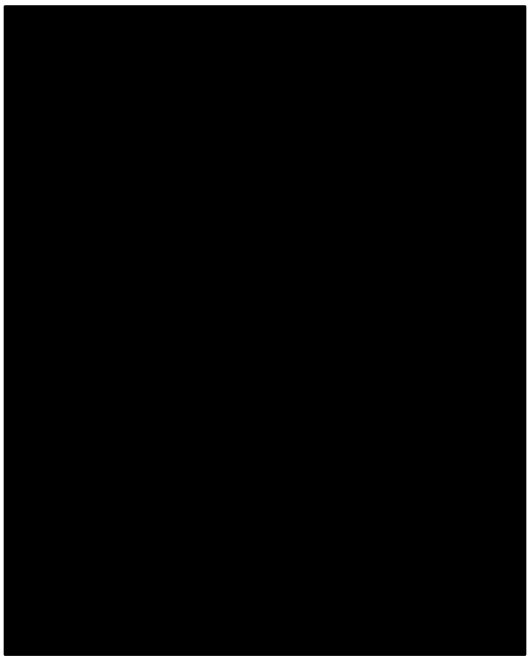

Figure 3: Drawbar hitch stand [3].

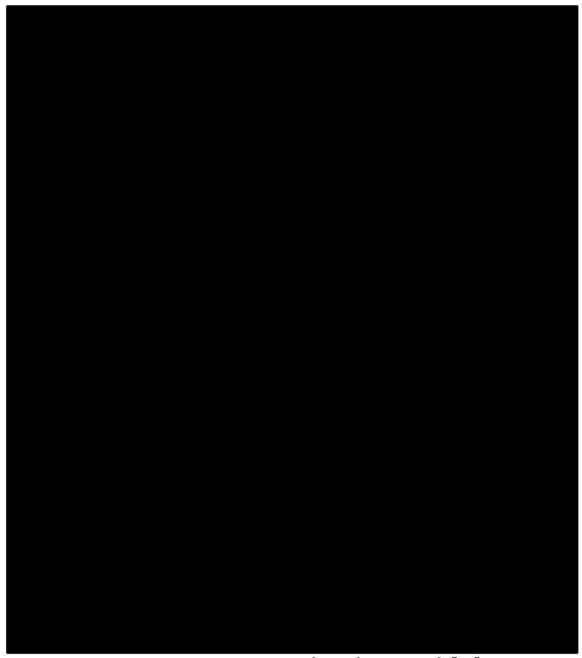

Figure 4: 2-point hitch stand [4].

When the drawbar style is not operational (downtime or out of season), the hitch is supported by a trailer jack. When the 2-point hitch configuration is not operational, it is supported by a built in stand.

1.1.1 Current Support for the Hitches

The existing drawbar hitch stand is a conventional trailer jack that can be purchased at any local hardware or auto parts store. Specifically, the jack utilized in the current drawbar hitch design is the BULLDOG Tubular Swivel trailer jack. It has a rated lift capacity of 3,000 lbs with a static load capacity of 5,000lbs [5]. The jack is readily available and it is easy to store. When the pull-type header is operational, the trailer jack is mounted on the hitch using a pin connection.

During down time the jack is removed from the swing tongue and mounted to the gearbox using the same connection method. This mounting location can swivel and allows for movement of the header. Both mounting locations can be seen in Figure 5 and Figure 6.

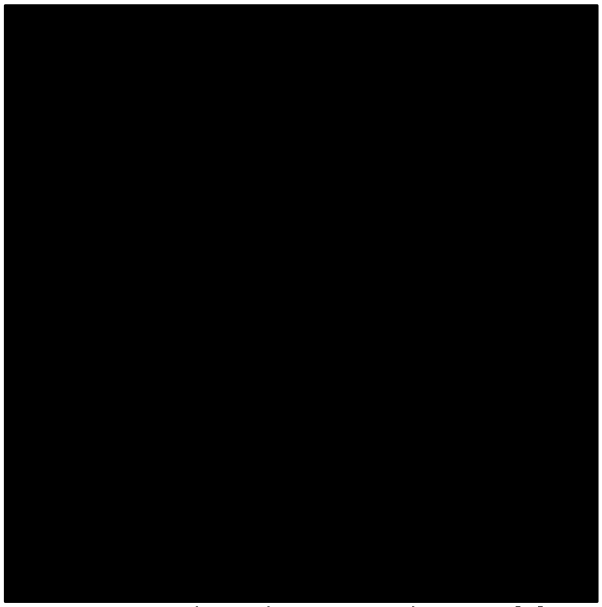

Figure 5: Trailer jack in storage location [6].

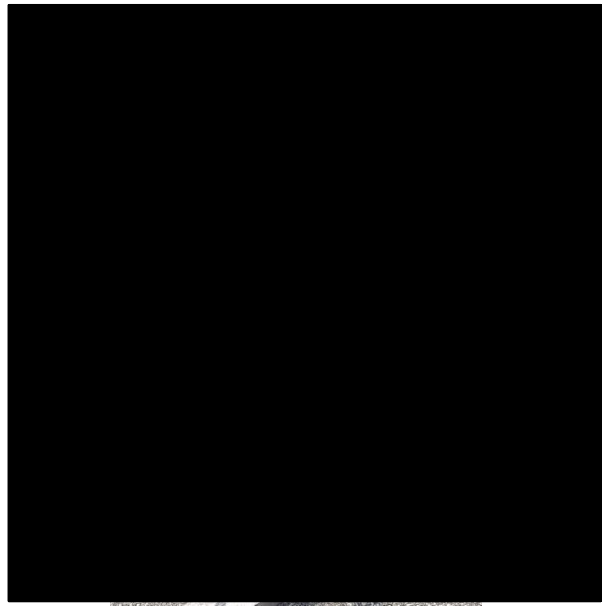

Figure 6: Trailer jack in operational position [7].

The 2-point hitch utilizes a built in stand that is mounted to the back of the gearbox. Images of the built in stand can be seen in Figure 7 and Figure 8.

Figure 7: 2-point stand in upright position [8].

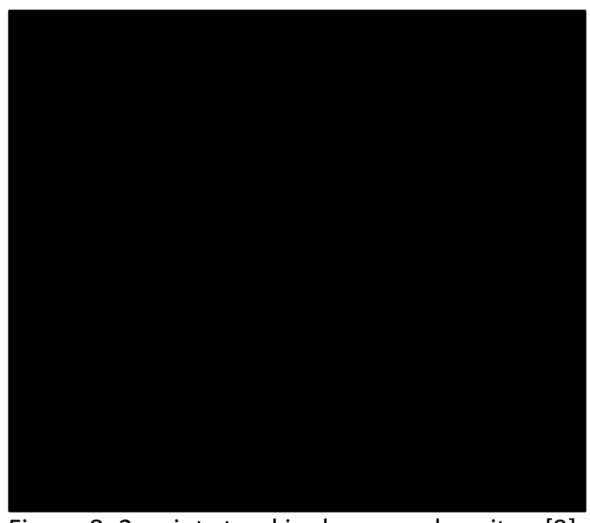

Figure 8: 2-point stand in downward positon [9].

When the header is operational, the stand is rotated 90 degrees upward and locked in position using a pin, as shown in Figure 7. During downtime the pin is removed, the stand is rotated downwards and locked in place as shown in Figure 8.

1.3 Problem Statement

Despite the current stand meeting the operational requirements, operators of the equipment have raised issues with the current designs. These issues include:

- Clearance with the stand and driveline as the gearbox can rotate causing the jack handle to impact the driveline.
- Instability of the header due to the gearbox rotation.
- Set up and removal of the stand are time consuming.
- Movement of the gearbox creates an inconvenience for the operator when hooking up to the tractor.

In order to solve these problems, the design team has been tasked with a series of objectives defined by the client in order to successfully innovate or redesign the existing hitch stands.

1.4 Project Objectives

The purpose of this design project is to do a complete design and analysis of a universal support device for both hitch styles of MacDon's rotary headers. Compared to the current design, the new device must provide a more time-efficient setup and removal while allowing for operation by a variety of operators. From a cost perspective, the new device should not cost more than the current hitch stands. The design will support the combined weight of the swing tongue and hitch while not impeding the driveline operation. Another aspect that will be improved upon includes maintaining stability in all weather and ground conditions.

1.5 Target Specifications

Upon meeting with MacDon's representative for this project, a list of customer needs was generated in order to give a better visualization of the design requirements. These needs were reflected upon throughout the design process to ensure every criterion is considered at every step. In order to prioritize the customer needs, an importance rating was given using a ranking scale from 1 to 5 (1 being of low importance and 5 being of high importance). TABLE I shows the list of 12 customer needs along with their importance rating, agreed upon by the team and client.

TABLE I: CUSTOMER NEEDS AND THEIR IMPORTANCE

#	Needs	Importance
1	The stand is lightweight	1
2	The stand is cost-effective	5
3	The stand is technically compliant	5
4	The stand is easily accessed for maintenance	3
5	The stand is stable in all directions when used	5
6	The stand is compatible with current products	4
7	The stand is durable	3
8	The stand is easy to store	1
9	The stand allows easy parts replacement	3
10	The stand provides sufficient operation clearance at all positions	5
11	The stand works in various condition	3
12	The stand is easy to use	4

After compiling the needs, the team developed our target specifications, which consist of metrics and units. The metrics were derived from the needs as a means of further evaluating each need in detail. In order to evaluate each metric, a unit with a quantitative value was assigned to every metric. TABLE II depicts the relationship between the needs statements, metrics and units.

TABLE II: METRICS AND CUSTOMER NEEDS CORRELATION

#	Affected Needs	Metrics	Units	Importance
1	1	Total mass	kg	1
2	2	Production cost/unit	CAD $	5
3	3	NA & EU standards	Pass/Fail	5
4	4, 9	Time to assemble/disassemble for maintenance	s	3
5	5	Deflection/deformation	mm	5
6	5	FEA analysis	-	5
7	5	Critical load	N	5
8	5	Impact testing	Pass/Fail	5
9	6	All hitch options	list	4
10	6	Previous model backward compatibility	Pass/Fail	4
11	7	# of usage per/year	#	3
12	8	Time to disassemble and store	s	1
13	9	Tools required for maintenance	list	3
14	10	Clearance from boom arm	mm	5
15	10	Clearance from decal location	mm	5
16	11	All weather conditions	list	3
17	11	All ground conditions	list	3
18	12	Force required to operate (i.e. turn the handle bar, etc.)	N	4
19	12	# of operator required to install	#	4
20	12	Time required to setup and install stand	s	4

Having the target specifications defined, the team was able to assign qualitative values, marginal and ideal, that correspond to the given units. The marginal values refer to the acceptable range that the design must be within, while the ideal values represent the “best” case scenario for the design. The marginal and ideal values were generated internally through team discussions and approved externally by the client. TABLE III summarizes the target specifications and their marginal and ideal values.

TABLE III: TARGET SPECIFICATIONS WITH MARGINAL AND IDEAL VALUES

#	Affected Needs	Metrics	Units	Marginal Values	Ideal Values
1	1	Total mass	kg	≤9.07	<6.80
2	2	Production cost/unit	CAD $	100	<100
3	3	NA & EU standards	Pass/Fail	Pass	Pass
4	4, 9	Time to assemble/disassemble for maintenance	s	<60	<30
5	5	Deflection/deformation	mm	≤5	≤2
6	5	FEA analysis	-	-	-
7	5	Critical load	N	≥6672	>16700
8	5	Impact testing	Pass/Fail	Pass	Pass
9	6	All hitch options	list	Pass	Pass
10	6	Previous model backward compatibility	Pass/Fail	Pass	Pass
11	7	# of usage per/year	#	365	730
12	8	Time to disassemble and store	s	<60	<30
13	9	Tools required for maintenance	list	Pass	Pass
14	10	Clearance from driveline	cm	6-8	>8
15	10	Clearance from decal location	mm	10-14	>14
16	11	All weather conditions	list	Pass	Pass
17	11	All ground conditions	list	Pass	Pass
18	12	Force required to operate (i.e. turn the handle bar, etc.)	N	≤225	180
19	12	# of operator required to install	#	1-2	1
20	12	Time required to setup and install stand	s	<60	<30

1.6 Constraints and Limitations

In order to gain a better understanding of the expectations for the project, constraints and limitations needed to be determined. Through consultation with the client, our design team was able to create six major branches of limitations which included time, cost, weight, size, engineering resources and safety. TABLE IV below summarizes the constraints and limitations that will dictate many factors and considerations throughout the course of this project.

TABLE IV: DETAILED DESCRIPTION OF PROJECT CONSTRAINTS AND LIMITATIONS

No.	Constraints	Detailed Description
1	Time	Final report and presentation of the final concept for the client is due on December 8, 2016 due to the short span of the course.
2	Cost	Final design cost of $50 CAN/unit.
3	Weight	Weight is limited to 20lbs or less. This includes all the hardware used to assemble the stand, mounting adapters (if any), and additional lines or fluid used for externally powered system (if any).
4	Size	No specific number is specified. Any size is acceptable as long as the stand does not interfere with driveline components (decals included) and crops; and the weight requirement is met.
5	Engineering Resources	Limited to U of M software installed in the computer labs and resources (textbooks and literatures) in the Engineering library, online resources, and information from our clients.
6	Safety	The final concept is to comply with EU Machinery Directive 2006/42/EU. It must be able to support 1500lb load while maintaining stability in all conditions.

2.0 External Research Results

In order to help generate a higher quality of conceptual designs, applicable research was performed in the following four major categories:

- Commercially Available Trailer Jacks
- Competitors' Offerings
- Relevant Patents
- Relevant Standards

Each of these categories are discussed in this section of the report. A thorough understanding of how these categories specifically apply to our design will aid in helping our team produce an overall better final conceptual design. As well, this research will ensure that the design does not infringe upon any existing designs, nor fail to comply with technical standards.

2.1 Commercially Available Trailer Jacks

Due to the fact that MacDon does not manufacture trailer jacks, the team decided to perform research into different trailer jack manufacturers. By performing this research, the team was able to determine what kind of mounting options and jack styles are commonly available to purchase. The following list contains all trailer jack manufacturers that were found when searching Google for the terms "trailer jack", "trailer jack manufacturer" and "trailer jack for sale".

- Bulldog*
- Fulton*
- Pro Series*
- Reese
- Braber Equipment
- McMaster-Carr
- Curt Manufacturing
- Ultra-Tow
- Haul-Master
- Husky Towing Equipment
- Ram (Midwest Marketing Solutions)
- Vestil Manufacturing

- Simol
- Butler Products
- Ironworks Industries

* indicates companies owned by Cequent Group

All of these jack manufacturers produce jacks that have weld on, pin and tube mount and a-frame style jack mounts. Additionally, all of these manufacturers produce trailer jacks with a rated capacity of 5000 lbs, which is the same as the jack currently used by MacDon. Most manufacturers offer trailer jacks with a capacity of 8000 lbs, while other manufacturers offer trailer jacks with a capacity 7000 lbs. Most manufacturers offer trailer jacks in the 5000 lbs class with both round and square cross sections. While supporting side loads, the square cross section jacks are up to 70% stronger than jacks with round cross sections.

2.2 Competitor Offerings and Products

The team had earlier identified research into competitors' products as necessary to successfully redesign the jack stand. By looking at competitors' designs, the team will gain insight into what the competitors' design featured. With the knowledge of the competitors' designs, some of their design features may be incorporated into our own design generation. The products that were included in this research include New Holland's Discbine 313/316, New Holland's H7650, Vermeer's MC3700, Kuhn's FC3650 and John Deere's 946 MoCo. TABLE V shows the results of the performed research.

TABLE V: RESULTS OF COMPETITORS' OFFERINGS RESEARCH

Rotary Header	Hitch Type	Stand Type	Stand Location	Stability
New Holland Discbine 313/316 [10]	Both	Round Trailer Jack	Off Swing Tongue Guard Bar	Very Stable
New Holland H7650 [11]	2 Point	Stand	Off Rear of Swivel Gearbox	Stable
Vermeer MC3700 [12]	Drawbar	Square Trailer Jack	Side of front swivel gearbox	Stable
Kuhn FC3650 [13]	Drawbar	Square Trailer Jack	Side of front swivel gearbox	Appears stable
John Deere 946 MoCo	2 Point	Round Trailer Jack or None	Side of 2 point hitch or hitch sits on ground	Stable or Very Stable

TABLE V shows a variety of competitors' designs that are similar to MacDon's pull-type rotary header, in that the competitors mount their jacks off the lower swivel gearbox. Most competitors do not use a round tube trailer jack, but instead use a square tube trailer jack. This may be due to the extra support that square trailer jacks have in terms of side loading. This will be a consideration the team will look into as the design progresses.

2.3 Relevant Patents

Patents are technical documents that allow companies to protect their intellectual property for a specified period of time, at the cost of disclosing the design. The length of protection for design ideas depends on the nation that the patent is filed in, but is generally 20 years. Due to the information that patents hold and the economic advantage that can be gained from patents, engineers must research patent protection for designs before committing to a design choice. This section outlines the intellectual property that is related to this project and is patented through the United States Patent and Trademark Office.

2.3.1 Patent 3970278 – Jack Stand

Patent 3970278 is a patent for a stationary jack stand. The concept features four removable legs, a base consisting of two metal crossed pieces and a cap [14]. The four legs are labelled as components 12 through 15 in Figure 9. The base is labelled as components 16 and 17 and the cap is labelled as components 11 and 40-47. The four legs of the jack stand are designed to be removable from the base to require less space for storage.

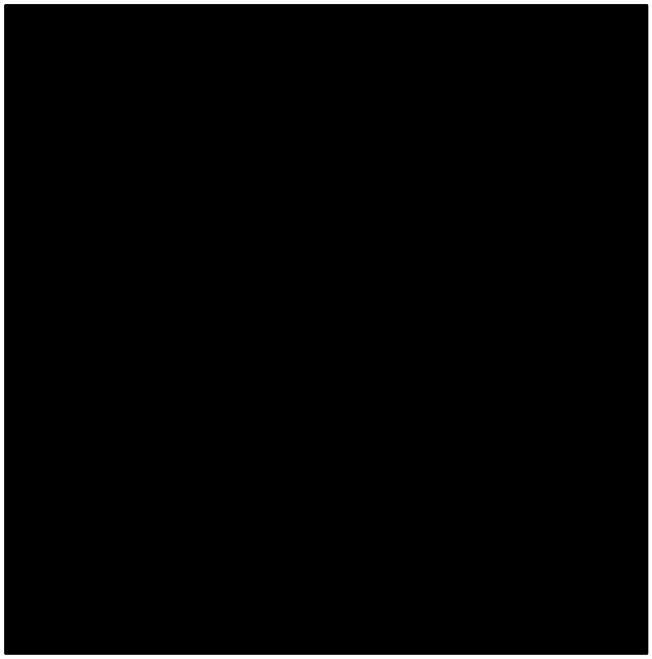

Figure 9: Trailer jack stand [14].

While this patent does not have a lifting mechanism such as a trailer jack in the design, this patent is still important to the project. This patent limits the ability of the team to design an independent stand which supports the weight of the swing tongue of the header.

2.3.2 Patent 6430908 – Self Leveling Hitch and Clevis Assembly

Patent 6430908 was filed by John Deere. This patent features a hitch mechanism for attaching a small rotary mower, such as those used for cutting grass in public parks to a tractor. The hitch mechanism does not feature a lifting device, but does include the standard tube style jack mounting system as that found on the MacDon R113/R116. The design presented in Figure 10 has two jack mounting locations. The first jack mounting location is labelled as component 90, found towards the left hand side of Figure 10. The second jack mounting point, labelled as 94, is a jack storage position that allows the jack to be stored out of the way while not in use. This jack mount is found on the right hand side of the sketch.

Figure 10: Rotary mower hitch mechanism [15].

The jack mount labelled as component 90 is attached to part of the hitch that is designed to rotate in position when the height of the hitch is adjusted [15]. This design enables the jack stand mount to remain vertical while the hitch is adjusted for different tractor hitch heights. Keeping the jack stand vertical throughout the range of adjustability of the hitch ensures that the foot plate of the jack stand will have a constant contact area with the ground. Having the foot plate of the jack in constant contact with the ground assists in the stability of the jack stand as it maximizes the potential weight distribution of the jack on all different ground types. This patent is relevant to the project as it contains a mechanism for ensuring that the jack mounting position stays vertical throughout the entire range of motion of the hitch.

2.3.3 Patent 6511279 – Automatic Fold-Up jack For an Implement Transporter

Patent 6511279 protects a spring loaded jack that moves from the lowered position to the raised position when the load is removed from the tongue of the trailer [16]. While in the downward position, the jack is locked in place via the weight of the tongue and the design of the stand. The jack can be seen in the lowered and raised position in Figure 11 and Figure 12, respectively.

[redacted]

Figure 11: Automatic leveling jack stand in the lowered position [16].

Figure 12: Automatic leveling jack in the raised position [16].

This patent is important to the project, as it features an easily and efficiently removable lifting jack. Patent 6511279 limits the team in terms of designing a spring loaded jack stand for the project.

2.3.4 Patent 6718747 – Rotary Cutter Parking Stand

This patent protects a drop down leg that fits inside the structural supports of the hitch. The main sketch of the patent can be seen in Figure 13. This stand is simple to design and manufacture and would be very easy to accidentally infringe upon in our design process [17].

[redacted]

Figure 13: Dropdown hitch stand [17].

This patent is not directly related to the project at hand, but is still important. The importance of this stand is derived from the fact that the patent allows the team to see the various types of stands MacDon's competitors' are using in the agricultural market place. The patent also offers some inspiration that could be used during concept generation.

2.4 Relevant Standards

Through research, detailed discussions with our client and help from our client's colleagues that are knowledgeable in this field of design, we were able to identify which specific sections of the applicable standards will apply directly to our design. Unfortunately, we were not able to obtain our own copy of the standards due to the high associated costs, so our client's brief summary of the applicable standards are provided and discussed in detail.

2.4.1 European Machinery Directive 2006/42/EC

The European Machinery Directive applies specifically to the stability of the design. In European Directive 2006/42/EC Clause 1.3.1 - Risk of Loss of Stability, states that "the machinery and components of that machinery must be stable enough to avoid falling over during rest, assembly, disassembly, or transport [18]." The best way to ensure compliance with this standard (through what is known as a "presumption of conformity") for our needs will be to conform to ISO Standard 4254-1:2013 as recommended by our client, although it is not explicitly referenced in the European Directive 2006/42/EC [18].

2.4.2 ISO Standard 4254-1:2013

To ensure the "presumption of conformity" is met, the best course of action would be to meet Clause 6.2 – Stability, in ISO Standard 4254-1:2013 [18]. This clause states:

> *"The design will be considered stable if it won't move by itself on firm ground with an incline of up to 8.5 degrees in any direction. Other than wheel, supports will have a maximum ground pressure of 400 [kPa]. Everything that is deployed while not in transport must be lockable in transport position. If finer details of ISO 4254-1:2013 cannot be achieved then a separate internal risk assessment can be done to ensure compliance to the European Directive 2006/42/EC (e.g. during a safety audit)* [18]*."*

Compliance with this standard is very important to the team's success, as the client expressed that non-compliance and a resulting risk assessment is highly undesirable

2.4.3 Relevant Standards Summary

All applicable standards apply directly to the stability of our design. Specifically, the design will achieve stability by avoiding falling over during rest, assembly, disassembly or transport as per European Directive 2006/42/EC Clause 1.3.1 - Risk of Loss of Stability. To achieve conformity to the European Directive, Clause 6.2 – Stability, in the ISO Standard 4254-1:2013 will be followed which requires no movement on inclines of up to 8.5 degrees, while supports (other than wheels) will have a maximum ground pressure of 400 [kPa]. If the design is deployed while not in operation, it must also be lockable for transport. Keeping all of these technical standard requirements in mind moving forward will help to produce a commercially viable design to meet our customer's needs.

3.0 Concept Generation

The purpose of the concept generation section of this project was to produce multiple viable designs with an approximate description of the technology, working principles and form of the final product. Through the use of a modified five step method, 10 conceptual designs were successfully generated that satisfy the customer's needs. These designs are summarized below in TABLE VI, with additional Information on these designs found in APPENDIX A.3.0.

TABLE VI: TOP 10 INITIAL CONEPTUAL DESIGNS

#	Name	Summary
1	Current jack and plate attachment	Current trailer jack with a plate attached to the top. When the jack is mounted to the gearbox, the plate will contact the swing tongue eliminating rotation.
2	Tripod stand	Tripod stand that utilizes current gearbox mount. The tripod design offers more points of contact in hopes of increasing stability and eliminating rotation.
3	Swing tongue plate and jack	With the thought of moving the mounting location away from the rotating gearbox. A plate would be welded to the swing tongue, were a jack can be mounted off of it.
4	Mount current jack off plate on back of gearbox	By utilizing the back of the gearbox, where the current 2-point stand is. A universal plate can be designed for both hitch styles, allowing for universal use of the trailer jack.
5	Mount jack off swing tongue	Trailer jack is mounted off of a plate on the swing tongue that uses the 4 bolt pattern.
6	Swing tongue and gearbox locking mechanics	A bracket with pin slot would be installed on to the swing tongue and gear box using the two lower and upper bolts respectively. When the header is not operational, a locking tab would be installed to eliminate rotation.
7	Folding tube jack	A compressible and extendible jack that mounts on to the gearbox. Allows for greater points of contact with the ground.
8	Current jack with corded pin and bracket	Utilizing the bracket ideas from design six, the current jack with a corded pin would be used to stabilize and eliminate rotation of the pull-type header.
9	Back of gearbox lock-in mount plate	Utilizing the lower gearbox weldment, a rotating locking plate would be installed. When the header is not operational, the plate is rotated upwards towards the swing tongue and locked in place, so that it contacts the swing tongue and eliminates rotation. A mounting location for the current jack is found on the portion of the plate facing the header.
10	Back of gearbox lock-in mount plate and bar (clamp)	Utilizing the lower gearbox weldment, a rotating plate with clamps would be installed that latches onto a bar located beneath the driveline on the swing tongue. A mounting location for the current jack id found on the portion of the plate facing the header.

This was accomplished by initially clarifying the problem and decomposing it into two main sub categories: 1) improve stability of the jack stand and 2) eliminate rotation of the gearbox. Next, an extensive internal research was completed by utilizing individual brainstorming, the "Gallery Method", SCAMPER and finally a concept combination table, which is a method of a systematic exploration. The external search included utilizing previously performed external research results on patents, technical standards and competitors' offerings to tweak our concepts before proceeding to the concept selection phase.

4.0 Concept Analysis and Selection

Concept selection is the most important phase of the design process, as it ensures that the "best" designs generated are continued with moving forward. In order to ensure that the team has selected the "best" designs available, an in depth screening and scoring process, using team developed criterion was undertaken. This section details the evaluation process performed by the team to identify the "best" conceptual designs from the generation phase.

4.1 Selection Criteria

A summary of the selection criterion used during the design selection process has been compiled in TABLE VII. This summary displays the most important and significant criteria considered for selecting the "best" conceptual designs moving forward into the detailed design phase.

TABLE VII: SUMMARY OF SELECTION CRITERIA

#	Criteria	Summary
1	Lightweight	Refers to the overall weight of the design.
2	Cost	Refers to the production and manufacturing costs of the concepts.
3	Maintenance Points Accessible	Refers to the impact that the concept would have on nearby service points.
4	Hitch Stability	Refers to the stability that the design will provide to the hitch of the pull type header.
5	Compatibility	Refers to the ability of the design to be incorporated onto the header with minimal required adjustments.
6	Durability	How the design responds to wear and how resistant it is too long term operational damage through use.
7	Easy to Store	How easy the design is to store, when the header is operational.
8	Driveline Clearance	How the design impacts/impedes the driveline when in use.
9	Easy to Use	Refers to how easy it is to mount the design and operate it.
10	Functionality	Refers to the effective function of the design.
11	Safe to Operate	Will the design be safe in all facets of its use.
12	Require Production Change	Does the design require a production change of the pull type header.
13	Overall Team Impression	Refers to the team's impression of the design using engineering knowledge and intuition gathered through education and work experience.
14	Manufacturability	Refers to how simple it is to create the final product.

4.2 Criteria Weighting

In order to determine the importance of each criterion, a weighting matrix was used. A weighting matrix evaluates each criterion relative to one another and determines their individual weights. This is done by measuring the total number of occurrences for each criterion within the matrix and dividing it by the total number of possible criteria occurrences. The criteria weighting matrix used during the design selection process is shown below in TABLE VIII.

TABLE VIII: SELECTION CRITERIA WEIGHTING

		Lightweight	Cost	Maintenance Points Accessible	Hitch Stability	Compatibility	Durability	Easy to Store	Driveline Clearance	Easy to Use	Functionality	Safe to Operate	Require Production Change	Overall Team Impression	Simple to Manufacture
Criteria		A	B	C	D	E	F	G	H	I	J	K	L	M	N
A	Lightweight		B	A	D	E	F	A	H	I	J	K	L	M	N
B	Cost			B	D	B	F	B	H	B	B	K	B	B	B
C	Maintenance Points Accessible				D	B	F	G	H	I	J	K	L	C	N
D	Hitch Stability					D	D	D	I	D	D	D	L	D	D
E	Compatibility						E	E	H	I	J	K	L	E	N
F	Durability							F	H	I	J	K	L	M	N
G	Easy to Store								H	I	J	K	L	M	N
H	Driveline Clearance									H	J	K	H	H	H
I	Easy to Use										J	K	L	I	N
J	Functionality											K	L	J	N
K	Safe to Operate												K	K	K
L	Require Production Change													L	L
M	Overall Team Impression														N
N	Simple to Manufacture														

Total Hits	2	10	1	11	4	4	1	10	7	8	12	10	3	8
Weightings	0.02	0.1	0.01	0.12	0.044	0.04	0.01	0.11	0.08	0.09	0.13	0.1	0.033	0.1

4.3 Concept Screening

Having determined the design criteria and their individual weights, each concept was evaluated using a concept screening decision matrix. The screening decision matrix utilized in this phase is known as a Pugh chart. A Pugh chart is a method of identifying the most promising design concepts amongst its alternatives. Each concept was assessed relative to a reference concept and for each determined criterion, was scored a “+”, “-“ or a “0”, relative to whether the concept in question was better than, poorer than or about the same as the reference concept. The designs that scored low, or provide only marginal improvements will be eliminated from the selection phase.

The reference concept selected for the Pugh chart was the BULLDOG tubular swivel trailer jack, currently utilized in the drawbar hitch style. Using the criterion determined in Section 4.1.15 and our student engineering intuition, the team was able to evaluate the generated concepts. A summary of the results is provided in TABLE IX.

TABLE IX: CONCEPT SCREEING RESULTS

		Design Concepts										
Selection Criteria		1	2	3	4	5	6	7	8	9	10	REF
A	Lightweight	-	-	-	0	-	0	-	0	-	-	0
B	Cost	-	-	-	-	-	-	-	-	-	-	0
C	Maintenance Points Accessible	0	-	0	-	0	0	-	0	-	-	0
D	Hitch Stability	+	+	0	+	0	+	+	+	+	+	0
E	Compatibility	0	0	-	0	-	-	0	-	0	0	0
F	Durability	+	-	-	0	-	0	-	+	+	-	0
G	Easy to Store	0	0	0	0	0	0	0	0	0	0	0
H	Driveline Clearance	+	+	+	+	+	+	+	+	+	+	0
I	Easy to Use	0	-	0	-	0	-	-	-	-	-	0
J	Functionality	0	-	-	0	-	0	-	0	0	0	0
K	Safe to Operate	0	-	0	0	0	+	-	+	0	-	0
L	Require Production Change	0	0	-	0	-	-	0	-	0	-	0
M	Overall Impression	+	-	0	0	-	-	+	0	+	-	0
N	Simple to Manufacture	-	-	-	-	-	-	-	-	-	-	0
	PLUSES	4	2	1	2	1	3	3	4	4	2	
	ZEROES	7	3	6	8	5	5	3	5	5	3	
	MINUSES	3	9	7	4	8	6	8	5	5	9	
	NET	1	-7	-6	-2	-7	-3	-5	-1	-1	-7	
	RANKING	1	8	7	4	8	5	6	2	2	8	
	REVIEW CONCEPT	YES	NO	NO	YES	NO	YES	NO	YES	YES	NO	

As shown in TABLE IX, five concepts were considered promising designs. These designs include the jack with plate attachment, jack with corded pin and bracket, back of gearbox lock-in plate and jack, back of gearbox plate and jack and the swing tongue and gearbox locking mechanism.

4.4 Concept Scoring

The final step of the concept selection process is the scoring of the concepts. Having determined the weighted selection criterion in Section 4.2, and the narrowed down designs from the screening process in Section 4.3, a weight matrix can be used to determine the "best" designs. With the above information and ranking, the expected performance of each design with an assigned score of 1-5 (1 being lowest and 5 being the best) in each criterion, a total weighted score can be generated. A summary of the results from the scoring process can be seen below in TABLE X.

TABLE X: SCORING OF INITIAL CONCEPTS THROUGH WEIGHTING MATRIX

		Design Concepts									
		Design 1		Design 8		Design 9		Design 4		Design 6	
Selection Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Lightweight	0.022	3	0.066	4	0.088	4	0.088	4	0.088	4	0.088
Cost	0.11	4	0.44	3	0.33	2	0.22	3	0.33	3	0.33
Maintenance Points Accessible	0.011	4	0.04	4	0.044	1	0.011	1	0.011	4	0.044
Hitch Stability	0.12	4	0.48	4	0.48	4	0.48	2	0.24	4	0.48
Compatibility	0.044	5	0.22	3	0.13	3	0.13	1	0.044	1	0.044
Durability	0.044	3	0.13	2	0.088	3	0.13	3	0.13	3	0.13
Easy to Store	0.011	5	0.05	5	0.055	4	0.044	5	0.055	2	0.022
Driveline Clearance	0.11	5	0.55	5	0.55	5	0.55	5	0.55	5	0.55
Easy to use	0.077	3	0.23	4	0.31	4	0.31	4	0.31	4	0.31
Functionality	0.088	2	0.18	4	0.35	4	0.35	2	0.18	4	0.35
Safe to operate	0.13	3	0.40	4	0.53	3	0.40	4	0.53	4	0.53
Require Production Change	0.11	5	0.55	1	0.11	5	0.55	5	0.55	1	0.11
Overall Team Impression	0.033	3	0.10	4	0.13	4	0.13	2	0.066	3	0.10
Simple to Manufacture	0.088	3	0.26	3	0.26	2	0.18	4	0.35	4	0.35
Total Score		3.703		3.462		3.571		3.429		3.440	
Rank		1		3		2		5		4	
Continue?		Yes		Yes		Yes		No		No	

As shown in TABLE X, the jack with plate attachment, the back of gearbox plate and jack, and jack with corded pin and bracket concepts, indicated in green, are the final designs to be considered for further analysis. To ensure that these designs are adequate for continuation on to the detailed design phase, the team sought the opinion of the client. The client reviewed the design generation and selection process, and provided the following feedback:

> "I agree with your selection of concepts to move forward with. All three designs could be feasible solutions, so I will leave it up to you to prove which one is best [18]."

With approval from the client, the team will continue with all three designs, moving forward on to the detailed design phase. It is important to note that this selection phase will be reviewed once the client provides further technical information.

4.5 Final Selected Concepts

This section will review the final designs selected by the team, via the design selection process.

4.5.1 Jack with Plate Attachment

The jack with plate attachment can be seen in Figure 14.

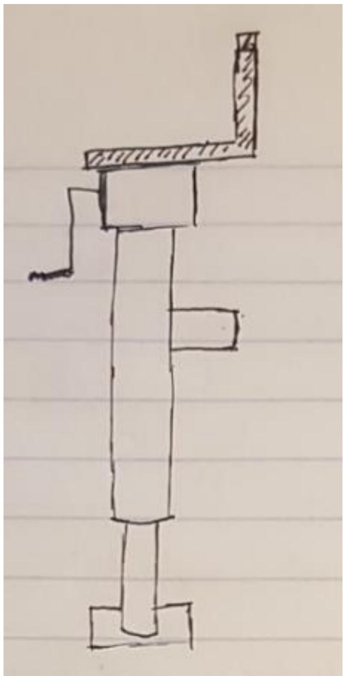

Figure 14: Jack with plate attachment.

The jack with plate attachment utilizes the current jack with a plate mounted to the top of it. When the pull-type header needs to be disconnected from the tractor, the jack is mounted to the lower gearbox and the plate makes contact with the swing tongue. The point of contact established by the plate is flush with the swing tongue, as the swing tongue and gearbox are parallel when the header and gearbox are straight. This point of contact created by the plate eliminates rotation and improves stability of the header.

4.5.2 Back of Gearbox Plate and Jack

The back of gearbox plate and jack design can be seen below in Figure 15.

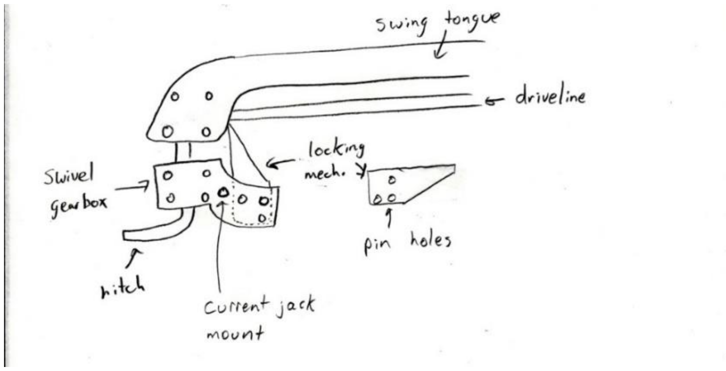

Figure 15: Back of gearbox plate and jack.

This design utilizes the already existing holes in the lower swivel gearbox weldment. On the rear of the weldment, there are three holes which are currently used for pinning the 2-point hitch stand in the appropriate location. This concept uses a flip up locking mechanism which includes the current jack mount. When in the upwards position, the locking mechanism fits between the sides of the swing tongue behind the upper swivel gearbox, thereby preventing rotation. When in the lowered or storage position, the locking mechanism sticks out behind the lower swivel gearbox.

4.5.3 Jack with Corded Pin and Bracket

The jack with corded pin and bracket can be seen in Figure 16.

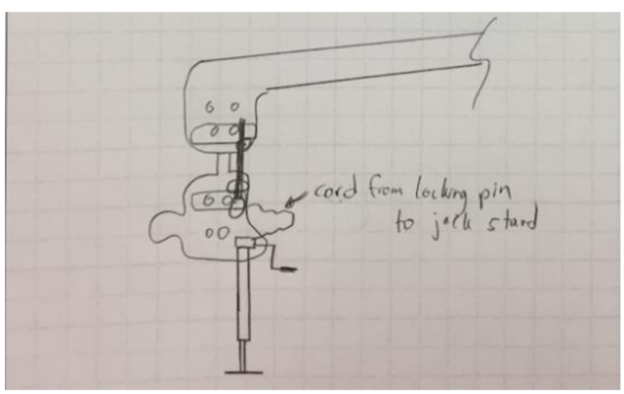

Figure 16: Jack with corded pin and bracket.

The jack with corded pin and bracket utilizes the current jack with a pin corded to the top of it. As well, this design uses the upper and bottom two bolts of the swing tongue and gearbox, respectively, to mount two brackets. When the pull-type header needs to be disconnected from the tractor, the jack is mounted to the lower gearbox and the corded pin is inserted into the brackets (when the header and gearbox are straight). The pin and brackets eliminate the rotation of the header improving its stability.

4.6 Summary of Initial Design Selection Process

In summary, ten of the team's most promising concepts were selected from the generation phase and subjected to a rigorous screening and scoring process. This process was performed in order to determine the teams "best" designs to be carried on, into the detailed design phase of the project. The screening and scoring process can be broken into three stages: determining the selection criterion and their weights, screening the concepts and lastly scoring the concepts.

The first stage of the concept selection involved the determination of the selection criteria. The selection criteria was selected, based on the design constraints and target specifications outlined by the client and design team respectively. From here, the 12 selection criteria was weighed against each other using a weighting matrix and their individual weights were determined.

With the selection criteria weighted, the team moved onto the second stage, which involved the screening of the ten designs. The screening was performed using a Pugh chart, where the designs were individually evaluated against a reference design; being the current trailer jack for the drawbar hitch style header. From the screening matrix, the team was able to narrow down the design selection to five concepts, which included the jack with plate attachment, jack with corded pin and bracket, back of gearbox lock-in plate and jack, back of gearbox plate and jack, and the swing tongue and gearbox locking mechanism.

Finally, with the five concepts and the weighted selection criteria, the team was able to determine the "best" designs. This was accomplished through the use of a weighting matrix, where the concepts were scored a 1-5 (with 1 being the worst and 5 being the best) in each criterion. The results of the design selection phase yielded the "best" designs being the jack with plate attachment, back of gearbox plate and jack, and lastly the jack with corded pin and bracket.

5.0 Conceptual Design Revisited

With the completion of the concept selection phase and submittal of our report, the design team determined that it was necessary to meet with the client and revisit the pull-type rotary header. This was done, to ensure that the top three designs met the required needs of the client before the final concept selection and continuation onto the detailed design phase of the project.

After consultation with the client and revisiting the rotary header, the design team identified potential issues with the top three designs. The first design (jack with plate attachment) had issues with alignment, as the mounting location for the jack on the gearbox did not allow for the plate to align with the swing tongue as previously thought (based on initial pictures). Another issue with this design was that the team had no solution for attaching the plate to the jack. The second design (jack with corded pin and bracket) created issues, as it would require bolts to be removed from the swing tongue and gearbox. This process would be noted as a production change and verification would be required from the clients design staff. Lastly, the third design (back of gearbox plate with current jack) did not meet its functional requirement, as it could not lock rotation. This was due to limited space between the back of gearbox and swing tongue, which was not apparent in the provided SolidWorks model. This meant that the design team was required to revisit the current designs, concept generation and selection phase of the design process.

5.1 Design Generation Part 2

With the thought of limiting disassembly and rotation of the header, the team looked to external and internal research to generate new concepts. A large part of this researching process involved looking for alternative mounting locations. Through revisiting the external research of our competitors' designs, specifically New Holland's Discbine (further information

Appendix A.1.0), the team considered using the driveline guards of the pull-type header. The driveline guards are two circular bars that extend down from each side of the swing tongue, protecting the driveline from incidental contact with the tires of the tractor when the operator makes sharp corners. The guards can be seen below in Figure 17, highlighted in blue.

Figure 17: Driveline guards.

With the use of internal research, via brainstorming, the team was able to generate two new concepts, Figure 18 and Figure 19 shown below.

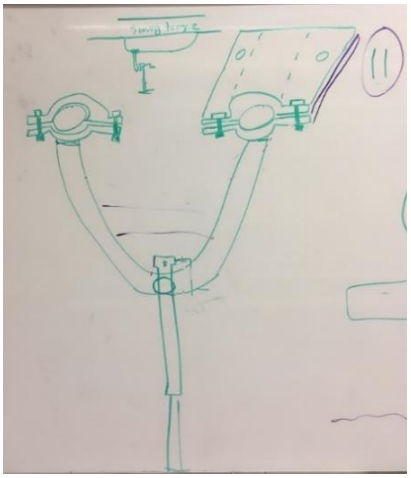

Figure 18: Concept 11.

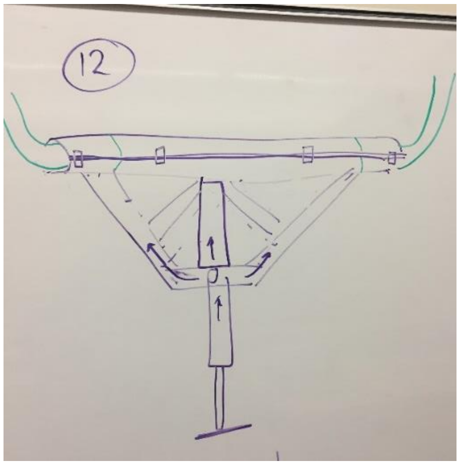

Figure 19: Concept 12.

Concepts 11 and 12 will be discussed in further detail in the following subsections.

5.1.1 Double Guard Bar V-Support

The first concept generated after the client meeting and internal/external research was concept 11, the double guard bar V-support. The driving idea behind this design, was that if the team can't limit rotation of the gearbox at the gearbox, we will do so by moving to a different mounting location. The v design of the support helps lessen the stresses seen in the structure, as it is distributes the load among two members allowing for a smaller cross-sectional area. An image of this concept can be seen in Figure 20 and Figure 21 below.

Figure 20: Double guard bar V-support side view.

Figure 21: Double guard bad V-support side view.

The inspiration for this design was the New Hollands Discbine. Similar to the Discbine, the V-support utilizes both guard bars to mount off of, with the use of sheet metal brackets. Welded to the brackets is v-tubed structure that will extend downwards to the same height of the jack mounting location on the gearbox. Here, there will be a similar mount, welded on to the tubes to accommodate the jack. The mounting location for the jack will go no lower, than that on the gearbox, to ensure sufficient coverage when the pull-type rotary header is operational. When the pull-type header needs to be disconnected from the tractor, the jack is mounted to mounting location on the v-tube via a pin connection and cranked until the header releases from the tractor. At this location, the team can eliminate rotation and improves stability of the header.

5.1.2 Single Guard Bar Truss Support

The second concept generated was the single guard bar truss support, which is similar to the first concept, but it only uses one guard bar. In doing so, an eccentric loading situation is created. To ensure that the load is effectively supported, a truss design is considered as they are inherently strong. An image of this concept can be seen in Figure 22 Figure 23 below.

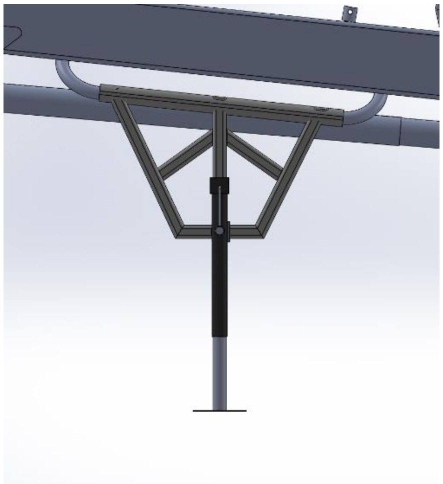

Figure 22: Single guard bar truss support side view

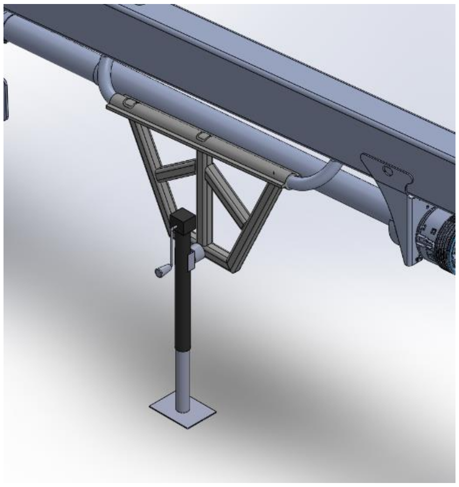

Figure 23: Single guard bar truss support side view.

The truss support will use either one long sheet metal bracket or 2 sheet metal brackets to mount onto the guard bar. Welded to the bracket(s) is a truss structure that will extend downwards to the same height of the jack mounting location on the gearbox. Similar to the V-support, the mounting location for the jack will extend down to the gearbox and the jack will be mounted to mounting location on the truss structure via a pin connection.

5.2 Concept Selection Part 2

With two new concepts created that the team feels better suits the client's needs, the design selection phase was performed for a second time. Having determined the weighted selection criterion in Section 4.2, and adding the two new designs to the final three designs from Section 4.5, a weight matrix can be used to determine the "best" design. With the above information and ranking, the expected performance of each design with an assigned score of 1-

5 (1 being lowest and 5 being the best) in each criterion, a total weighted score can be generated. A summary of the results from the scoring process can be seen below in TABLE XI.

TABLE XI: FINAL SCORING OF CONCEPTS THROUGH WEIGHTING MATRIX

		Design Concepts									
		Design 1		Design 8		Design 9		Design 11		Design 12	
Selection Criteria	Weight	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Lightweight	0.022	3	0.066	4	0.088	4	0.088	3	0.066	2	0.044
Cost	0.11	4	0.44	3	0.33	2	0.22	2	0.22	1	0.11
Maintenance Points Accessible	0.011	4	0.04	4	0.044	1	0.011	5	0.055	5	0.055
Hitch Stability	0.12	4	0.48	4	0.48	4	0.48	4	0.48	4	0.48
Compatibility	0.044	5	0.22	3	0.13	3	0.13	5	0.220	5	0.220
Durability	0.044	3	0.13	2	0.088	3	0.13	4	0.18	4	0.18
Easy to Store	0.011	5	0.05	5	0.055	4	0.044	5	0.055	5	0.055
Driveline Clearance	0.11	5	0.55	5	0.55	5	0.55	5	0.55	5	0.55
Easy to use	0.077	3	0.23	4	0.31	4	0.31	4	0.31	4	0.31
Functionality	0.088	2	0.18	4	0.35	4	0.35	4	0.35	4	0.35
Safe to operate	0.13	3	0.40	4	0.53	3	0.40	4	0.53	4	0.53
Require Production Change	0.11	5	0.55	1	0.11	5	0.55	5	0.55	5	0.55
Overall Team Impression	0.033	3	0.10	4	0.13	4	0.13	5	0.165	3	0.10
Simple to Manufacture	0.088	3	0.26	3	0.26	2	0.18	3	0.26	3	0.26
Total Score		3.703		3.462		3.571		3.989		3.791	
Rank		3		5		4		1		2	
Continue?		No		No		No		Yes		No	

From TABLE XI, the double guard bar V-support (design 11), indicated in green, is considered to be the best design out of all the concepts considered and will be continued on with into the detailed design phase.

6.0 Design Evolution

After having a final design chosen and agreed upon by our team and client, it was now time to further refine the design. This refinement process focussed on the specific details of the design, such as how it would be mounted to the swing tongue and designing for manufacturability. Additionally, the process involved the inclusion of hand calculations, as to ensure that the design was capable of supporting the weight of the hitch. Our client was also consulted several times during this process to ensure our design would fully meet their needs and expectations. Through this design refinement process and incorporating client feedback, our final design followed a systematic evolution into an end product.

6.1 V-support Version 1

The first version of the V-support was defined in Section 5.1.1 of the report. In summary, the V-support structure will mount to the driveline guards via brackets and extend downwards to the previous mounting location height, where the previously used jack can be applied to lift and stabilize the header. The brackets mounted on the guard bars would be welded to the top of the V-support tubing as seen below in Figure 24 and Figure 25. The top of the tubing on each member of the V-support would be notched with the 2.10 [in] diameter profile of the bracket on a 6˚ angle to have a seamless fit with the bracket. This 6˚ angle coincides with the angle of the swing tongue driveline guard bars relative to the horizontal at average operational height which would allow the V-support to remain perfectly vertical when mounted on the driveline guards.

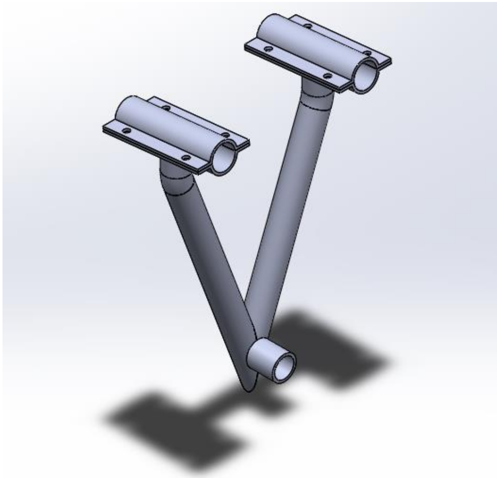

Figure 24: V-support version 1 rear iso view.

Figure 25: V-support version 1 side view.

The round tubing directly below the brackets would be perfectly vertical for 3 [in] before it was bent in order to come together at the base of the V-support. The reason for the 3 [in] vertical before the bend was this would allow more of a vertical force to be applied to the mounting bracket clamps to help prevent rotation of the clamps about the driveline guard bars. The bottom of the V-support would then extend below the height of the existing jack stand mount on the gearbox approximately 2 [in]. This would provide an adequate width on the V-supports cross section at the height of the existing jack stand mount to weld the cylindrical mount onto the V-support. Keeping the same mounting height as the previous location on the gearbox, allows for the client to use of the original jack stand.

The major issue initially identified with this design version by our team, was the fact that the cylindrical mount did not have sufficient contact with the circular tubing as seen below in Figure 26.

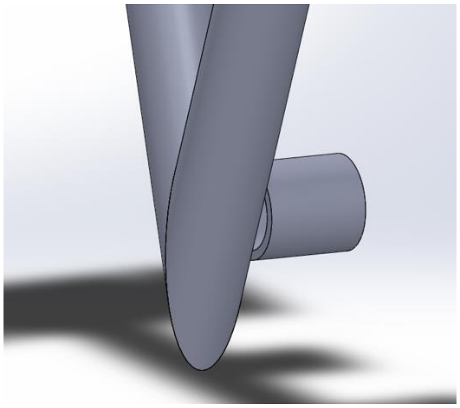

Figure 26: V-support version 1 weld gap.

A gap of approximately 1/2 [in] in width was present on both sides of the cylinder where it meets the round tubing. This gap was determined to be too large to be filled in with welding which would mean large portions of the cylindrical mount could not be welded to the round tubing of the V-support. This was a huge concern to our group since all the weight of the header was transferred from the cylindrical mount to the V-support at this location. Resultantly, our group decided a full weld bead must be achieved at the contact of the cylindrical mount to the V-support tubing.

6.2 V-support Version 2

The second version of the V-support featured the addition of a triangular mounting plate as seen in Figure 27 and Figure 28.

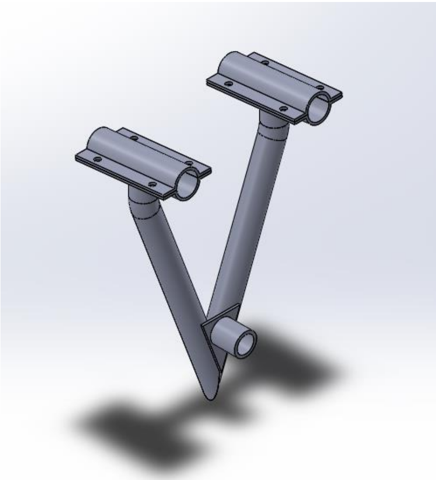

Figure 27: V-support version 2 rear iso view.

Figure 28: V-support version 2 side view.

The addition of this triangular mounting plate would allow a flat surface for the cylindrical mount to be entirely welded to around its circumference. Next the mounting plate could be designed geometrically to have its two angled edges align perfectly with the round tubing's centerline, allowing for welds of sufficient length to attach the triangular plate to the tubing. This V-support version 2 design was now complete, and presented to our client for their feedback.

6.3 V-support Version 3

Upon meeting with our client and presenting version 2 of the V-support, a major issue was identified which required attention. The mounting bracket clamps were found to be an issue as they extended horizontally 1.2 [in] outward from the driveline guards as seen in Figure 29.

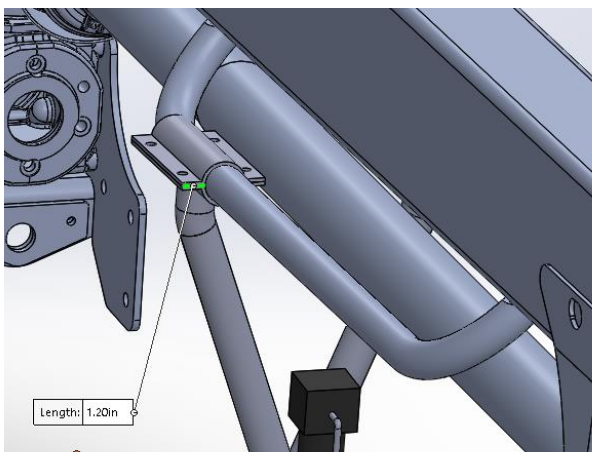

Figure 29: Horizontal mounting bracket problem.

The issue of this outward extension of the mounting bracket was a concern to our client because the driveline guard must also act as a tire protector. When the pull type header is in operation and if the operator of the tractor jack knives the header while turning, the driveline guards will prevent the tractor tires from hitting the driveline as seen in Figure 30 and Figure 31.

[redacted]

Figure 30: Driveline guard in use [19].

Figure 31: Zoomed view of driveline guard in use [20].

If the tires were to contact the mounting brackets of our current design, this could cause failure of our design. For this reason, having the mounting brackets extend outwards from the driveline guards was deemed unacceptable, and re-designing would be required. The client had suggested changing the design so it would mount on the inside of the driveline bars.

6.4 V-support Version 4

The goal for version 4 of the V-support design was to meet the client requirements by moving the design to the inside of the driveline guard bars. As a team we each individually brainstormed ideas to achieve this. Then collectively we went through possible ideas and realized that in an attempt to keep the budget low we would have to stick with the sheet metal mounting bracket clamps as other ideas required machined and cast parts. However, by rotating the brackets 90˚ we could eliminate the outwards extension which can be seen in Figure 32.

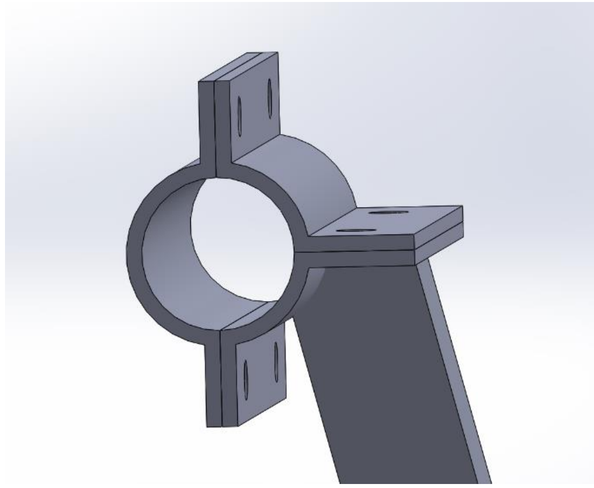

Figure 32: V-support version 4 mounting brackets.

With the Brackets rotated 90˚, the tubing had the original 3 [in] bend removed and was shaped as a “perfect V” to allow the tubing to come into the inside surface of the bracket as seen in Figure 32. However, in order to have the tubing mount on the inside of the bracket this required a reduction in tubing size. Originally we had planned to use round 1.5 [in] NPS SCH 80 piping to match the piping used for the driveline guards. 1.5 [in] NPS pipe coincided to an outer diameter of 1.90 [in] which was too large as much of the tubing would extend inwards from the inside edge of the clamp. It was therefore decided that a reduction in tubing outer diameter would be needed. 1.5 [in] outer diameter round tubing was chosen instead (square tubing seen in figure will be explained later). This still hung off the clamp’s outer edge quite substantially so the choice was made to split the inside half of the clamp into two equal pieces as seen in Figure 32. The bracket flange that joined the two quarter clamps would also multipurpose as a horizontal surface to support the tubing that extended off the inside edge of the clamps cylindrical face. The tubing could then be welded to the clamps cylindrical face and the bottom of the clamps flange face.

The choice to switch the tubing geometry from circular to square was decided mostly for ground clearance issues. With the tubing now coming from the interior of the mounting clamps (as opposed to directly from the centerline) this narrowed the distance across the top of the “V” shape. This narrowed top dimension resulted in a sharper angle of incidence between the tubes at the base of the “V”, and left more tubing extending down past the mount decreasing ground clearance. The round tubing extended 6.19 [in] past the mounts center, where as the square tubing extended only 4.22 [in] past the mounts center as seen in Figure 33 and Figure 34 respectively.

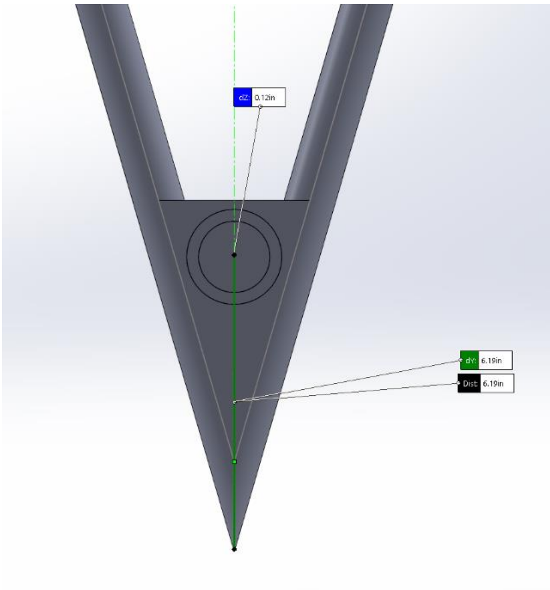

Figure 33: Round tubing ground clearance.

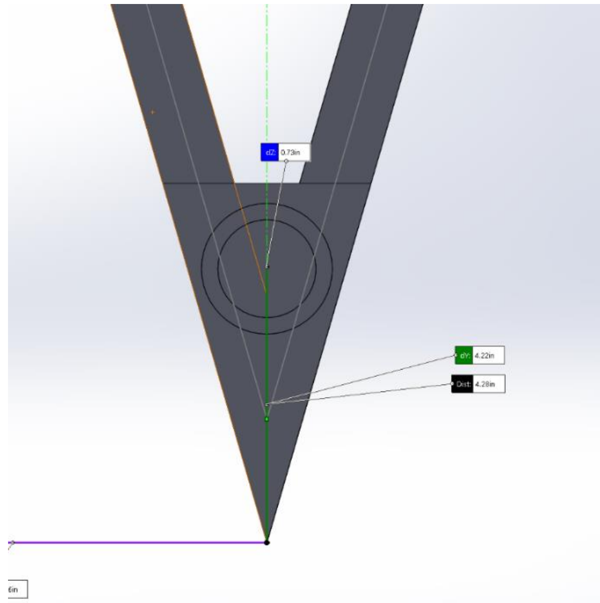

Figure 34: Square tubing ground clearance.

For increased ground clearance the square tubing was chosen. This is a result of the fact that the plate can now be welded directly to the square tubing’s outer edges, as opposed to the round tubing’s centerline. The final images of version 4 can be seen in Figure 35 and Figure 36, noting that the second mounting bracket was omitted to show the required tube notching profile.

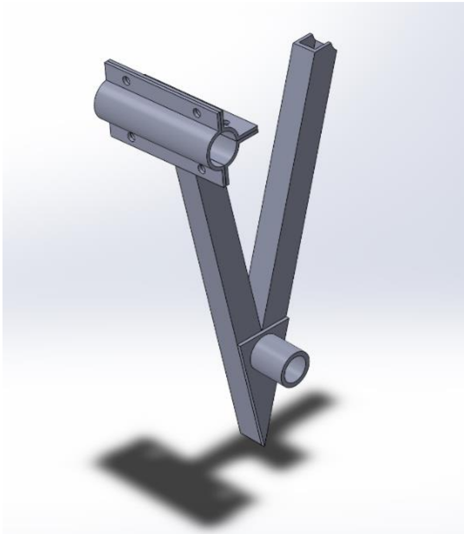

Figure 35: V-support version 4 rear iso view.

Figure 36: V-support version 4 side view.

The issues our team still had with version 4 of the design were mostly due to the overall weight of the design. We felt there was excess material in several parts of the design that was not required. This would need to be addressed in the next version of the design. At this stage of the detailed design process, the team determined that it was crucial to begin analytical calculations on the driveline guards and the design itself. Analytical calculations involved structural analysis, material selection and fatigue analysis where applicable. This was done to ensure that the guards and structure are capable of supporting the weight of the hitch, before the team progressed any further.

6.4.1 Driveline Guard Calculations

To perform the structural analysis on the driveline guards, the material of the guards had to be determined. Our client informed the team that the driveline guards are manufactured from ATSM A53, Grade B, Type E. Grade B is a designation that represents the structural strength of the pipe, while Type E designates the pipe manufacturing process, which was electrical resistance welding. TABLE XII shows the published material properties of this structural pipe [21]. The shear strength of the material was calculated using the Tresca failure criteria, which says that the shear strength of a material is half of the materials yield strength.

TABLE XII: ASTM A53 TYPE B PIPE MATERIAL PROPERTIES

A53 Type B Pipe Material Properties	
Tensile Strength	415 MPa
Yield Strength	240 MPa
Shear Strength	207.5 MPa

Our client also informed the team that the design for the driveline guards calls for a 1.5 [in] nominal pipe size of Schedule 80 pipe. TABLE XIII lists the outer and inner diameters and the wall thickness of this pipe [21].

TABLE XIII: ASTM A53 TYPE B PIPE 1.5 INCH NOMIAL PIPE SIZE DIMENSIONS [21]

Dimension	Value, mm (in)
Outer Diameter	48.26 (1.9)
Inner Diameter	38.1 (1.5)
Wall Thickness	5.08 (0.2)

With the dimension of the driveline guard known, the cross-sectional area of the driveline guard and the area moment of inertia were calculated. To calculate the cross-sectional area of the pipe, the inner and outer diameter of the pipe where changed into the inner and outer radius, as seen in Figure 37.

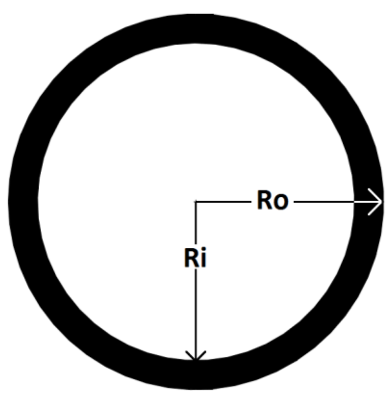

Figure 37: Cross-sectional view of the driveline guard, indicating the inner and outer radius of the guard.

The cross-sectional area was calculated as shown.

$$A = \pi * ({r_o}^2 - {r_i}^2)$$

$$A = \pi * (0.02413^2 - 0.01905^2)$$

$$A = 0.000689\ m^2$$

The area moment of inertia was calculated as shown.

$$I = \frac{\pi * ({d_o}^4 - {d_i}^4)}{64}$$

$$I = \frac{\pi * (0.04826^4 - 0.0381^4)}{64}$$

$$I = 1.63 \times 10^{-7}\ m^4$$

Once the material properties, dimensions and material area effects of the pipe were calculated, a stress analysis on the pipe was performed. The stress analysis was broken down into two separate parts, the straight portion of the driveline guards and the mounting between the driveline guards and the swing tongue. This separation of the two sections was done as the driveline guards are manufactured with a large bend in them at each end. This large bend cannot be analyzed using analytical methods and must be analyzed using finite element analysis with further verification from field testing.

6.4.2 Beam Analysis

To perform the stress analysis on the straight portion of the driveline guards, it was assumed that this section was a beam fixed at both ends with a distributed load applied to it. The distributed load accounts for the mounting location of the v-support structure. In this analysis, the structure was clamped to the guards via a bolted bracket. An image of this loading situation is shown in Figure 38.

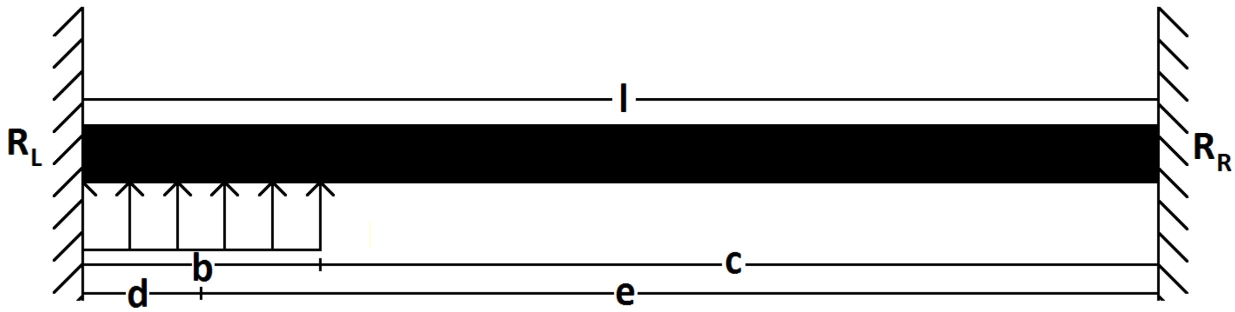

Figure 38: Driveline guard loading simplification as a distributed load on a beam fixed at both ends.

The dimensions for this loading situation are shown in TABLE XII.

TABLE XIV: DIMENSIONS OF BEAM AND LOADING

Distance Symbol	Distance (m)
b	0.203
c	0.718
d	0.1015
e	0.8195
l	0.921

The team was informed by the client that the weight on the hitch was 1500 lbs and the recommended factor of safety was 2.5. For the analysis, the weight of the hitch was split in half, as it was assumed that the weight of the hitch would be evenly distributed through both driveline guards. The weight of the hitch was converted into metric units, thereby creating an applied distributed load of 8337.5 N on the driveline guard. Once the applied load was known, the shear forces were calculated at each end. The shear forces at R_L were calculated as shown.

$$R_L = \left(\frac{wb}{4l^3}\right)[4e^2 * (l + 2d) - bc]$$

$$R_L = \left(\frac{-8337.5 * 0.203}{4 * 0.921^3}\right)[4 * 0.8195^2 * (0.921 + 2 * 0.1015) - 0.203 * 0.718]$$

$$R_L = -1619.35\ N$$

The shear forces at R_R were calculated as shown below.

$$R_R = wb - R_L$$

$$R_R = -8337.5 * 0.718 - 1619.35$$

$$R_R = -73.16\ N$$

From the calculated shear forces, the shear stress at both locations can be calculated.

$$\sigma = \frac{R}{A}$$

$$\sigma = \frac{-1619.35}{0.000689}$$

$$\sigma = -2.34\ MPa$$

Using the same formula, the shear stress at R_R were calculated to be -0.14 MPa. TABLE XV summarizes the calculated shear forces and shear stresses at each end of the driveline guards.

TABLE XV: SUMMARY OF CALCULATED SHEAR FORCES AND SHEAR STRESSES AT EACH END OF THE GUARDS

Location	Force (N)	Shear Stresses (MPa)
R_L	-1619.35	-2.34
R_R	-73.16	-0.11

As both of the calculated shear stresses are less than the maximum allowable shear stress (207.5 MPa), the driveline guards will not fail due to shear stresses caused by the weight of the swing tongue.

Next, the maximum allowable bending moment on the driveline guards was calculate. This was done so that the team could easily compare the calculated bending moment to an allowable limit. The maximum allowable bending moment was calculated as follows.

$$M_{max} = \frac{\sigma I}{y}$$

$$M_{max} = \frac{207500000 * 1.63 \times 10^{-7}}{0.0241}$$

$$M_{max} = 1400\ Nm$$

With the allowable bending moment in the driveline guards calculated, the moment at each end of the driveline guard caused by the loading was calculated. This was done to ensure that the shear stresses caused by bending would not cause failure in the pipe. The moment acting at R_L was calculated as shown.

$$M_L = \frac{wb}{24l^2}[b^2(l + 3c) - 24e^2d]$$

$$M_L = \frac{-8337.5 * 0.203}{24 * 0.921^2}[0.203^2 * (0.921 + 3 * 0.718) - 24 * 0.8195^2 * 0.1015]$$

$$M_L = 125.48\ Nm$$

The moment acting at R_R was calculated as shown.

$$M_R = R_L l - wbe + M_L$$

$$M_R = -1619.35 * 0.921 - 8337.5 * 0.203 * 0.8195 + 125.48$$

$$M_R = 21.07\ Nm$$

The shear stresses caused by the bending moments in the driveline guards are calculated as shown below.

$$\sigma_L = \frac{-My}{I}$$

$$\sigma_L = -\frac{-125.48 * 0.02413}{1.63 \times 10^{-7}}$$

$$\sigma_L = 18.6\ MPa$$

Using the same formula, the stresses caused by the bending moment at the right hand end of the support were 3.12 MPa.

Once the moments were calculated, the maximum moment between each support was calculated. The calculation for the maximum moment between the supports was performed as follows.

$$M_{max} = M_L + \frac{{R_L}^2}{2w}$$

$$M_{max} = 21.07 + \frac{-1619.35^2}{2 * 8337.5}$$

$$M_{max} = -31.78\ Nm$$

The stress caused by this bending moment was calculated in the same procedure as performed for the reactionary moments and was found to be -4.71 MPa. The location of this bending moment was also calculated. The location of the bending moment was calculated as follows.

$$x = \frac{R_L}{w}$$

$$x = \frac{-1619.35}{-8337.5}$$

$$x = 0.1942\ m$$

TABLE XVI summarizes the important moments and their shear stresses found throughout the straight portion of the driveline guard. The maximum allowable moment for the driveline guards was found to be 1400 Nm, which is significantly greater than the moments caused by the loading with a factor of safety of 2.5 applied to the loading.

TABLE XVI: SUMMARY OF THE MOMENTS AND RESULTANT STRESSES SEEN THROUGHOUT THE DRIVELINE GUARDS

Location	Moment (Nm)	Shear Stresses (MPa)
M_L (x=0)	125.48	18.6
M_R (x=0.921)	21.07	3.12
M_{max} (x=0.1942)	-31.78	-4.71

The von Mises' stress was used as a failure criteria for the design of driveline guards. The von Mises stress in the driveline guards were calculated using the values shown in TABLE XVII.

TABLE XVII: SUMMARY OF FORCES USED TO CALCULATE THE VON MISES STRESS IN THE GUARDS

Symbol	Stress (MPa)
Σx	18.6
Σy	0
Txy	0

The von Mises stress was calculated following the step shown below.

$$\sigma_v = \sqrt{\sigma_x{}^2 + \sigma_y{}^2 - \sigma_x\sigma_y + 3\tau_{xy}{}^2}$$

$$\sigma_v = \sqrt{18600000^2 + 0^2 - 18600000 * 0 + 3 * 0^2}$$

$$\sigma_v = 18.6\ MPa$$

Since the calculated von Mises's stress at the highest loading point is significantly below the maximum allowable shear stress in the driveline guards, this portion of the design can be considered to be safe.

6.4.3 Moment at Mounting Location

The driveline guards are welded to the underside of the swing tongue. The joint caused by welding the driveline guards to the swing tongue was considered to be stronger than the driveline guards themselves. Therefore, the only forces that needed to be considered for this section are the moment and compressive stresses in the base of the driveline guard. The driveline guard was also assumed to be meeting the bottom of the swing tongue in a non-deformed state, to simplify the cross-sectional area and allowable bending moment calculations. Additionally, it was also assumed that the weight of the hitch would be applied evenly throughout both driveline guards and that the weight would be loaded through the center of the driveline guard and the driveline guard mounting location. Due to the nature of the initial design, the loading was assumed to be completely vertical in the clamp portion of the design.

The distance between the center of the driveline guard and the center of the mounting location had to be measured. This was done so the moment created by the weight of the hitch being applied to the driveline guards could be calculated. These dimensions were measured using the measurement tool in SolidWorks 2016 on a model provided by the client. Figure 39, below shows the measurements taken on the model.

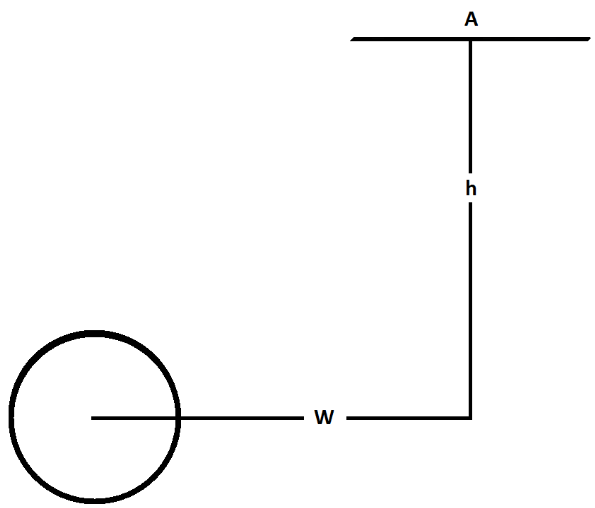

Figure 39: Distances measured from driveline guard.

In Figure 39, variable w represents the horizontal distance from the center of the driveline guard to the center of the mounting location. Variable h represents the vertical distance between the center of the driveline guard to the center of the mounting location. The measured values of these dimensions are shown in TABLE XVIII.

TABLE XVIII: DRIVELINE GUARD DISTANCES

Distance	Distance (m)
H	0.10744
W	0.10934

For the initial design, the load through the driveline guard is vertical. Therefore, the moment around the driveline guard mount is only caused by forces in the vertical direction and the horizontal distance between the center of the driveline guard and the driveline mount. The moment acting on the driveline guards was calculated as follows.

$$M = F_y * \mathrm{W}$$

$$M = 8337.5 * 0.10934$$

$$M = 911.62\ Nm$$

Since the bending moment is less than the maximum allowable stress, the driveline guard will not fail under the weight of the hitch. With the driveline guard being able to support the weight of the hitch, the team was confident moving onto analyzing the fourth version of the V-support structure.

6.4.4 V-Support Material Selection and Analysis

This section outlines the V-support tubing material selection and analytical analysis of the design. Failure of the design due to exceeding the maximum allowable shear stress, normal stress and buckling were all considered in the analysis. As well, the structure was analyzed further, with a fatigue analysis to ensure that the design would last the life expectancy of the header.

To begin the analysis, different materials and geometrical cross-sections were analyzed in order to choose an optimal cross section. Given that the rotary header driveline guard material was ASTM A53, the team decided to select a material that conformed to ASTM standards. Through research, ASTM A513 Grade B was found to be the most suitable material for this design and application, as it is a commonly used material for structural square tubing. ASTM A500 and 513 were both suitable choices but only 513 came in the smaller required cross sections (less than 2 [in]). Moreover, the material is readily available and pricing information was easily obtained.

With the material selection complete, the team analyzed different standard geometries available for A513 square tubing. The cross-sectional analysis involved looking at the geometries Factor of Safety (FOS), cost, and weight. TABLE XIX below summarizes the results from the different cross-sections. Values for the weight and cost were all obtained from Metal Depot's website [22]. For the sake of cost approximation, we used the length of the two tubes of 18.12 [in] each and an additional 6 [in] to account for manufacturing processes, which totalled 3 ft, 6 [in]. The FOS illustrates the highest FOS that could be obtained before the structure failed in at least one of the three (shear stress, normal stress and buckling) analyses performed. This factor is accurate to one decimal place as iterations were performed by changing the first decimal place in the FOS until failure was reached.

TABLE XIX: GEOMETRICAL CROSS-SECTIONAL COMPARISSONS [22]

MATERIAL	WT (LB/FT)	TOTAL WT	TOTAL COST (3' 6")	FOS
1 x 1 x .065 (16GA)	0.82	2.46	CAD 9.51	1
1 x 1 x .083 (14GA)	1.04	3.12	CAD 9.56	1
1 x 1 x .12 (11GA)	1.44	4.32	CAD 10.96	1.3
1.25 x 1.25 x .065 (16GA)	1.04	3.12	CAD 11.79	1.3
1.25 x 1.25 x .083 (14GA)	1.32	3.96	CAD 12.00	1.6
1.25 x 1.25 x .12 (11GA)	1.8	5.4	CAD 15.53	2.1
1.25 x 1.25 x .1875	2.4	7.2	CAD 42.91	2.7
1.5 x 1.5 x .065 (16GA)	1.26	3.78	CAD 14.31	1.9
1.5 x 1.5 x .083 (14GA)	1.67	5.01	CAD 13.92	2.3
1.5 x 1.5 x .12 (11GA)	2.21	6.63	CAD 17.18	3.1
1.5 x 1.5 x .1875	3.04	9.12	CAD 30.24	4.2

From the results in TABLE XIX , the team decided to pursue the 1.5 x 1.5 x 0.12 (11GA) cross-section, as it was the cheapest geometry that came closest to our required FOS of 2.5 without failure. TABLE XX summarizes typical mechanical properties of ASTM A513 used for our analysis [23] [24].

TABLE XX: TYPICAL ASTM A513 PROPERTIES [23] [24]

PROPERTIES	ASTM A513 - GRADE B	[UNIT]
Yield Strength ($\boldsymbol{\sigma_{yld}}$)	46000	psi
Ultimate Strength ($\boldsymbol{\sigma_{ult}}$)	52000	psi
Young's Modulus (E)	$2.90\text{x}10^7$	psi

6.3.3.1. Normal Stress Analysis

After the optimum cross-section was chosen, a structural analysis was performed on the design. Figure 40 below depicts the load applied and its projection on the beam-like member.

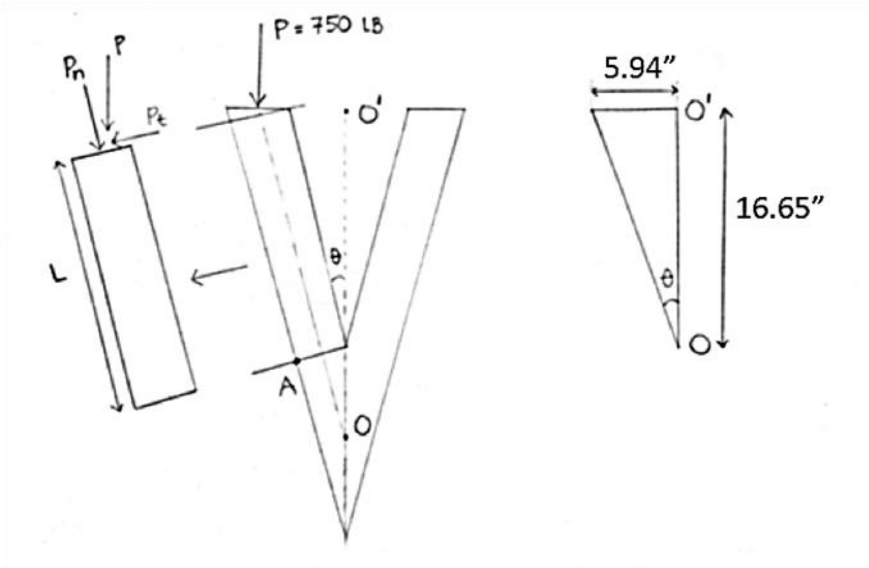

Figure 40: Structure geometry with load projection.

To simplify the analysis of the tubing, the structure was treated as a beam. Specifically, the portion defined as "L" of the beam illustrated in Figure 40 was analyzed. Force P can be projected into two forces: axial force (Pn) and translational force (Pt). The axial force is the force acting perpendicular to the cross-section of the beam, while the translational force is the force acting parallel to the cross-section of the beam. The axial force and translational force, respectively, create the normal stress and shear stress on the beam. The shear stress, will then contribute to the bending phenomena in the beam structure. Both forces can be calculated as follows:

$$P_n = P\cos\theta = 750\cos(0.34268rad) = 706.39\ lb$$

$$P_t = P\sin\theta = 750\sin(0.34268rad) = 252.01\ lb$$

$$\theta = \text{atan}\left(\frac{5.94}{16.65}\right) = 0.34268\ rad$$

where P is the loading applied vertically at the top of the beam and θ is half the angle between the two angled beams.

Taking a cross section of the tube, we can then calculate the shear force and bending moment:

$$v = -P_t = -252.01lb$$

$$M = P_t \, . x$$

Where x is the distance from the end of the beam to a certain point along the length of the beam. In other words, the bending moment is a function of beam length. As x increases, bending moment increases. The moment at the end of the beam is then:

$$M = P_t \, . L = 252.01 * 15.89 = 4004.45 \; lb.in$$

The shear force and moment distribution diagram is illustrated below in Figure 41:

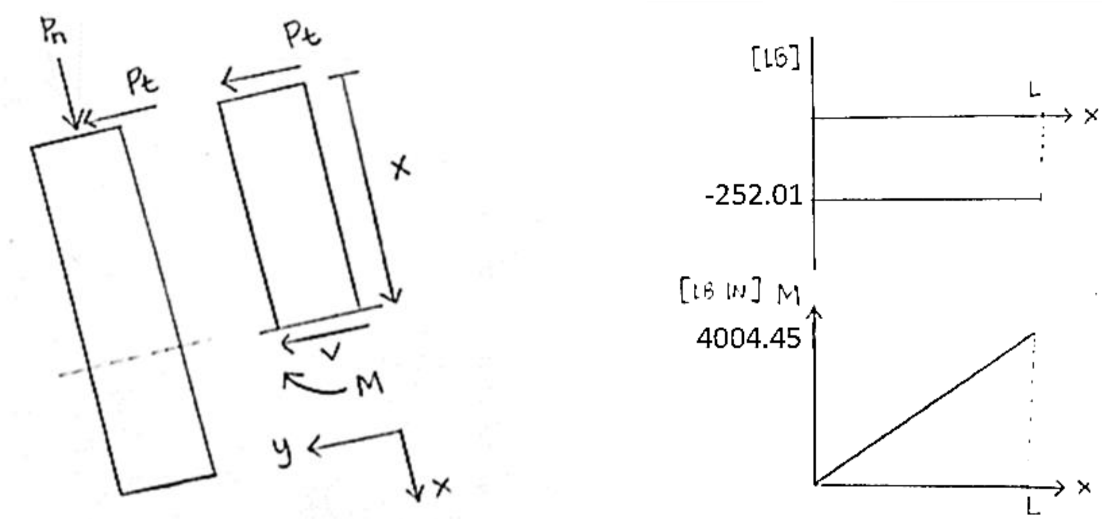

Figure 41: Beam section (left) used to illustrate distribution of shear force and moment along the length of the beam (right).

Looking at the beam structure, the largest bending moment occurs at point A of the beam (Figure 40). This can be explained through principle of superposition, which states that beam under several distributed loads can be easily computed by combining the slope and deflection caused by each of the loads. This principle is illustrated below in Figure 42.

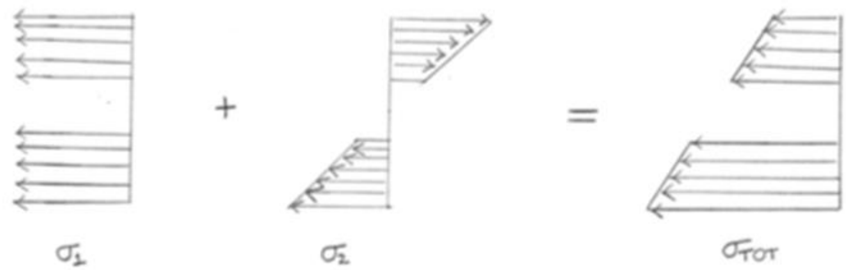

Figure 42: Principle of Superposition -stress distribution on cross-section.

The total stress acting on the beam is calculated by adding the normal stress caused by the axial force (σ_1), with the bending moment caused by the shear force or the translational force (σ_2). The normal force, translational force and subsequent total force was calculated as follows:

$$\sigma_1 = \frac{P_n}{A} = \frac{706.39}{0.6624} = 1066.41\, psi$$

$$A = {X_{out}}^2 - {X_{in}}^2 = 1.5^2 - 1.26^2 = 0.6624\, in^2$$

$$\sigma_2 = \frac{M.y}{I} = \frac{4044.45 * (\frac{1}{2} * 1.5)}{0.21184} = 14177.67\, psi$$

$$I = \frac{1}{12}(Xout^4 - Xin^4) = \frac{1}{12}(1.5^4 - 1.26^4) = 0.21184\, in^4$$

$$\sigma_{tot} = \sigma_1 + \sigma_2 = 1066.41 + 14177.67 = 15244.08\, psi$$

From the equations above, A is the cross-sectional area of the square hollow tube, Xout is the length of the outer square, Xin is the length of the inner square, y is the distance from the center of the square to the outer most point of the beam cross-section, and I is the moment of inertia of the beam.

Next, in order to see whether or not the structure is safe, we have to compare the values obtained above with the material's strength. If the results above are less than the allowable stress of the material, then the design is considered safe. The allowable stress was calculated as follows:

$$\sigma_{allow} = \frac{\sigma_{yld}}{FOS} = \frac{46000}{2.5} = 18400\,psi$$

Where FOS is the factor of safety, used for the design and obtained from the client. Based on the value above, we can conclude that the design is safe from the normal stress perspective.

6.3.3.2. Shear Stress Analysis

The equations below determine whether or not the dimensions and material would be safe under the shear stress applied on the structure. The LHS of the equation computes the allowable shear stress by dividing the maximum shear stress coming from the material by the factor of safety of the design. Shear stress on the RHS, on the other hand, depends on the dimensions of the design.

LHS: $$\tau_{allow} = \frac{\frac{\sigma_{yld}}{2}}{FOS} = \frac{\frac{46000}{2}}{2.5} = 9200\,psi$$

RHS: $$\tau = \frac{V.Q}{I.t} = \frac{252.01\,.\,0.04296}{0.21184\,.\,0.12} = 425.87\,psi$$

$$Q = \frac{1}{2}A_{out}\,.\,\frac{1}{2}\overline{y_{out}} + \frac{1}{2}A_{in}\,.\,\frac{1}{2}\overline{y_{in}}$$

$$= \left(\frac{1}{2}.\,1.5^2.\frac{1}{2}.\frac{1}{2}.\,1.5\right) + \left(\frac{1}{2}.\,1.26^2.\frac{1}{2}.\frac{1}{2}.\,1.26\right) = 0.04296\,in^3$$

t = 0.12 in

Where V is the shear stress on the beams cross-section and t is the thickness of the hollow square tube. Q is the first moment of the shaded area above the point of maximum shear with respect to the line x-x', as seen in Figure 43 below.

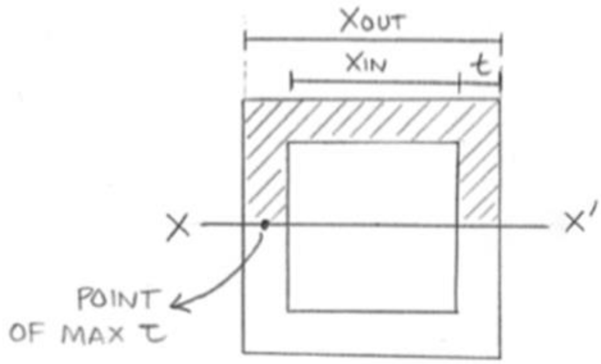

Figure 43: Cross-sectional area for calculating Q.

The design is considered safe since the LHS is bigger than the RHS.

6.3.3.3. Buckling Analysis

Aside from material failure, failures can also occur through buckling. Buckling refers to the loss of stability of a structure and generally occurs on a slender or thin-walled material under a compressive force. This phenomenon is called "Euler Buckling" where the long slender member subjected to compression moves laterally to the direction of the applied force. The force causing a buckling motion varies depending on the boundary conditions of both ends of the member. Figure 44 below depicts the four different boundary conditions, starting from the left to right: pivoted-pivoted, fixed-fixed, fixed-rounded, and fixed-free.

Figure 44: Boundary conditions for Euler buckling [25].

The critical Euler buckling force can be calculated with the following formula:

$$P_{euler} = \frac{n.\pi^2.E.I}{Le^2}$$

where n is the modifying factor for the boundary and E is the Young's modulus. E represents the stiffness of a linear elastic solid material. Le is the effective length of the column, which illustrates the length of unsupported section of the column.

For our case, considering a fixed-fixed support on the beam, we obtain an n of 4. We can then calculate the critical load that the beam can support before it buckles as follows:

LHS: $$P_{allow} = P_n\,.FOS = \ 706.39 * 2.5 = \ 1765.98\ lb$$

RHS: $$P_{cr} = \frac{n\,.\pi^2\,.E\,.I}{Le^2} = \frac{4\,\pi^2\ (29007547.546)\ 0.21184}{7.945^2} = 3843100.4\ lb$$

The above RHS equation represents the maximum load the structure can hold before buckling occurs. The LHS represents the actual loading applied on the structure. Since the LHS is much less than the RHS, it is safe to assume that the structure would not buckle due to the applied load of the hitch.

6.3.3.4. Modified Goodman Analysis for Fatigue

Any components subjected to a cyclic loading are susceptible to fatigue failure. Fatigue failure is usually not suspected as it often occurs below the yield strength and tensile strength of the material. Therefore, when analyzing whether or not the design is safe the endurance limit has to be taken into the consideration. The endurance limit represents the amount of stress that can be applied without causing fatigue failure.

For our fatigue analysis, we approximated that the structure would be used twice a day, every day for a year, for 10 years. This would give a total cycle of:

N = 2 * 365 * 10 = 7300 cycle

Considering the high amount of cycles the structure had to go through, the team felt that it was crucial to consider fatigue analysis as part of the design selection. Since ASTM A513 has a very limited amount of information readily available for public use another material with similar properties would need to be used for gathering data for the fatigue analysis. AISI 1020 low carbon steel was chosen as it is actually the parent material ASTM A513 is created from, and has a nearly identical carbon content which will be the major factor in dictating material properties. The AISI 1020 S-N curve was then utilized to determine how our structure would perform under the given cyclic loading [26].

Figure 45 below illustrates typical S-N curves for ASTM A1020 with various treatments [26]. The S-N curve illustrates the number of cycles to failure vs. the stress or repetitive load. Typical S-N curves are downward sloping. Moreover, the higher the number of cycles, the smaller the stress it takes to cause fatigue failure.

Figure 45: ASTM A513 S-N curve [26].

Since the material we picked is typically made through cold-working process, the line used in the S-N curve above would be the Fully Hardened. Extending that line to our cycles of about 7000, we approximated the stress value to be 900 MPa (130534 psi). This value represents the endurance limit of the material. Modification factors then had to be applied to the endurance strength to adapt so it realistically represents the value for our design and application. This was required since the graph only represents experimental values for specimens in pure/pristine conditions. Modification factors considered include surface finish, size, loading type, and reliability. TABLE XXI below illustrates the modification factors chosen for our design purpose.

TABLE XXI: CHOSEN MODIFICATION FACTORS

FACTORS	DESCRIPTION	VALUE
Surface Finish (ka)	Cold Drawn	0.9476
Size (kb)	Rectangular	0.8611
Loading (kc)	Axial	0.85
Reliability (kd)	99.99%	0.70248

Multiplying the endurance strength obtained from S-N curve by all 4 of the modifying factors, we got a modified endurance strength for our design:

$$S_e' = S_e.k_a.k_b.k_c.k_d = 130534\,(0.9476)(0.8611)(0.85)(0.70248)$$

$$= 63599.47\, psi$$

Similarly for the modified yield strength:

$$\sigma_{yld}' = \sigma_{yld}.k_a.k_b.k_c.k_d = 50000\,(0.90444)(0.8611)(0.85)(0.70248)$$

$$= 22412.37\, psi$$

Based on the stress amplitude and mean stress, we computed the modified Goodman line. The area below the Modified Goodmann line indicates the “safe zone” where a design is considered sufficient. In order to plot the modified Goodman line, values for the stress amplitude and mean stress had to be calculated.

$$\sigma_{max} = \frac{\sigma_{yld}}{FOS} = \frac{46000}{2.5} = 18400\, psi$$

$$\sigma_{min} = 0\, psi$$

$$\sigma_{mean} = \frac{1}{2}(\sigma_{max} + \sigma_{min}) = \frac{1}{2}(18400) = 9200\, psi$$

$$\sigma_{amp} = \frac{1}{2}(\sigma_{max} - \sigma_{min}) = \frac{1}{2}(18400) = 9200\, psi$$

For our purpose, the maximum stress was obtained by dividing the material's yield strength by the factor of safety. The minimum stress, however, represented the stress when the structure was not used which is 0 psi. Figure 46 below depicts the stress values obtained above:

Figure 46: Stress Fluctuation during cyclic loading [27].

The stress ratio (R) can then be calculated as a ratio between the minimum stress and the maximum stress.

$$R = \frac{\sigma_{min}}{\sigma_{max}} = \frac{0}{18400} = 0$$

This value indicates a case 2 loading case of zero-to-max loading, with the stress amplitude factor (A) of 1. Stress amplitude factor represents the ratio between the stress amplitude with respect to the mean stress. The different loading cases are depicted in Figure 47 below.

Figure 47: Different cyclic loading cases [27].

The modified Goodman line equation can be written as follows [27]:

$$\frac{\sigma_{amp}}{S_e{}'} + \frac{\sigma_{mean}}{\sigma_{yld}} = 1$$

The use of yield strength instead of ultimate strength is indicative that a modified Goodman is being calculated rather than normal Goodman.

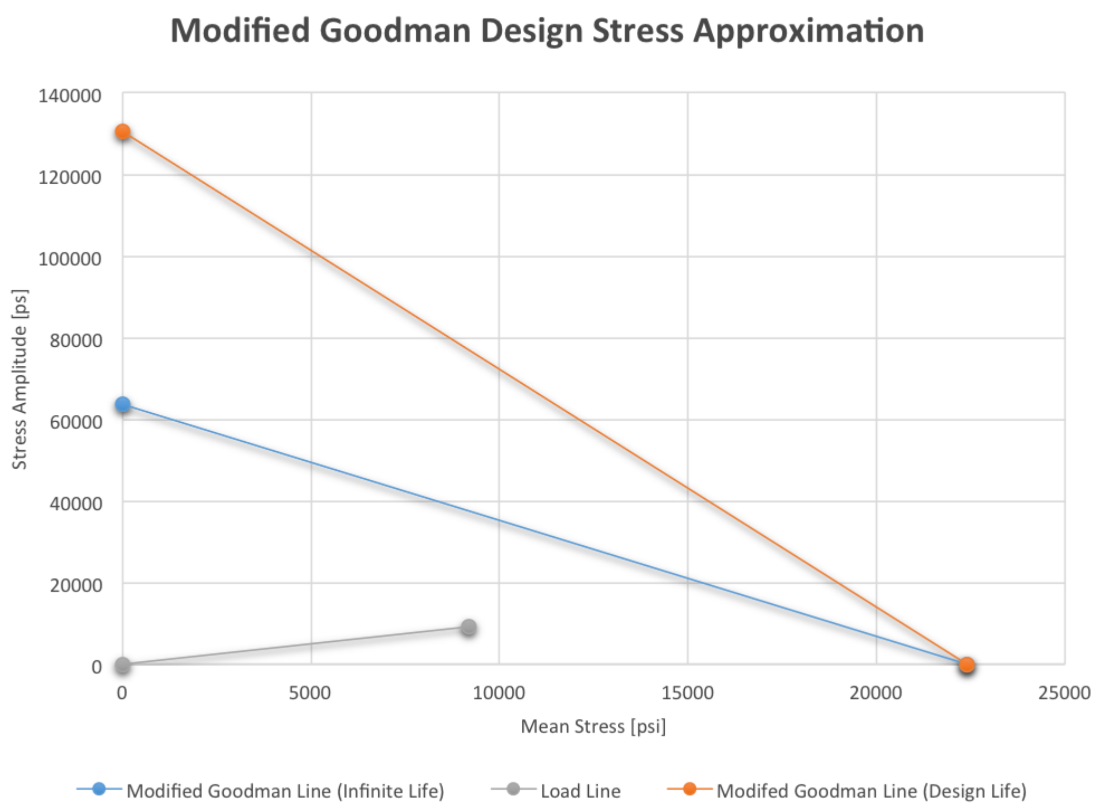

Figure 48: Modified Goodman design stress approximation.

Interpreting the graph above, the red line y-intercept represents the original endurance strength at 7000 cycles of 130534 psi, while the x-intercept represents the modified yield strength of 22412.37 psi. Any points that fall under the red line are considered safe for the required design life under the given cyclic loading. Similar to the red line, the blue line y-intercept represents the modified endurance strength of 63599.47 psi. This implies an infinite life for any points that fall under this line. Lastly, the grey line represents all three possible cyclic loading cases oscillating from 0 to 18400 psi. Since it is a case 2 loading with R=0, the line equation for this is y = x = 9200 psi. It can be concluded from the graph that the design is safe from fatigue failure, as the grey line does not cross the red line.

6.3.3.5. V-support Calculations Conclusion

To sum up, the V-tube structure will be constructed out of ASTM A513 Grade B square tubes. The optimal cross-section was 1.5 x 1.5 x 0.12 (11GA) since it gave a safe design at FOS 2.5 up to FOS 3.1. FEA analysis/simulation needs to be conducted to verify that this geometry will withstand the actual loading scenarios.

6.5 V-support Version 5

Version 5 of the V-support was designed with the goal of reducing excess material before we presented the design to our client again. As a team we identified two major areas where excess material could be removed. Firstly, in the mounting clamps, the flange area at the ends of the clamps and between the bolt holes sees substantially lower stress since the bolts clamping load would not travel a great distance axially out from the bolt. Secondly, the tubing extension below the cylindrical jack mount is essentially all wasted material as structurally it serves no purpose other than decreasing ground clearance. So it was decided the flanges would have material removed, and the tubing below the cylindrical mount would be cut off. This can be seen in Figure 49 and Figure 50 which display version 5 of the design.

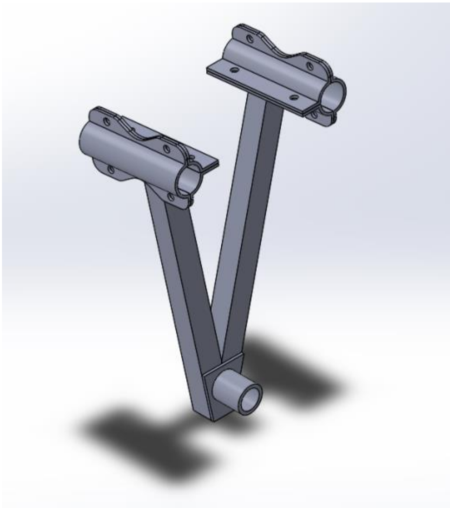

Figure 49: V-support version 5 rear iso view.

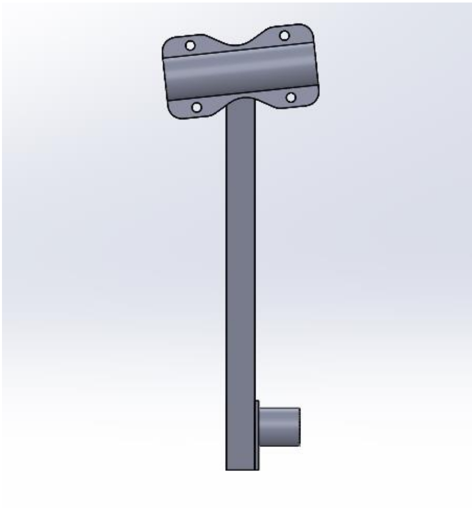

Figure 50: V-support version 5 side view.

With version 5 of the design now complete it was sent off to our client for further feedback. Upon the client reviewing the design with his rotary header team, additional issues were identified. Although the mounting brackets now did not extend outward past the center vertical plane of the driveline guards, they still were not happy with the clamping mount design. The client was worried that since the tractor tire treads are large enough to allow the driveline guard to fit between the treads (as seen in Figure 31), the possibility for tire damage was still present with this design and could create a huge expense for the owner. As a solution our client was willing to compromise. They were willing to allow us to make a design change in the rotary header to extend the driveline guards further forward so the tires would not contact our design. Also they gave us the option to allow us to weld the design directly onto the driveline guards, as they were more concerned about the tires, rather than a bolt on solution. This feedback would now need to be incorporated into the next version of the design.

6.6 V-support Final Design Version

The final version of the V-support was designed to incorporate the client's feedback. Firstly, our team focussed on determining how far the driveline guards could be extended to still allow complete rotation of the rotary header gearbox. It was determined that the maximum allowable extension was 12 [in]. The original and extended driveline guards can be seen in Figure 51 and Figure 52 respectively.

Figure 51: Side view of current driveline guards.

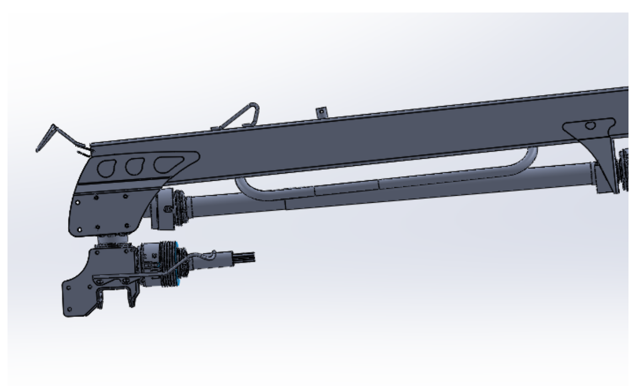

Figure 52: Side view of driveline guards extended 12 [in].

With the driveline guards now extended 12 [in] and still providing 3 [in] of horizontal clearance between the flat section of the guards to the edge of the gearbox when completely rotated, the new mounting height could now be calculated for the V-support. The new driveline guard mounting height for the V-support was only approximately 2 [in] lower than the previous one. As a result, this new mounting height didn't require many changes to the V-support design other than decreasing it's overall height accordingly to maintain the same mounting height as that of the gearbox. The other major change was the elimination of the mounting clamps all together as this was suggested for consideration by the client. Removal of the mounting clamps was favoured by our team because if not designed perfectly presented the opportunity for rotation and movement about the driveline guards. This movement could compromise the overall strength of the design should slippage occur and ultimately result in possible failure. The final version of the V-support design is seen in Figure 53 and Figure 54.

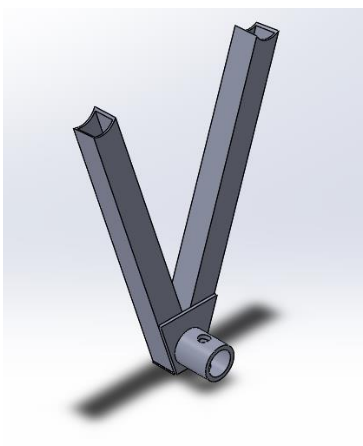

Figure 53: V-support final version rear iso view.

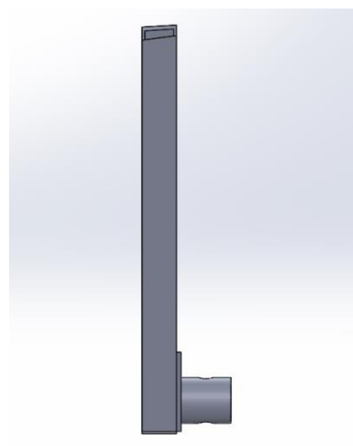

Figure 54: V-support final version side view.

The other change that was implemented into the final version of this design which is not easily seen in the isometric and side views is the addition of a bottom cap plate. Since the square tubing was cut flat below the cylindrical mounting bracket, this left the tubing wide open as seen in Figure 55. To completely enclose the design, as well as to add structural strength, an end cap plate was placed on the bottom of the design to completely enclose it which can be seen in Figure 56.

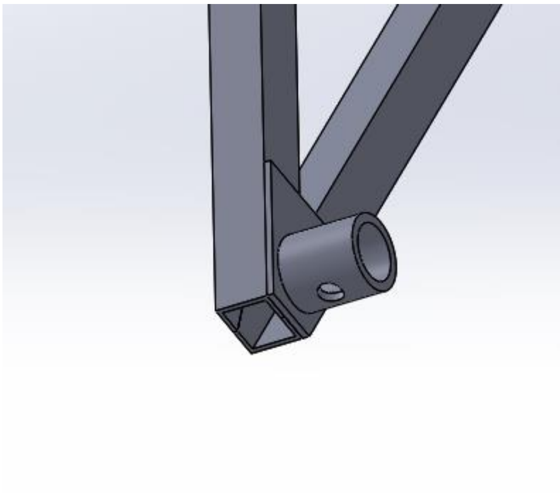

Figure 55: Bottom of V-support without cap.

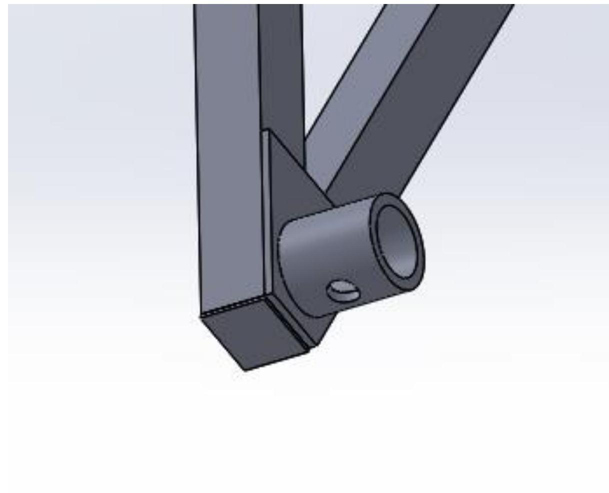

Figure 56: V-support with cap.

With the final design version now complete (refer to Appendix A.6.0 for detailed design drawings) it was then added to the master assembly to see if any issues were overlooked. The final version of the V-support in the master assembly is seen in Figure 57 and Figure 58.

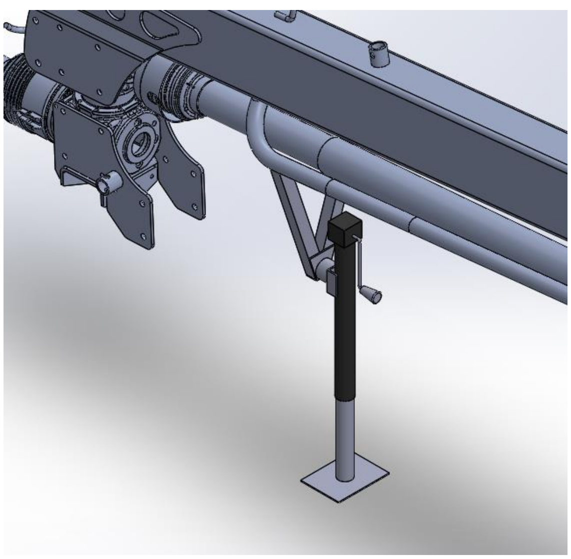

Figure 57: Iso view of V-support final version in assembly.

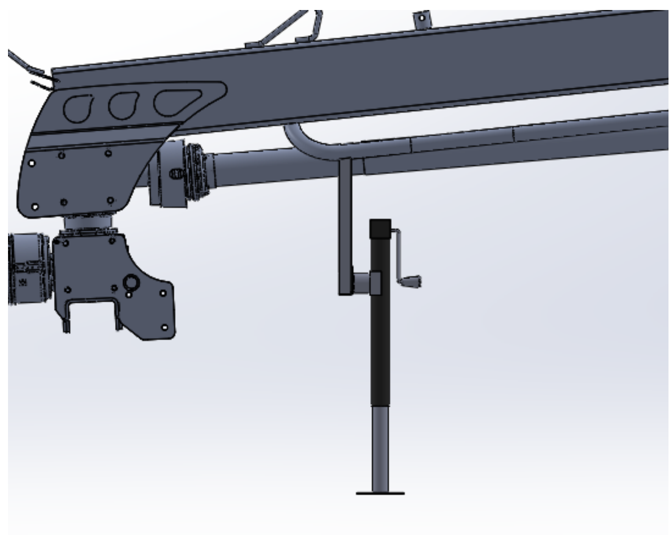

Figure 58: Side view of V-support final version in assembly.

Our team was very pleased with the final design version when placed into the master assembly. There were no unforeseen issues and the design fit exactly as planned. Furthermore, all concerns and issues raised by our client were now addressed.

After changes to the design and allowable production changes to the length of the guard bars, analytical calculations had to be performed on these extended guards. The analytical analysis had to be performed to ensure that the driveline guards could still support the load of the hitch weight after the geometric change.

6.6.1 Extended Driveline Guard Beam Analysis

The geometric change in the driveline guards was an addition of 304.8 mm in the straight section of the driveline guard. Additionally, due to the extension of the guard bars and the ability to mount the V-support to the guards via welding, the moment applied now has an x and y component. As the guards would not be changing material, the area moment of inertia remains the same for this section as for the previous section and was 1.63×10^{-7} m^4. Similar to the initial case performed in section 6.4.2, this section of the driveline guard was assumed to be a beam fixed at both ends with a distributed load applied to it. It is also important to note, that since the V-support structure is now mounted via a weld, the distance b of the applied load has been reduced. This loading situation is identical to Figure 38 above. The new dimensions for the driveline guard and loading situation are shown in TABLE XXII.

TABLE XXII: DIMENSIONS OF BEAM IN LOADING

Distance Symbol	Distance (m)
B	0.0381
C	1.1877
D	0.01905
E	1.20675
L	1.20675

Given that the driveline guard is round, the applied force can be rotated to any position around surface and the area moment of inertia does not vary. Due to this, the calculations were performed so that the view plane was parallel to the direction of the applied load on the guards. The shear force and bending moment calculations were performed in the same manner as previously discussed. The results of these calculations are shown in TABLE XXIII.

TABLE XXIII: SUMMARY OF CALCULATED SHEAR FORCES, BENDING MOMENTS ABD THEIR RESPECTIVE STRESSES ON THE EXTENDED DRIVELINE GUARD

Location	Force	Stress (MPa)
R_L	-317.36	0.46
R_R	-0.30	0.000438
M_L	5.8	0.086
M_R	0.12	0.0018
M_{max}	0.24	0.00035

The von Mises stress was calculated for the extended driveline guards in the same manner and was found to be 0.086 MPa.

6.6.2 Extended Driveline Guard Mounting

To determine the moment acting on the driveline guard mounts, the weight of the hitch had to be decomposed into x and y components based on the angle of the V supports. Figure 59 shows the weight of the header acting on the driveline guards and the x and y dimensions required to decompose the force into its x and y components.

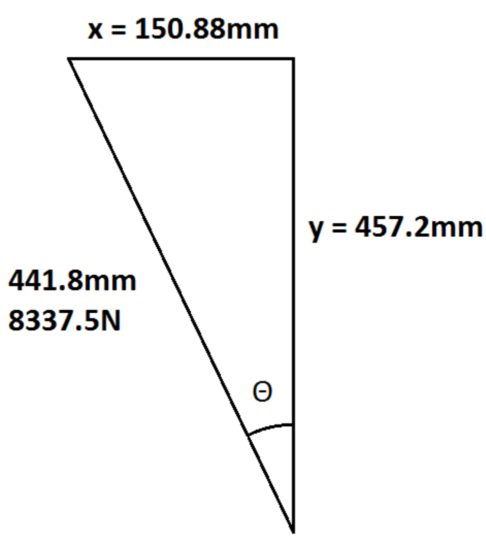

Figure 59: Force decomposition based on angle of v support.

The angle of the applied load was calculated as follows.

$$\theta = tan^{-1}(\frac{opp}{adj})$$

$$\theta = tan^{-1}\left(\frac{150.88}{457.2}\right)$$

$$\theta = 19.63°$$

With the applied angle known, the x and y forces could be calculated from the applied load. The x and y force components that are applied to the driveline guards of the header were calculated as follows.

$$F_x = \sin\theta * 8337.5$$
$$F_x = \sin(19.63) * 8337.5$$
$$F_x = 2800.29\ N$$

$$F_y = \cos\theta * 8337.5$$
$$F_y = \cos(19.63) * 8337.5$$
$$F_y = 7853.17\ N$$

The distance between the center of the driveline guard and the center of the mounting location had to be measured, similar to that of the normal guard bar analysis in Section 6.1.1.2. Again, taking the same measurements as those in Figure 39 these dimensions were determined using the measurement tool in SolidWorks. The measured values of these dimensions are shown in TABLE XXIV.

TABLE XXIV: DRIVELINE GUARD DISTANCES

Distance	Distance (m)
H	0.10744
W	0.10934

With the applied load decomposed into the x and y directions, and the distances from the applied load and the driveline guard, the moment around the mounting base of the driveline guard could be calculated. This calculation was performed by multiplying the force by the perpendicular distance between the force and the driveline mounting location. The forces were assumed to be acting at the center of the driveline guard.

$$M_A = F_x * h + F_y * w$$

$$M_A = 2800.29 * 0.10744 + 7853.17 * 0.10934$$

$$M_A = 1159.53\ Nm$$

As the moment around the mounting base of the driveline guard is less than the maximum allowable moment, the driveline guard will not fail due to bending around the mounting base. At this point our team was confident in the design and was ready to take it into the numerical analysis phase of the detailed design process. This will allow the team to verify the final design and ensure it will not fail in the field of operation.

7.0 Numerical Analysis of Design

To verify the analytical analysis of the design, a numerical analysis utilizing finite element analysis (FEA) was performed. In no way is this FEA to be considered a professional final analysis, however every effort was made to ensure the most accurate results possible were obtained given our skills and resources. Using ANSYS Workbench 17, three separate loading scenarios will be considered for the V-support design. This section covers the meshing and its convergence, boundary conditions and loads applied to the finite element model, followed by the corresponding stress results, and finally a summary to conclude the findings.

7.1 Meshing

To ensure the best possible accuracy of the obtained results, it was important to produce a properly meshed finite model. This was accomplished by using the adaptive meshing feature within ANSYS. The maximum allowable change in stress between the final iterations of meshing was defined as 6%, ensuring a reasonable convergence in stress results. Unfortunately, a greater degree of convergence was not obtainable with our resources as the ANSYS Workbench student license is limited to a defined number of nodes which was exceeded when a greater degree of convergence was specified. Preferably, a lower convergence percentage would be desired but was not achievable given the limitations. The maximum number of refinement loops for the solution was also set to five. This allowed the software to perform at max four additional iterations of mesh refinement to achieve a minimum of 6% or greater convergence in the stress results. The initial mesh prior to any adaptive meshing iterations is seen in Figure 60.

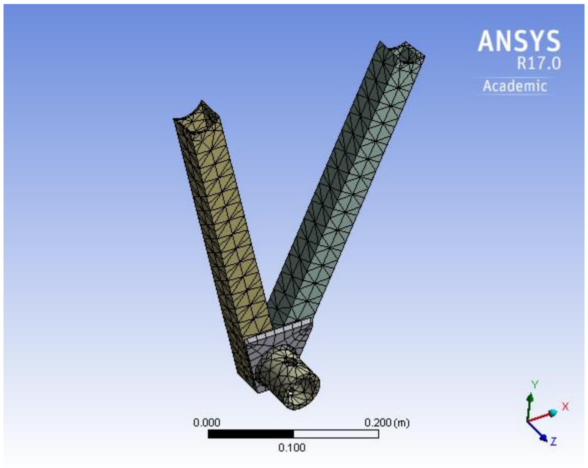

Figure 60: Mesh prior to adaptive iterations.

For each loading scenario, the defined minimum convergence of 6% with five iterations will be used to obtain an adaptive mesh for that loading scenario.

In an effort to reduce the effects of stress singularities, a modified FEA model of the design was produced which is seen in Figure 61.

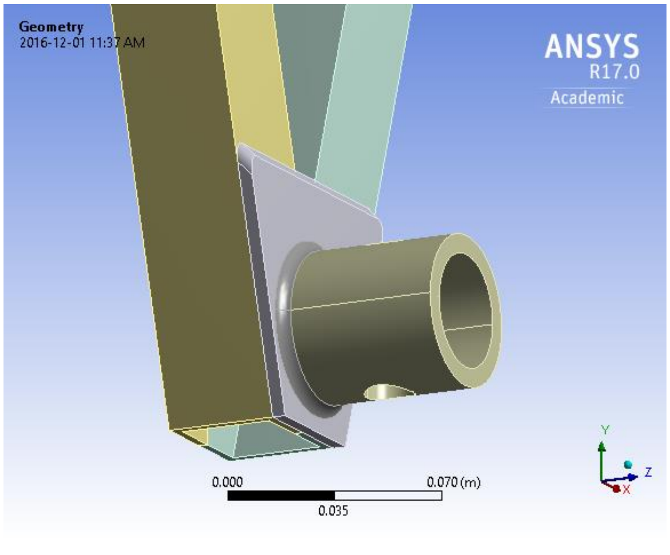

Figure 61: Modified FEA design model.

This modified model features a fillet around the cylinder where it joins the mounting plate. The fillet replicates a 1/8 [in] weld bead, and will allow for stress convergence at the end of the cylinder's outer face where it meets the plate. Without this fillet the stress would never converge at this location because of the square edge and this location being the point of highest stress in the design. The top edge of the mounting plate where it met the square tubing also saw high stresses, so it required modifications as well to allow for convergence. A chamfer was applied to the top of the plate and the top two corners were then filleted to remove the sharp corners.

7.2 Boundary Conditions

Picking the correct boundary conditions is crucial to obtain the correct results. Under or over constraining the model will greatly skew these results. Therefore, it was essential only the most realistic boundary conditions be applied. To replicate the real life fixations of the V-support, the top faces of the square tubing were completely constrained from movement as seen in Figure 62.

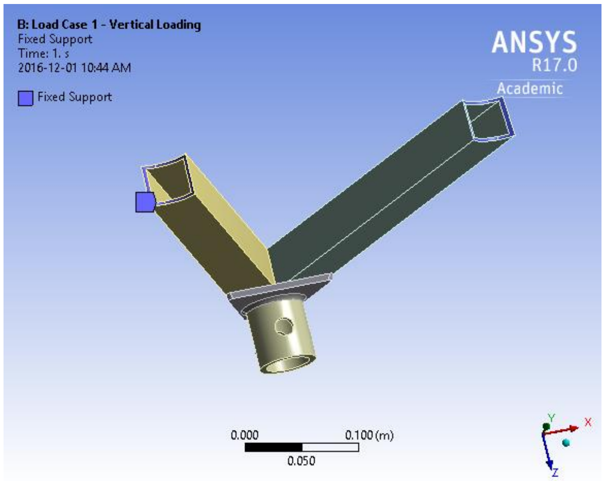

Figure 62: Fixed boundary conditions for design analysis.

By completely constraining the movement of the tubing's top faces in all directions, this replicates the welds around the entire perimeter of the square tubing which would fix the design to the swing tongue. The swing tongue is by far the most solid part of the design which would see nearly zero displacement under loading across the span where the V-support is attached, indicating this is an accurate fixation for this finite model.

7.3 Loading Scenarios

The client specified that the design must support a vertical load of 1500 [lbs] which will be the first loading scenario considered. This loading case replicates a perfect scenario where the V-support does not see any other external forces. However, our team recognized that this scenario is likely not what will actually cause the design to fail. In a real life scenario, the trailer jack can easily be accidentally left down while in operation. If this was the case, it is important to know what force the design can withstand from an impact if the trailer jack were not removed for operation and made contact with the ground. Therefore, the second and third loading scenarios will determine the maximum allowable force that can be applied to the front and side of the design respectively.

7.3.1 Vertical Loading

For the first loading scenario, a 1500 lbs vertical load in the positive y-direction was applied to the bottom half of the trailer jack cylindrical mount face on the V-support as seen in Figure 63.

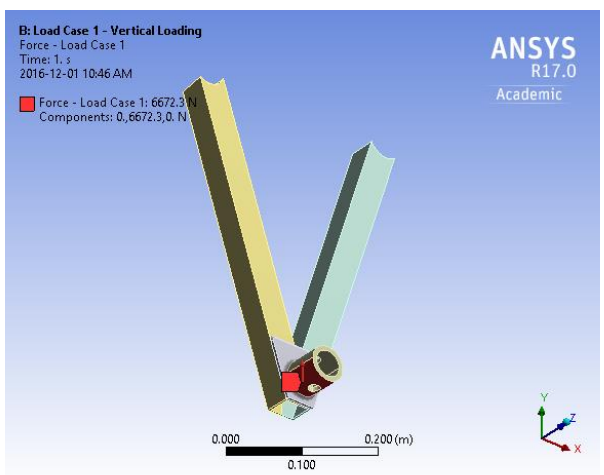

Figure 63: Vertical loading scenario.

This replicates the force that would be transmitted from the trailer jack mount through to the mating mount on the V-support. It was important to only apply the distributed load across the bottom of the cylindrical mounting surface as this is what is realistically occurring.

7.3.2 Frontal Loading

For the second loading scenario, it was assumed that if a frontal impact occurred it would take place at the base of the trailer jack. Assuming the trailer jack was at its average location of operational height this would result in the force from the impact being applied at a location of 26.5 [in] (0.673 metres) from the center of the cylindrical mount on the V-support. Therefore, by multiplying the applied load at the base of the jack stand by the distance from the V-supports cylindrical mount to the base of the jack stand, the equivalent moment on the V-support can be calculated. In Figure 64, the corresponding positive moment applied about the x-axis on the entire outer surface of the V-support's cylindrical mount can be seen.

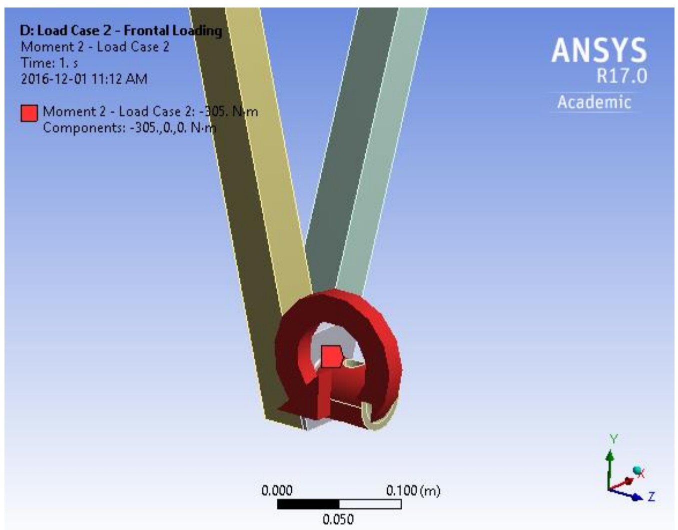

Figure 64: Frontal loading scenario applied moment.

Since the purpose of loading scenarios two and three were to determine the force to cause failure, iterations were run until the applied moment caused the V-support to reach its yield strength of 126 [MPa], which already incorporates the safety factor of 2.5. The moment that caused yielding, and the corresponding max force along with the stress results will be presented in the results section.

7.3.3 Side Loading

The third loading scenario which covers side loading at the base of the jack was analysed in the exact same approach as the second loading scenario. To replicate a force being applied about either side of the jack's base, a corresponding moment was applied on the V-supports entire cylindrical mounting face about the z-axis as seen in Figure 65.

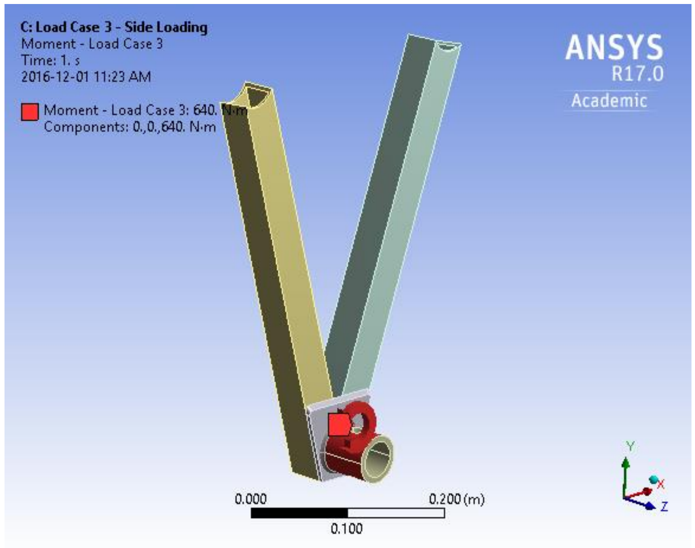

Figure 65: Frontal loading scenario applied moment.

As in scenario two, iterations were run until the moment that caused yielding of the V-support was found. The results of this will also be presented in the next section, detailing the stress results and maximum allowable side force.

7.4 FEA Results

The results obtained for adaptive mesh convergence for each of the three loading scenarios is presented in this section. Following the adaptive mesh details, the stress results for the final iterations of adaptive meshing for each of the scenarios will also be presented.

7.4.1 Vertical Loading Results

For the case of the vertical load applied in the positive y-direction about the bottom of the cylindrical mounts outer face, three adaptive meshing iterations were required. The convergence plot is seen in Figure 66.

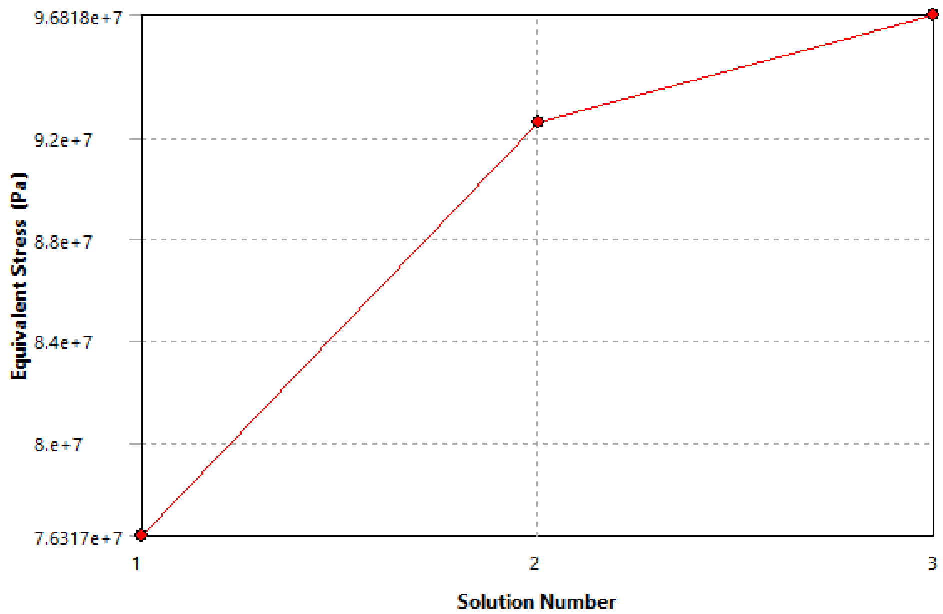

Figure 66: Loading scenario one convergence plot.

The summary of the results for this adaptive solution for mesh refinement are presented in TABLE XXV. The table displays the progression of the iterations used to achieve a convergence of 4.45%.

TABLE XXV: LOADING SCENARIO ONE ADAPTIVE MESH DETAILS

Iteration	Max Equivalent Stress [MPa]	Change %	Nodes	Elements
1	76317000		7063	3153
2	92601000	19.28	24649	13429
3	96818000	4.45	74614	45812

This final adaptive mesh was now used to obtain the stress results for the first loading scenario which are presented in Figure 67 and Figure 68.

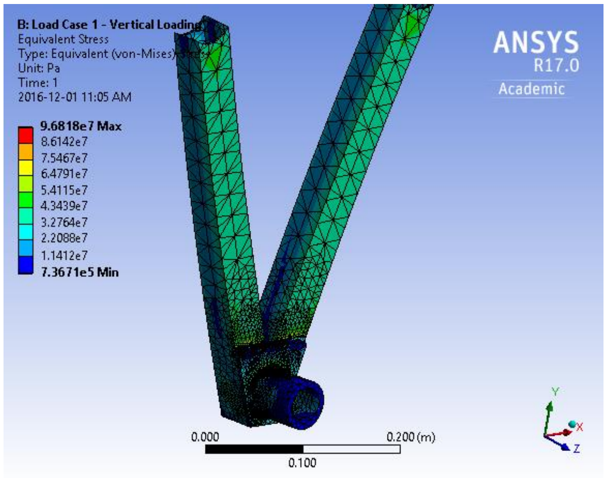

Figure 67: Loading scenario one stress results.

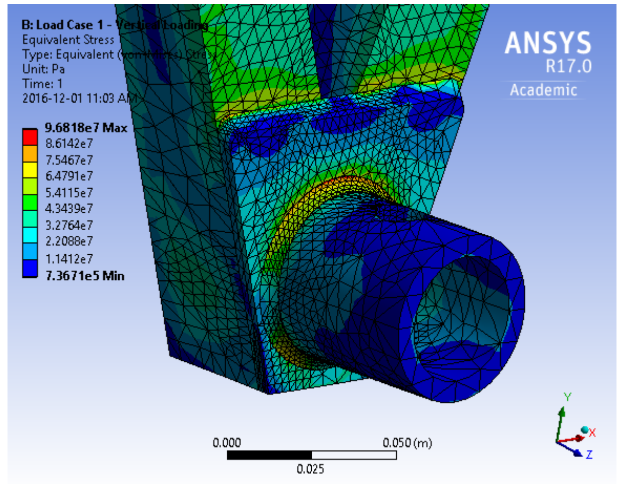

Figure 68: Zoomed view of max stress location in loading scenario one.

A max stress of 9.6818×10^{7} Pa was obtained when the 1500 lbs vertical load was applied. This indicates a safety factor of 3.30 is present given the materials yield strength is 3.1716×10^{8} Pa. The highest stress by far was seen at the top of the cylindrical mount where it meets the sheet metal plate on the edge of the fillet as expected.

7.4.2 Frontal Loading Results

For the case of frontal loading, three adaptive meshing iterations were required. The convergence plot is seen in Figure 69.

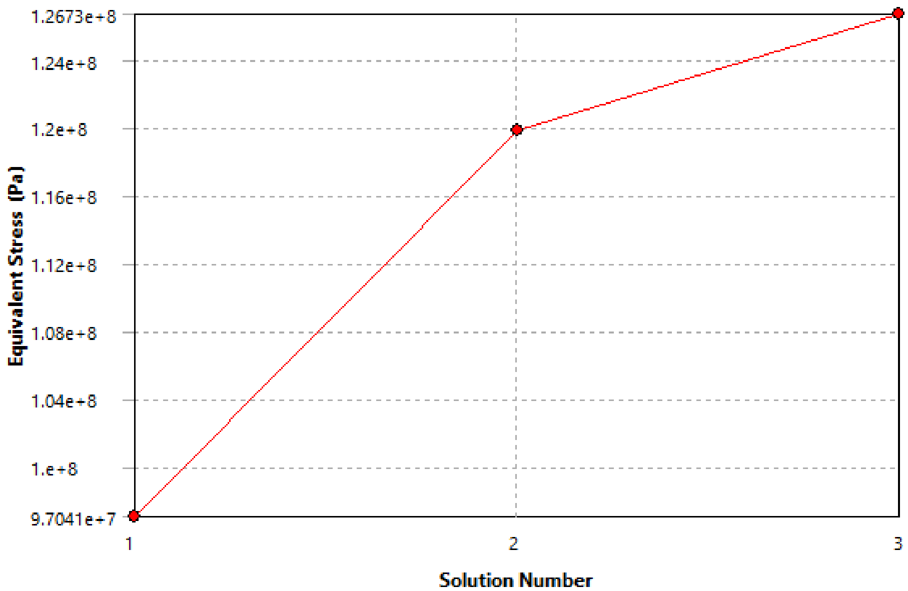

Figure 69: Loading scenario two convergence plot.

The summary of the results for this adaptive solution for mesh refinement are presented in TABLE XXVI. The table displays the progression of the iterations used to achieve a convergence of 5.60%.

TABLE XXVI: LOADING SCENARIO TWO ADAPTIVE MESH DETAILS

Iteration	Max Equivalent Stress [MPa]	Change %	Nodes	Elements
1	97041000		7063	3153
2	119830000	21.02	19425	10279
3	126730000	5.60	45294	26755

This final adaptive mesh was now used to obtain the stress results for the second loading scenario which are presented in Figure 70 and Figure 71.

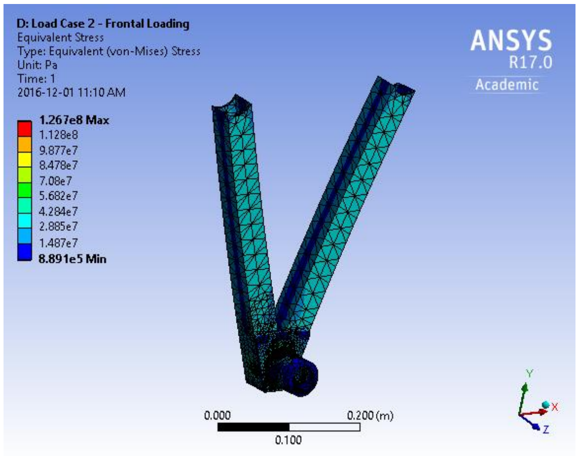

Figure 70: Loading scenario two stress results.

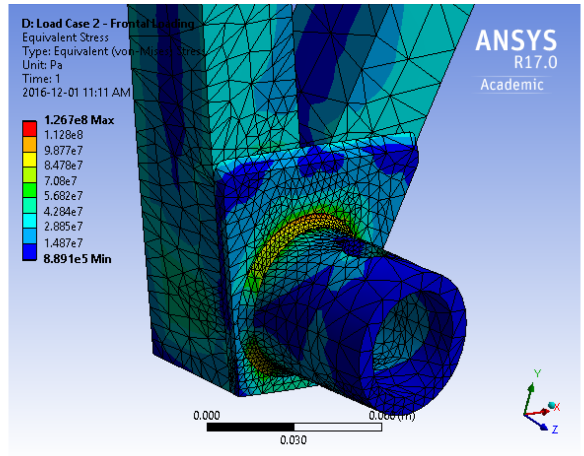

Figure 71: Zoomed view of max stress location in loading scenario two.

Maintaining the safety factor of 2.5, the maximum allowable stress of 1.2686×10^8 Pa was obtained when a moment of 305 N*m was applied. Dividing the moment by 0.673 m, the distance from the jack stands base to the center of the cylindrical mount, a force of 453.2N (101.88 lbs) was calculated. This indicates the V-support can withstand a force of 101.88 lbs applied at the base of the jack stand in the positive z direction with a safety factor of 2.5 applied.

7.4.3 Side Loading Results

For the case of side loading, only two adaptive meshing iterations were required. The convergence plot is seen in Figure 72.

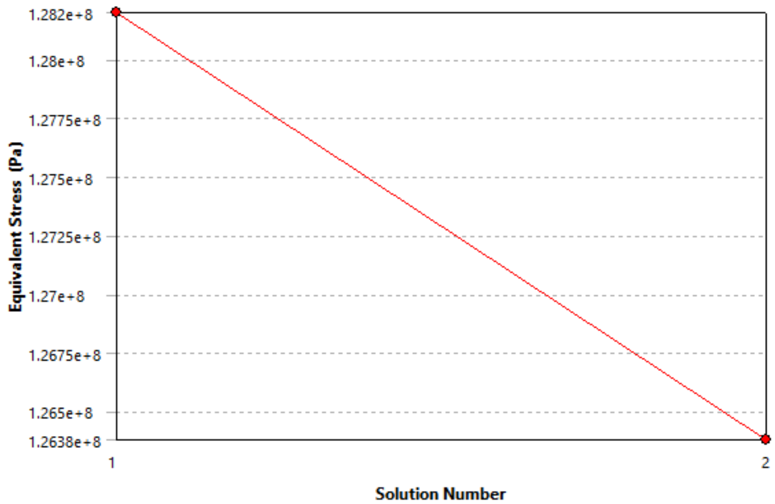

Figure 72: Loading scenario three convergence plot.

The summary of the results for this adaptive solution for mesh refinement are presented in TABLE XXVII. The table displays the progression of the iterations used to achieve a convergence of -1.43%.

TABLE XXVII: LOADING SCENARIO 3 ADAPTIVE MESH DETAILS

Iteration	Max Equivalent Stress [MPa]	Change %	Nodes	Elements
1	128200000		7063	3153
2	126380000	-1.43	24853	13525

This final adaptive mesh was now used to obtain the stress results for the third loading scenario which are presented in Figure 73 and Figure 74.

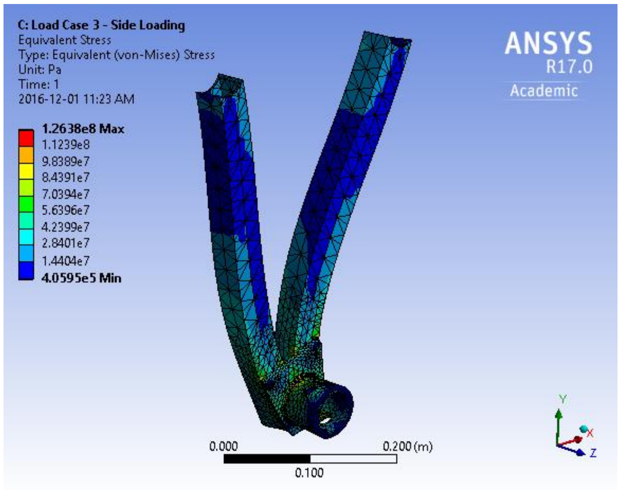

Figure 73: Loading scenario three stress results.

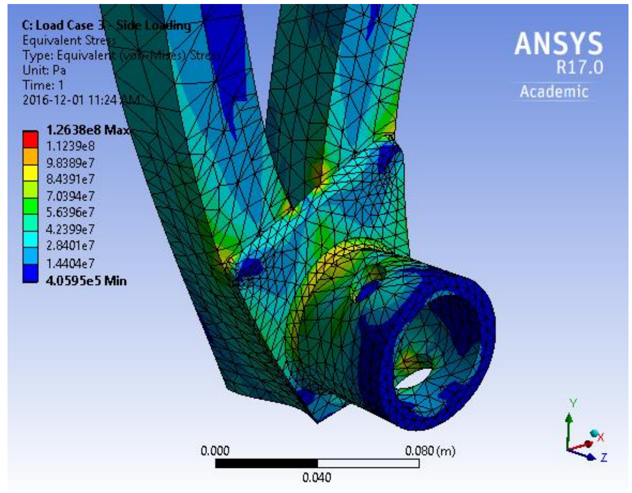

Figure 74: Zoomed view of max stress location in loading scenario three.

Maintaining the safety factor of 2.5, the maximum allowable stress of 1.98×10^{8} Pa was obtained when a moment of 640 [Nm] was applied. Dividing the moment by 0.673 m, the distance from the jack stands base to the center of the cylindrical mount, a force of 950.97 N (213.79 lbs) was calculated. This indicates the V-support can withstand a force of 213.79 lbs applied at the base of the jack stand in the positive x direction, with a safety factor of 2.5 applied.

7.5 FEA Summary

Through numerical analysis of the V-support design in ANSYS 17, the analytical hand calculations were verified. The design successfully supports the 1500 lbs load applied on it by the swing tongue, while maintain a safety factor of 3.30 which is slightly greater than the required value of 2.5. It was found the design can withstand a fontal load applied at the base of the trailer jack of 101.88 [lbs] while maintaining a safety factor of 2.5. For the case of side loading, a maximum allowable force of 213.79 [lbs] is permissible at the side of the trailer jacks base, for the design to maintain this same factor of safety of 2.5. The highest stress for all 3 cases occurred at the edge of the cylindrical mount and the mounting plates face. Given welds will be applied entirely across this area, the stress will likely be less than calculated in this analysis indicating a successfully designed V-support that is safe to use.

8.0 Numerical Verification of Driveline Guards

In order to ensure the driveline guards were able to withstand the forces being transmitted from the mounting of the V-support to it, an FEA verification of the analytical analysis was done. The team felt this FEA analysis was especially important as our analytical method assumed a perfectly straight beam fixed at both ends. These assumptions and neglecting the curves at the ends of the driveline guards were necessary for hand calculations in order to achieve a result. However, in doing so the accuracy of these calculations may have been skewed so FEA verification will help to ensure failure of these guards is not likely. The same methodology used in the previous FEA section was utilized.

8.1 Meshing

Again to ensure a good level of accuracy in the results adaptive meshing was used. However, this time manual adaptive meshing was performed to save computational requirements and time as the driveline guard bar was a much larger structure as opposed to the V-support. The point of highest stress was found to be at the very top inside corner of the guard bar along its edge where it is welded onto the swing tongue. For this reason, the mesh was refined two additional times along this edge with the final mesh refinement seen in Figure 75 and Figure 76.

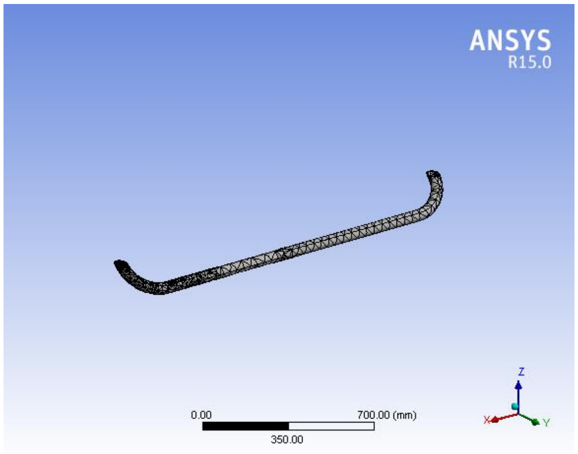

Figure 75: Refined guard bar mesh.

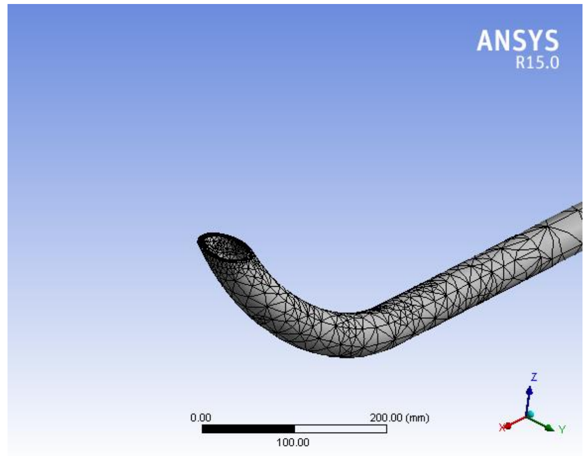

Figure 76: Refined guard bar mesh in high stress area.

This refined mesh was now utilized for the entire analysis of the driveline guard bars.

8.2 Boundary Conditions

To accurately represent how the driveline guard bars are attached to the swing tongue in real life, completely constrained fixations were applied to the two faces oat the ends of the tubing. These applied boundary conditions are seen in Figure 77.

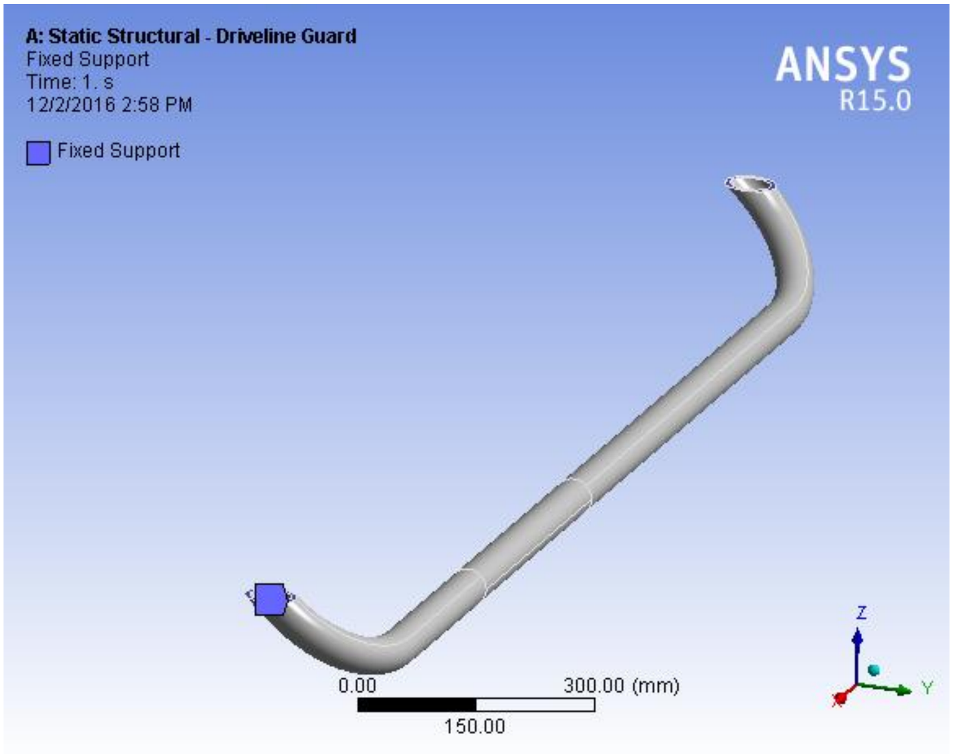

Figure 77: Fixed boundary conditions applied to guard bar.

Having the two faces of the guard bar completely restrained reflects exactly what is experienced in real life, when these two ends of the tubing are welded onto the swing tongue around their perimeter completely.

8.3 Loading

Each driveline guard bar will experience half of the required load that the V-support must support of 1500 lbs. Therefore, a 750 lbs (3113.8 N) was applied to the guard bar as a distributed load. This load was distributed over a 1.5 inch square imprinted face as seen in Figure 78 with a better image of the imprinted face displayed in Figure 79.

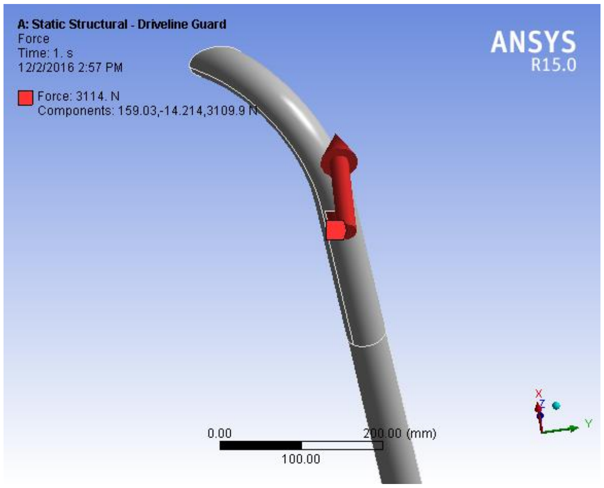

Figure 78: Applied distributed load to guard bar.

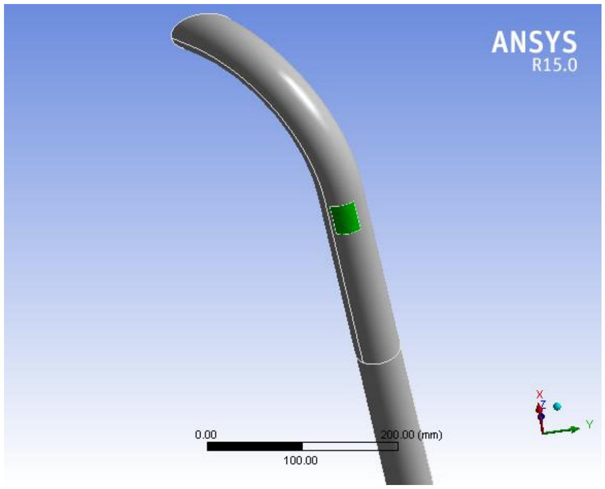

Figure 79: Guard bar imprinted face for load application.

The 750 lbs load in the positive z-direction was applied on the 1.5 inch imprinted face as this is the most realistic way to represent the actual loading scenario. The imprinted face was also positioned such that it was right at the start of the straight portion of the guard bar, exactly where the v-support is mounted.

8.4 Results

Running the analysis with the applied refined mesh, boundary conditions, and load a convergence of 12.39% was achieved in the stress results between the second and third manual iteration. The convergence plot is seen in Figure 80.

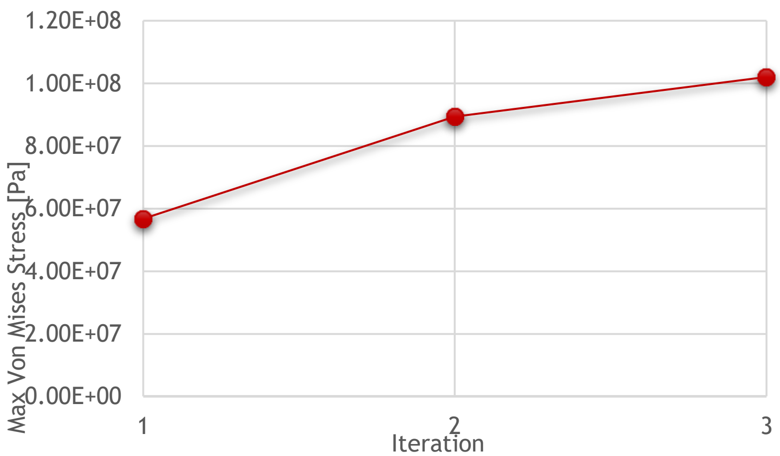

Figure 80: Guard bar FEA mesh convergence results.

The max stress results obtained from the first two iterations of mesh refinement can be seen in APPENDIX A.4.0.

A final max stress was obtained of 102.06 MPa. The stress distribution can be seen in Figure 81, with the zoomed view of the highest stress region pictured in Figure 82.

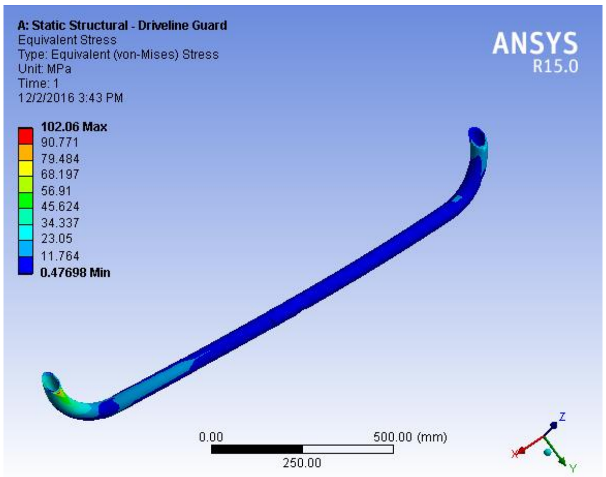

Figure 81: FEA von Mises stress results.

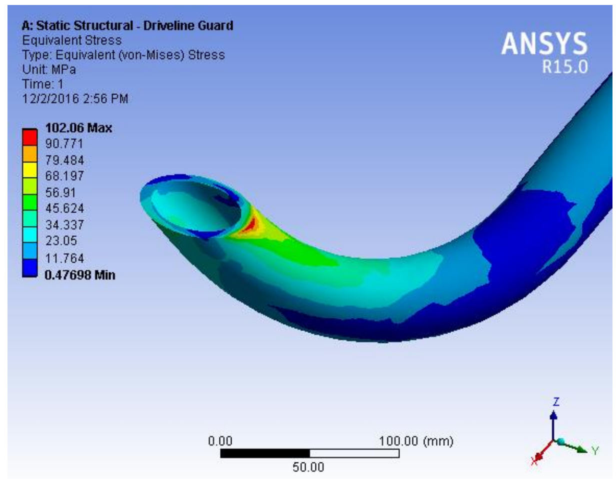

Figure 82: Zoomed view of highest stress region.

The max stress experienced of 102.06 MPa indicates a factor of safety of 2.34 when compared to the yield strength of the driveline guard bars ASTM A53 Grade B, Type E material of 240 MPa.

A higher convergence was desirable however the area of highest stress was reduced to less than a 2 millimetre area from the edge of the guard bar tubing's face which realistically would be completely welded. Having this area covered in weld bead would increase the cross sectional area, and would therefore see a decrease in this stress singularity allowing for a higher level of convergence, with a higher factor of safety as well. The zoomed view showing the size of the max stress area of less than two millimetres in seen in Figure 83, noting the scale on the bottom of the image indicating the scale size.

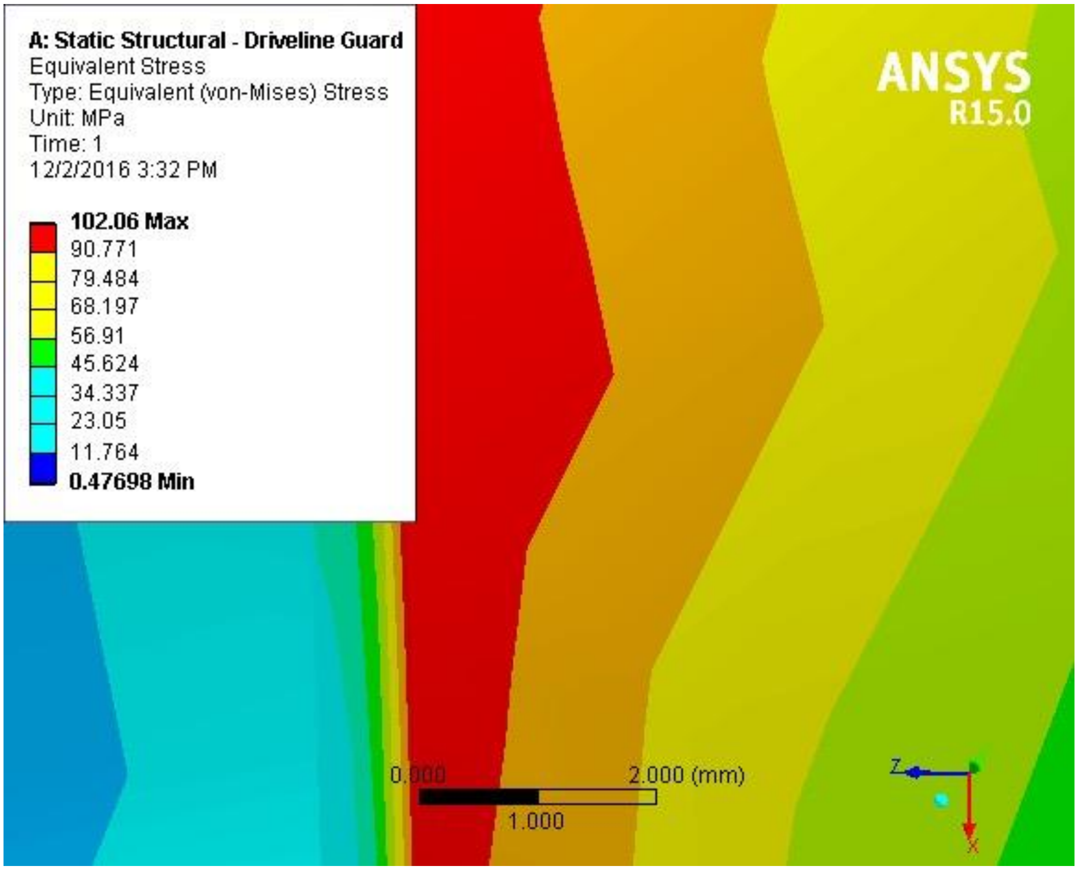

Figure 83: High stress area contained with weld bead.

The final max stress of 102.06 MPa pictured in red, can be seen to have been contained within a distance of 2 mm from the tubes edge. This ensures that the weld bead of 2 mm will completely surround this area of concern.

8.5 FEA Summary

The numerical finite element analysis of the driveline guard bars successfully verified the hand calculations and their assumptions. The guard bar was found to have a factor of safety of 2.34 corresponding to a max stress of 102.06 MPa. This max stress area was contained to a 2 mm region which will be completely surrounded in a weld bead, indicating that a decreased max stress and higher factor of safety will be achieved in the real life scenario.

9.0 FMEA

Our client advised the team that the design has to conform to European Union Machinery Directive 2006/42/EC. Conformity to this standard can be achieved in one of two methods, either by testing a prototype or by performing an analysis of the machine that includes all potential loading conditions. This analysis is accomplished by performing an FMEA, which looks at the stability of the machine. The high prototyping and testing costs were not within the team's budget, therefore an analysis was performed to ensure that the hitch was stable under the potential loading conditions.

FMEA is performed by determining and assigning numerical values to the severity of potential failures, how likely that failure is to occur and how likely it is that the failure would be detected. Once each aspect of the failure was analyzed, the numerical values were multiplied by each other to determine the risk priority number. For any failure with a risk priority number above 100, the design would have to be further analyzed for that failure and design changes would be required to lower that risk priority number.

For each failure mode analyzed, the team had to assign a value of severity to the failure, a value to the frequency of the failure and a value to the ability of the design to detect the failure. For the severity of the failure, the assigned value was based on the perceived level of customer dissatisfaction with the jack stand and possible safety considerations of the failure. The numerical value for the severity of the failure mode was determined using the criteria shown in TABLE XXVIII. The numerical value for the frequency of the failure was based on the rate of occurrence that could cause jack stand failure. TABLE XXIX displays the numerical values for the frequency of the failure and the rate of occurrence each frequency applies to. The relationship between the ability of the design to detect the failure and the numerical value applied to the failure is shown in TABLE XXX. The teams FMEA can be seen in TABLE XXXI. As the jack stand is a currently used MacDon product, the team was not required to analyze it.

TABLE XXVIII: SEVERITY INDEX

Design	Description	Ranking
Hazardous w/o Warning	**VERY HIGH** Severity Ranking when Variation or Potential Failure Mode affects Safe Machine Operation and/or Involves Non-Compliance with Government Regulations - **WITHOUT WARNING**	**10**
Hazardous with Warning	**VERY HIGH** Severity Ranking when Variation or Potential Failure Mode affects Safe Machine Operation and/or Involves Non-Compliance with Government Regulations - **WITH WARNING**	**9**
Very High	Vehicle / Item **INOPERABLE, LOSS OF PRIMARY FUNCTION**	**8**
High	Vehicle / Item OPERABLE, But at a **REDUCED LEVEL OF PERFORMANCE.** *Customer Very Dissatisfied.*	**7**
Moderate	Vehicle / Item OPERABLE, But **COMFORT / CONVENIENCE ITEM(S) INOPERABLE.** *Customer IS Dissatisfied.*	**6**
Low	Vehicle / Item OPERABLE, But **COMFORT / CONVENIENCE ITEM(S) OPERABLE at a REDUCED LEVEL** of Performance**.** *Customer Somewhat Dissatisfied.*	**5**
Very Low	Fit & Finish / Squeak & Rattle Item Varies or Does not conform. *Defect* ***NOTICED BY MOST CUSTOMERS*** *(>75%)*	**4**
Minor	Fit & Finish / Squeak & Rattle Item Varies or Does not conform. *Defect* ***NOTICED BY 50% CUSTOMERS***	**3**
Very Minor	Fit & Finish / Squeak & Rattle Item Varies or Does not conform. *Defect* ***NOTICED BY DISCRIMINATING CUSTOMERS (<25%)***	**2**
None	No Discernable Effect	**1**

TABLE XXIX: FREQUENCY INDEX

Probability of Failure	Possible Failure Rates	Frequency	Ranking
Very High: Persistent Failures	≥ 20 per 100 vehicles, OR 200 per thousand vehicles/items	≥ 1/10	10
Very High: Frequent Failures	10 per 100 vehicles, OR 100 per thousand vehicles/items	1/20	9
	5 per 100 vehicles, OR 50 per thousand vehicles/items	1/50	8
Moderate: Occasional Failures	3.3 per 100 vehicles, OR 33 per thousand vehicles/items	1/100	7
	2 per 100 vehicles, OR 20 per thousand vehicles/items	1/200	6
Low: Relatively Few Failures	1 per 100 vehicles, OR 10 per thousand vehicles/items	1/500	5
	0.5 per 100 vehicles, OR 5 per thousand vehicles/items	1/1000	4
	0.3 per 100 vehicles, OR 3 per thousand vehicles/items	1/2000	3
Remote: Failure is Unlikely	0.2 per 100 vehicles, OR 2 per thousand vehicles/items	1/10,000	2
	0.1 per 100 vehicles, OR 1 per thousand vehicles/items	< 1/100,000	1

TABLE XXX: DETECTION INDEX

Probability of Design Detection	Design Detection	Ranking
Almost Impossible	Design Control will not and/or cannot detect a potential cause/mechanism and subsequent failure mode; or there is no design control.	10
Very Remote	Very remote chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	9
Remote	Remote chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	8
Very Low	Very Low chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	7
Low	Low chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	6
Moderate	Moderate chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	5
Moderately High	Moderately High chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	4
High	High chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	3
Very High	Very High chance the Design Control will detect a potential cause/mechanism and subsequent failure mode.	2
Almost Certain	Design Control will almost certainly detect a potential cause/mechanism and subsequent failure mode	1

TABLE XXXI: FMEA RESULTS FOR THE V-SUPPORT JACK STAND

Item or Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s)/Mechanism(s) of Failure	Frequency	Current Design Controls	Detectability	RPN
Driveline Guards	Driveline Guard Tube Bends	Jack Slightly Bent (operator is still able to lift and lower jack)	6	Jack is in use while tube bends under higher than expected loading, structural tubes bent in field and jack is used and loading changes on jack.	3	Company Standards, Supplier Quality Standards, ASTM Standards	5	90
			6	Jack is used on even terrain, thereby causing uneven loading on driveline guards.	3	Company Standards, Supplier Quality Standards, ASTM Standards	5	90
		Jack Significantly Bent (operator is no longer able to lift and lower jack)	8	Jack is in use while tube bends under loading that is significantly higher than expected, structural tubes significantly fail in field use (hit the ground) and jack is still used.	3	Company Standards, Supplier Quality Standards, ASTM Standards	4	96
	Driveline Guard Tube Breaks	Damage to Driveline Guard	8	V-support structural tubes are bent enough (via hitting object/ground) to bend driveline guards.	3	Company Standards, ASTM Standards	4	96
		Jack Breaks	9	Significantly higher than expected loading is applied to swing tongue while jack is in operation, jack fails.	2	Company Standards, Supplier Quality Standards, ASTM Standards	3	54
		Unit Falls to Ground	9	Structural tubes fail while header is in use, operator still uses jack and tubes are not able to support header weight.	2	Company Standards, ASTM Standards	5	90
	Side Impact	Damage to Driveline	8	V-support tubes are bent due to side loading. The tubes then bend the driveline guards enough to hit driveline, bending driveline.	3	Company Standards, ASTM Standards, AWS Standards	3	72

		Jack Slightly Bent (operator is still able to lift and lower jack)	7	Jack is structurally compromised due to side loading on the jack causing the jack to bend	6	Company Standards, Supplier Quality Standards, ASTM Standards, AWS Standards	2	84
		Jack Significantly Bent (operator is no longer able to lift and lower jack)	8	Jack is structurally compromised due to side loading on the jack and the jack fails completely	3	Company Standards, Supplier Quality Standards, ASTM Standards	4	96
		Header Leans	8	Structural tubes are damaged slightly from a side impact, jack is not damaged in any way, the header leans slightly to one side or the other.	6	Company Standards, Supplier Quality Standards, ASTM Standards	2	96
		Header Falls to Ground	10	V-support structural tubes are bent enough to prevent V-support from supporting weight of swing tongue.	3	Company Standards, ASTM Standards	3	90
	Weld Failure	Damage to Driveline Guard	8	Side impact to the swing tongue causes weld failure in V-support, thereby bending driveline guards enough to hit driveline.	3	Company Standards, AWS Standards, ASTM Standards	2	48
		Jack Slightly Bent (operator is still able to lift and lower jack)	6	Driveline guards are improperly welded to the swing tongue. Welds begin to fail under loading, thereby bending the driveline guards when hitch is put on jack.	2	Company Standards, Supplier Quality Standards, AWS Standards	7	84
		Jack Significantly Bent (operator is no longer able to lift and lower hitch)	8	Driveline guards are improperly welded to the swing tongue. Welds significantly fail under loading, thereby bending the driveline guards and jack when hitch is put on jack.	1	Company Standards, Supplier Quality Standards, ASTM Standards	8	64
Jack Mount	Weld Failure	Unit Falls to Ground	10	Driveline guards are improperly welded to the swing tongue. Welds fail completely under loading, driveline guards fall off when hitch is on jack	2	Company Standards, AWS Standards	4	80

		Jack Slightly Bent (operator is still able to lift and lower jack)	6	Jack mount plate is improperly welded to jack mount structure	2	Company Standards, Supplier Quality Standards, AWS Standard	6	72
		Jack Significantly Bent (operator is no longer able to lift and lower hitch)	8	Jack mount plate is improperly welded to jack mount structure, significant weld failure	2	Company Standards, Supplier Quality Standards, AWS Standards	5	80
	Over Loaded	Unit Falls to Ground	10	Jack mount plate shears off jack mount structure	1	Company Standards	3	30
		Jack Slightly Bent (operator is still able to lift and lower jack)	6	Jack mount plate is too thin for loading, begins to buckle	2	Company Standards, Supplier Quality Standards	7	84
		Jack Significantly Bent (operator is no longer able to lift and lower jack)	8	Jack mount plate buckles under applied load, beings to significantly deform.	2	Company Standards, Supplier Quality Standards	6	96
V-support Structure	Square Tubing Bends	Unit Falls to Ground	10	Jack mount plate shears off jack mount structure	1	Company Standards	8	80
		Jack Slightly Bent (operator is still able to lift and lower jack)	6	V-support structure hits the ground while in use deforms slightly, hitch is overloaded. Jack is still used on deformed structure and is affected due to different loading.	4	Company Standards, Supplier Quality Standards	4	96
		Jack Significantly Bent (operator is no longer able to lift and lower jack)	8	V-support structure hits the ground and deform significantly while in use, hitch is overloaded. Jack is still used on deformed structure and bends due to unintended use.	3	Company Standards, Supplier Quality Standards	4	96
	Square Tubing Breaks	Unit Falls to Ground	10	Jack is structurally compromised due to being bent from the V-support structure being bent. Square tubing is bent enough to no longer be able to support header weight but jack is operational.	1	Company Standards, Supplier Quality Standards	6	60
	Weld Failure	User can not use V-support	8	V-support is significantly bent in field use when unit is being used in the field.	2	Company Standards, Supplier Quality Standards, AWS Standards	4	64

		Jack Slightly Bent (operator is still able to lift and lower jack)	6	V-support structure is improperly welded to driveline guards. Welds being to separate under normal loading conditions.	2	Company Standards, Supplier Quality Standards, AWS Standards	7	84
		Jack Significantly Bent (operator is no longer able to lift and lower jack)	8	V-support structure is improperly welded to driveline guards. Welds separate under normal loading conditions.	2	Company Standards, Supplier Quality Standards, AWS Standards	5	80
		Unit Falls to Ground	10	V-support structure is improperly welded to driveline guards. V-support shears off driveline guards under normal loading conditions.	1	Company Standards, Supplier Quality Standards, AWS Standards	8	80

Some of the failure methods have a risk priority number close to, but not exceeding the 100 point limit. Some of the critical failure modes may require additional testing or verification from internal MacDon databases, to confirm that the chosen failure rates are realistic. By this preliminary FMEA, the team can conclude that the design meets European Machinery Directive 2006/42/EC.

10.0 Costing

Creating a cost breakdown for the proposed design is important, as the customer defined a total cost of $100 CAD. This cost was expanded from the original cost of $50 CAD once it was known that the proposed design would include the current jack, which has a cost of $50. For this costing process, material and processes were priced as if the component was being purchased by a person looking for a single unit purchase of the V-support structure. Where possible, the team used larger than necessary purchases, to reduce the price of materials. For example, ordering a full 8 ft x 4 ft piece of sheet metal, rather than the small amount necessary to make the part. The cost of these larges pieces was broken down to the cost per unit length or cost per unit area and then used for the calculation of the part costs. The costs of the materials used in the production of the V-support are summarized in TABLE XXXII.

TABLE XXXII: MATERIAL COSTS FOR THE MANUFACTURE OF THE V-SUPPORT

Material	Cost
A513 1-1/2 x 1-1/2 x 11Ga	$2.43/304.8mm* [22]
7 Ga A1011 CQ	$273.74/8 ft x 4 ft sheet [28]
12 Ga A1011 CQ	$174.33/8 ft x4 ft sheet [28]
1020 DOM 2 x 0.25 Round Tube	$15.56/304.8mm* [29]

*Note: Exchange rate taken at 1CAD = 0.74USD [30]

The process cost for laser cutting was found online, while the remaining costs were estimated based on known shop rates. The process costs are shown in TABLE XXXIII.

TABLE XXXIII: PROCESS COSTS FOR THE MANUFACTURE OF THE V-SUPPORT

Process	Cost
Laser Cutting	$0.009843/mm* [30]
Weld Preparation	$70/hr
Welding	$70/hr
Painting	$60/hr
Machining	$70/hr

*Note: Exchange rate taken at 1CAD = 0.74USD [29]

With all of the known costs, the cost per part was calculated. The v-support has an extra weld preparation step for it, as each side of the V-support will require grinding to create the proper initial surface to weld the two parts together. The V-support fabrication step includes welding the two V-support sides together and welding the bottom cap onto the V-supports once welded together. The final fabrication step consists of welding the bolster plate and the jack mount tube to the V-support and painting the final assembly. The costs per part and costs for each fabrication step are shown in TABLE XXXIV. The cost breakdown does not include the cost of the jack mount tube, as this part is a purchased part by MacDon that is used in the original design and this design. Additionally, this cost breakdown does not factor in the associated costs in regards to extending the driveline guards. These associated costs are a production change that MacDon would implement through their purchasing department.

TABLE XXXIV: PART AND FABRICATION COST BREAKDOWN AND TOTAL PROPOSED DESIGN COST

Description	QTY	Material/Process	Unit Cost	Unit	Cost ($)
V-support side	2	A513 1-1/2 x 1-1/2 x 14Ga	0.0083 $/mm	495.3 mm	4.11
		Laser Cutting	0.007874 $/mm	436.41 mm	3.44
		Weld Preparation	$70/hr	0.1 hr	7.00
			Total Part Cost:		14.55
Boslter Plate	1	7 Ga A1011 CQ	0.000092 $/sq. mm	6296.26 sq. mm	0.58
		Laser Cutting	0.007874 $/mm	328.21 mm	2.58
			Total Part Cost:		3.16
Jack Mount	1	A513 2x0.25 Round Tube	0.0510 $/mm	60 mm	3.06
		Maching	$70/hr	0.25 hr	17.50
			Total Part Cost:		20.56
Bottom Plate	1	12 Ga A1011 CQ	0.000059 $/sq. mm	1640.49 sq. mm	0.10
		Laser Cutting	0.007874 $/mm	163.32 mm	1.29
			Total Part Cost:		1.39
V-support Fabrication	1	Welding	$70/hr	0.25 hr	17.50
			Total Fabrication Cost:		17.50
Final Fabrication	1	Welding	$70/hr	0.25 hr	17.50
			Total Fabrication Cost:		17.50
				Total Cost	89.21

The final cost of this part, not installed on the swing tongue driveline guards is $89.21 CAD, with the addition of the $50 CAD jack. The final proposed design comes to $139.21 CAD. The team acknowledges that the final cost of the proposed design is higher than the goal cost, but is aware that MacDon's true part cost will be significantly lower through volume pricing and a stronger negotiating position with vendors.

11.0 Revisiting Target Specifications

This section details how the V-support meets the target specifications laid out by the team at the beginning of the design project. The marginal and ideal values for each target specification were created through discussion internally and later approved by the client. If the design met the marginal technical specifications, the design would be considered a success. TABLE XXXV below outlines how the final design meets the target specifications.

TABLE XXXV: REVISED METRICS AND DESIGN VALUE COMPARISSONS

#	Metrics	Units	Marginal Values	Ideal Values	Design Values
1	Total mass	kg	≤9.07	<6.80	3.85*[1]
2	Production cost/unit	CAD $	100	<100	140*[2]
3	NA & EU standards	Pass/Fail	Pass	Pass	Pass
4	Deflection/deformation	mm	≤3	≤1	0.96
5	FEA analysis FOS	-	2	>2.5	3.3
6	Critical load	N	≥6672	>16700	22020
7	Impact testing	Pass/Fail	Pass	Pass	Pass
8	All hitch options	list	Pass	Pass	Pass
9	# of usage per/year	#	365	730	700
10	Time to disassemble and store	s	<60	<30	30
11	Tools required for maintenance	list	Pass	Pass	Pass
12	Clearance from driveline	cm	6-8	>8	9
13	Clearance from decal location	mm	10-14	>14	>14
14	All weather conditions	list	Pass	Pass	Pass
15	All ground conditions	list	Pass	Pass	Pass
16	Force required to operate (i.e. turn the handle bar, etc.)	N	≤225	180	225
17	# of operator required to install	#	1-2	1	1
18	Time required to setup and install stand	s	<60	<30	30

*[1]: The total mass refers to the mass of the V-support structure. Once the support structure is fabricated to the driveline guard bars, the operator will only have to move the current jack. The current jack weighs ≈ 7.26 kgs, which is within the marginal range.

*[2]: The production cost to manufacture one support structure is $140. If the design is implemented, the structure will be mass-produced, drastically reducing the manufacturing price. With mass production the price will be reduced below the specified limit of $100.

The V-support design meets all of the needs and target specifications defined by the client. This review of the target specifications verifies that the V-support design is a practical solution for the problem initially defined by the client. Note, that the team recognizes that MacDon's production costs will be significantly less than the prices used for the cost breakdown, thereby reducing the part costs.

12.0 Summary

The purpose of this project was to create and design a universal support device for MacDon's pull-type rotary header. MacDon requested that the new device must eliminate the current issues of the hitch stands, which include driveline clearance, hitch instability, time consuming setup/ removal, and difficulty connecting to the tractor. Through problem definition, concept generation and detailed design, Team 7 was able to generate a final design that addresses and solves these problems.

This report successfully covered all aspects of the design process. First, relevant background information was provided on our client and the piece of harvesting equipment the project focuses on. The report then outlined the problem statement, objectives, target specifications and constraints and limitations. From here, the report summarized the research completed to initiate the concept generation and selection phase. With a final concept generated, the report moved onto the detailed design, where the evolution of the design, analytical, numerical finite element analyses were performed to ensure that the design is capable of supporting the weight of the hitch. The report finishes with a cost breakdown and an overview of how the final design meets the client's needs.

In summary, the V-support structure was successfully designed to replace the existing trailer jack stand and built-in stand on the drawbar and 2-point hitch style headers, respectively. The final V-support design is a product of five earlier versions, where each version was refined through a combination of client feedback and necessary design improvements. This final V-support design weighs 8.4 [lbs] and is constructed from 1.50 [in] square ASTM A513 Grade B tubing. To ensure no interference with tractor tires, the driveline guard bars were extended 12 [in] forward, with the V-support being directly mounted and welded to the front most horizontal portion of the extended driveline guards. The V-support was designed to also have the same mounting height for the trailer jack, allowing the existing trailer jack model to be used. Analytical verification confirmed this design supports the 1500 [lbs] weight of the swing tongue while maintaining a factor of safety of 3.30, deeming this fully functional design safe for use.

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A.1.0 Discbine Research

New Holland Discbine product line is a direct competitor to the MacDon R1 product line. The Discbine product series is offered in the same widths and similar overall options. In terms of attaching the header to the tractor, New Holland offers a 2 point hitch stand option or a drawbar hitch option. The 2-point hitch option comes with a stand that mounts at the rear of the lower swivel gearbox as seen in Figure 1. For 2-point hitches, the option of a fixed stand between the lower swivel gearbox and the ground is best, as the hitch arms on the tractor are able to raise and lower to enable the tractor to attach the header no matter the ground height.

Figure 1: New Holland 313 Discbine with 2-point hitch and stand [1].

Drawbar hitch options are a fixed height above ground and consist of a pin that sits inside a channel on a tractor. Due to this, height adjustment must be built into the stand. For this option, New Holland was able to utilize the driveline guard bars so that their design does not have to worry about the issues caused by the rotational nature of the lower swivel gearbox. An example of a 313 Discbine with a drawbar hitch style and the jack mounting location off the driveline guards can be seen in Figure 2

Figure 2: New Holland Discbine with driveline guard jack mounting visible [2].

. Figure 3 shows a close up of the jack mounting bracket when the header is attached to a tractor.

Figure 3: Close-up of jack mounting bracket [3].

The pin can be removed to lower the drop down jack mount. Once the pin is reinstalled, the jack can be mounted on the header and used to lift the hitch of the header off of the tractor.

A.2.0. Initial 40 Concept Generation Results

Appendix A.2. contains sketches with brief descriptions of all of the concepts generated by the team through the concept generation phase.

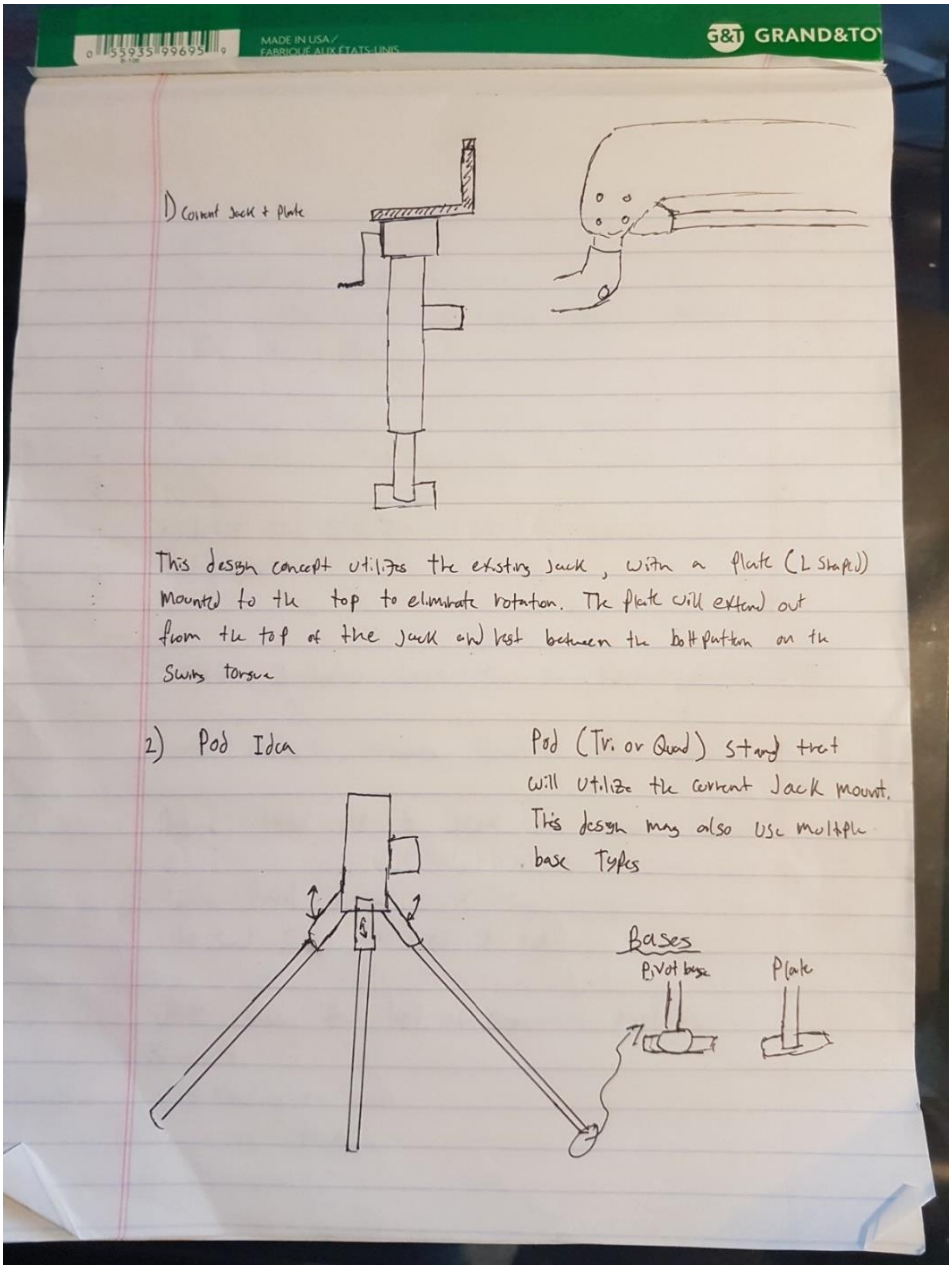


FIGURE 4: CONCEPTS 1-2.

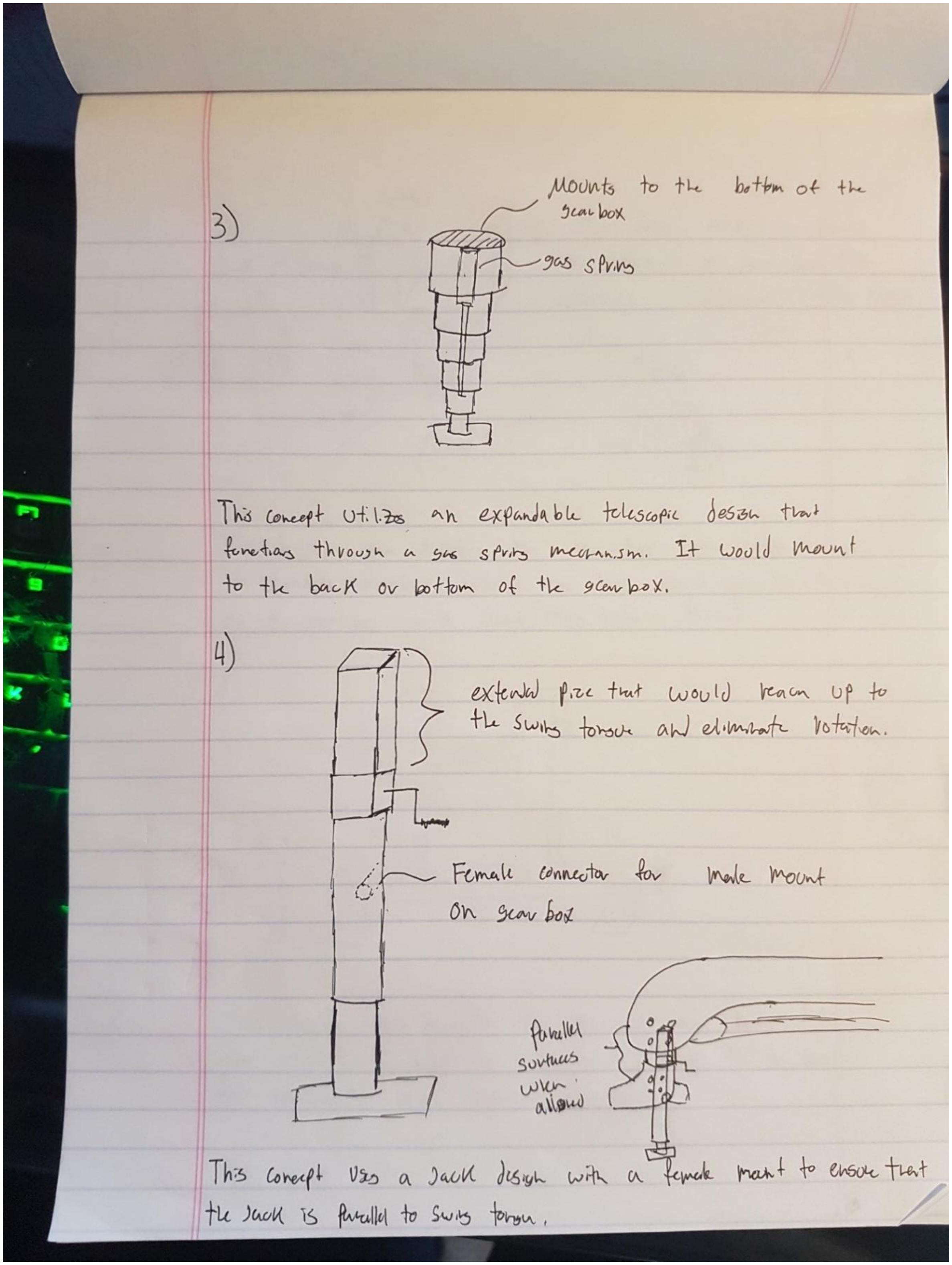


Figure 5: Concepts 3-4.

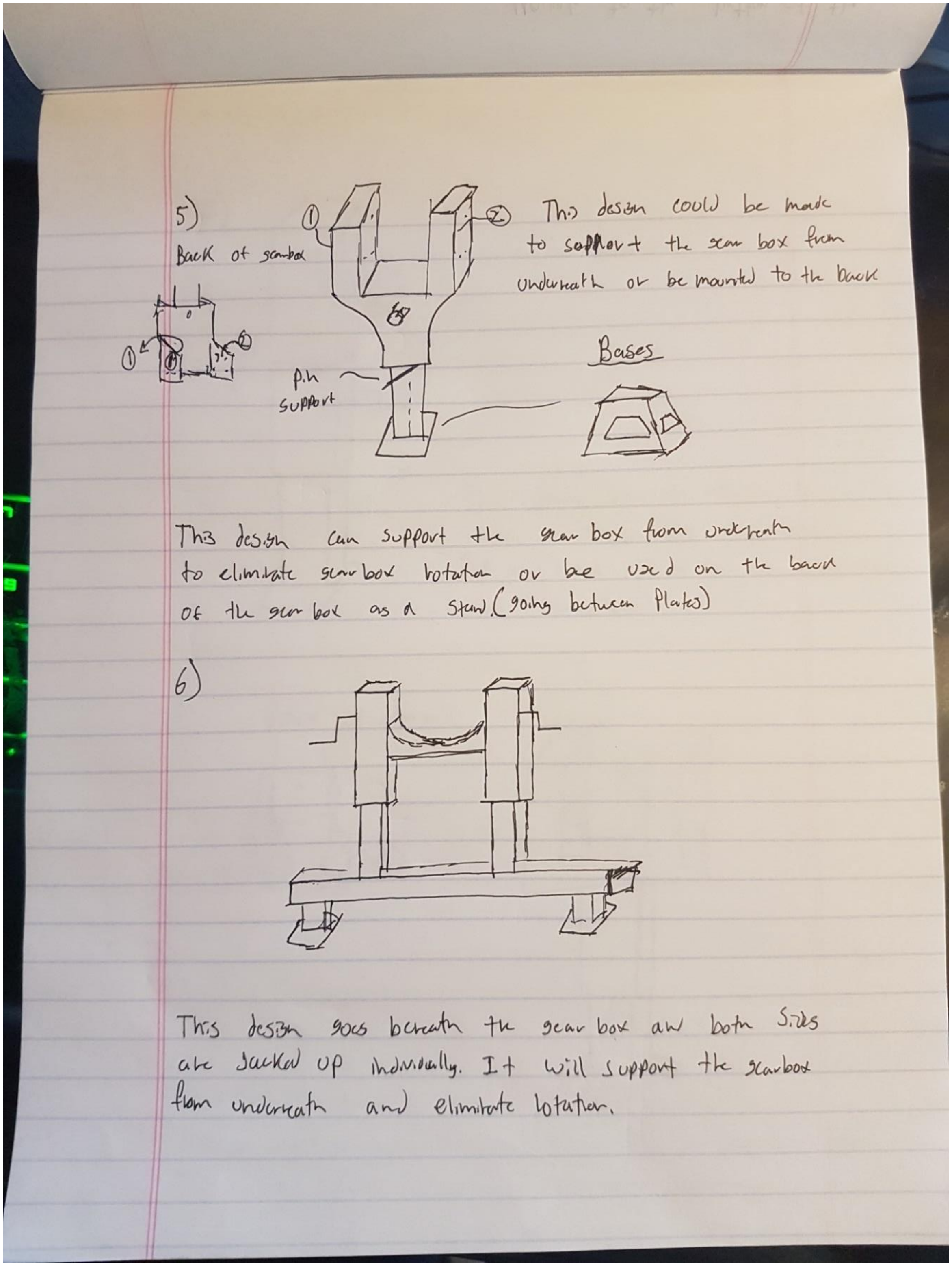


Figure 6: Concepts 5-6.

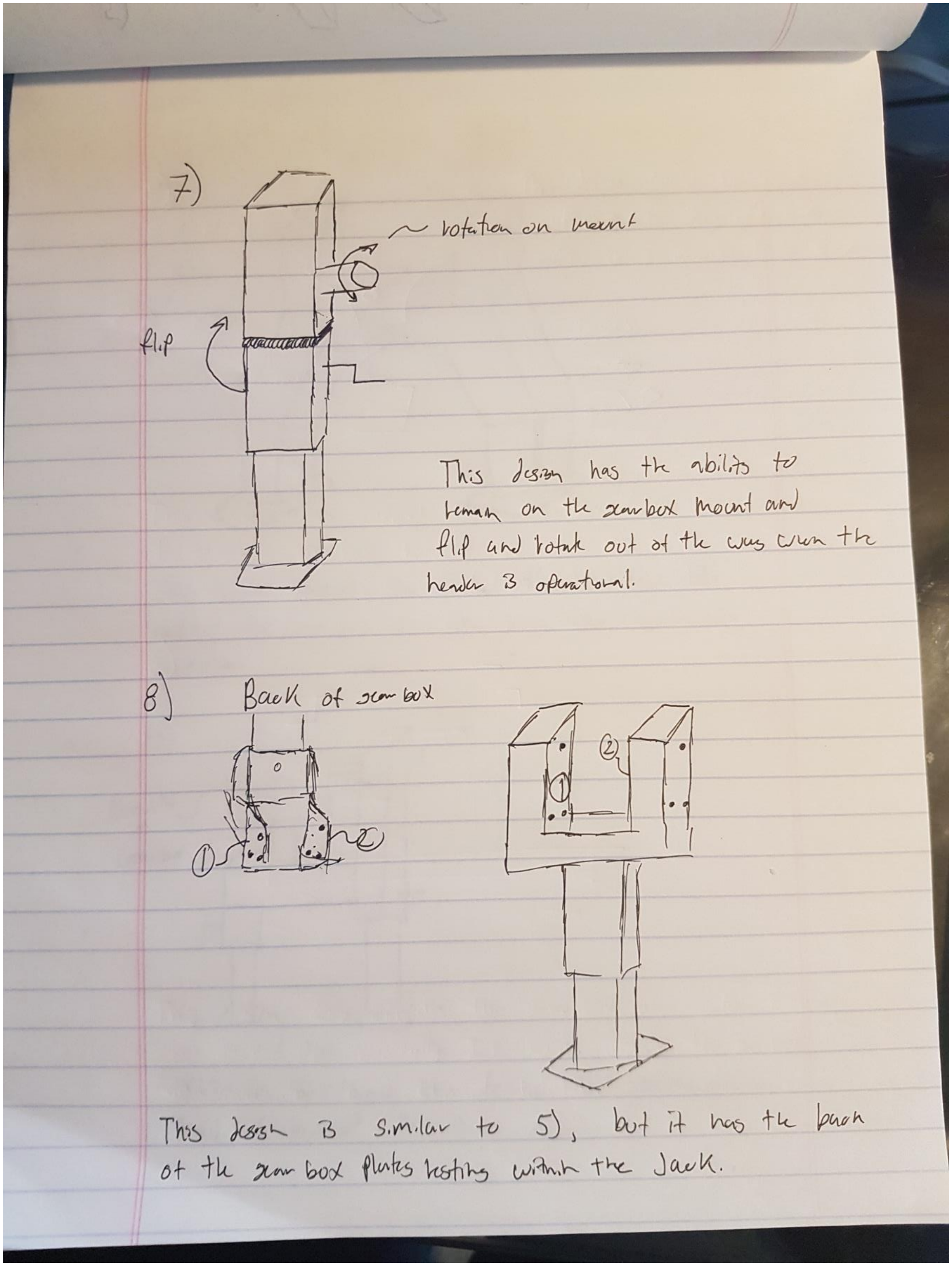


Figure 7: Concepts 7-8.

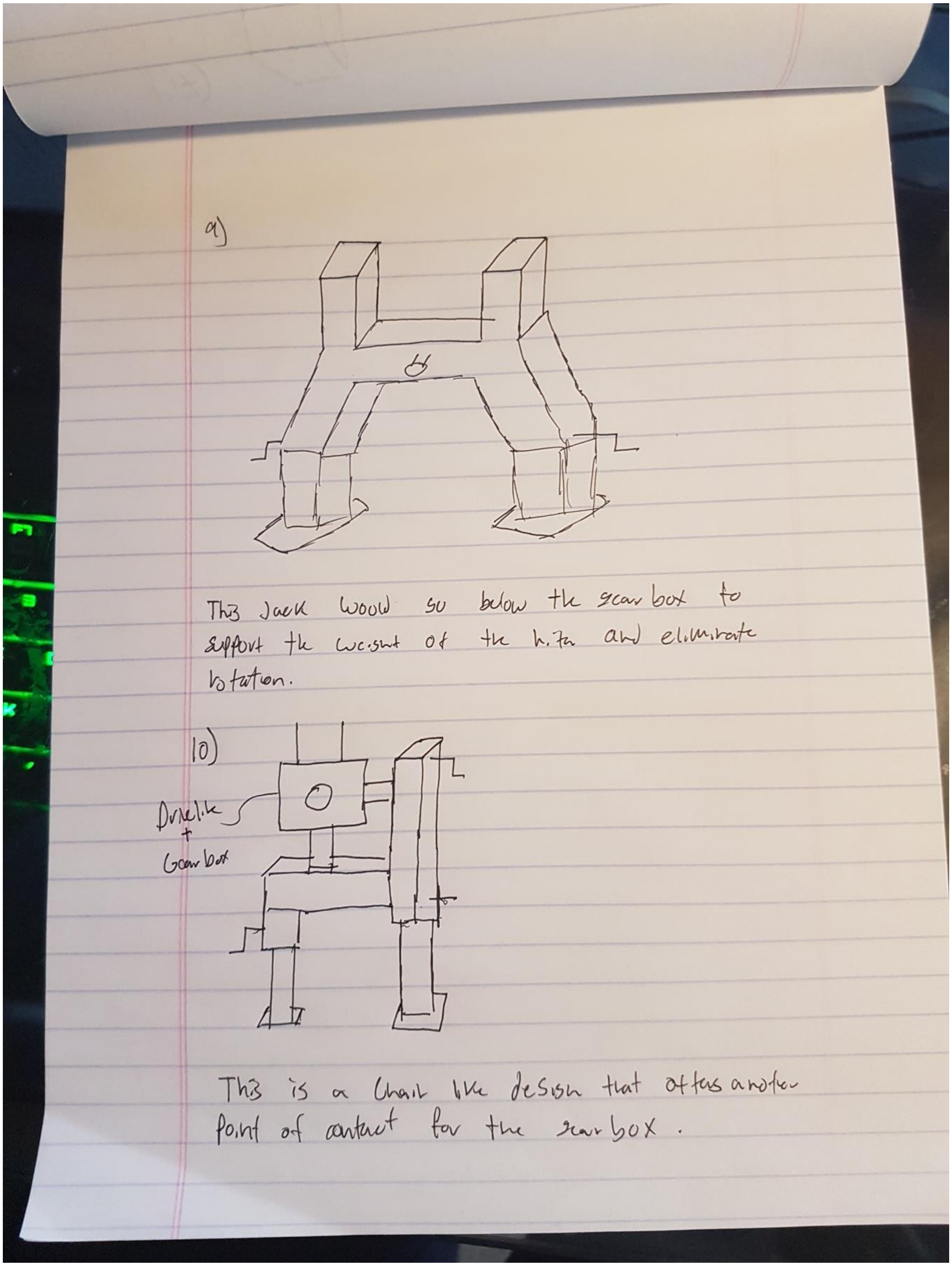


Figure 8: Concepts 9-10.

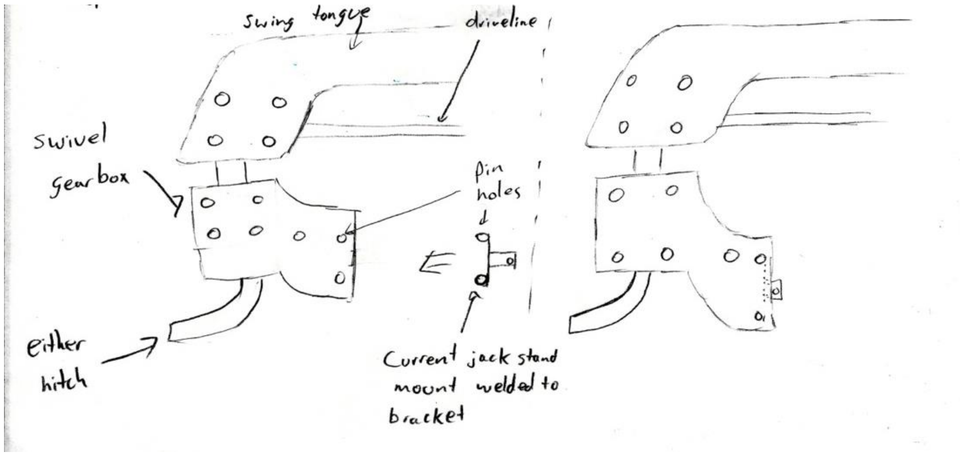


Figure 9: Concept 11 - Rear Mount.

Concept 11 focuses on moving the mount of the jack from the left hand side of the lower half of the swivel gearbox to the rear of the lower swivel gearbox. To relocate the jack mount to this location, a plate with holes or bushings that align with the current holes in the lower swivel gearbox weldment is required. Pins would then be used to hold the jack mount plate in position.

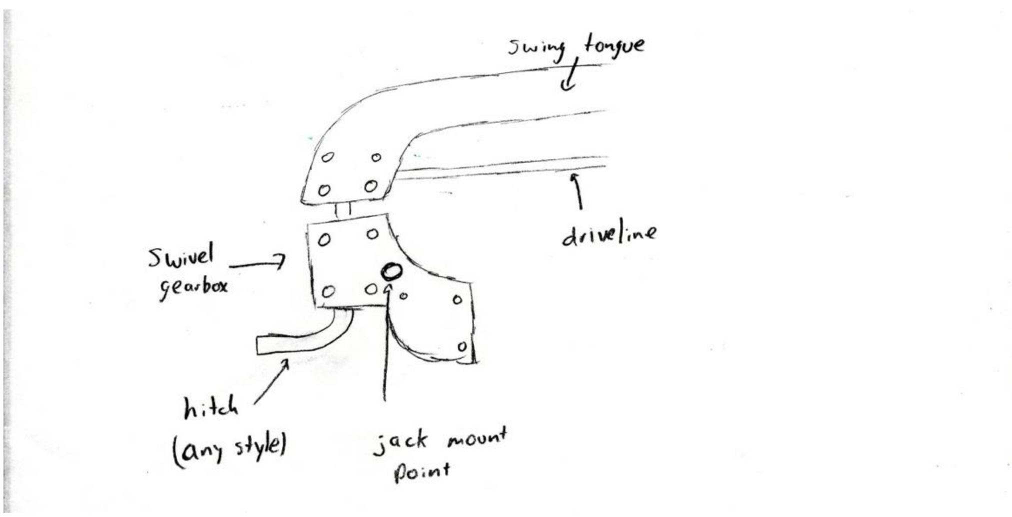


Figure 10: Concept 12 - Keep Everything the Same.

Concept 12's main idea is to keep all of the components the same. This concept would be the worst case scenario, as the client is not happy with their current design. Concept 12.1 is a sub-concept of Concept 12. In this concept, a square tube trailer jack would be used instead of the current jack, which uses a round tube. This idea is an improvement over the current concept as square tube jacks are up to 70% stronger than round tube jacks with regards to side load and support strength.

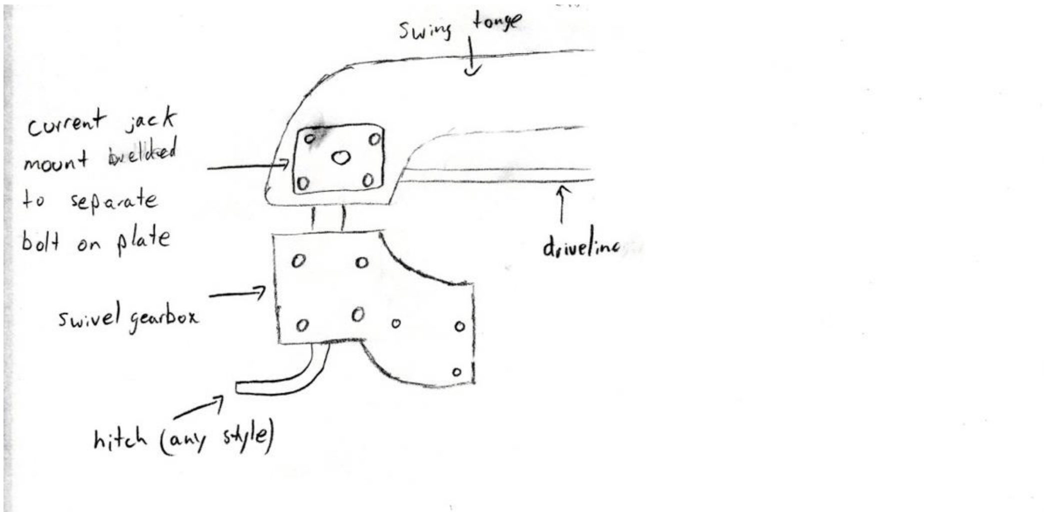


Figure 11: Concept 13 - Top Mount.

Concept 13's goal is to move the jack from the current mounting position on the lower swivel gearbox weldment to the upper portion of the swivel gearbox. This was done to remove the jack mount from the rotation of the lower swivel gearbox, which is the cause of the instability in the trailer jack.

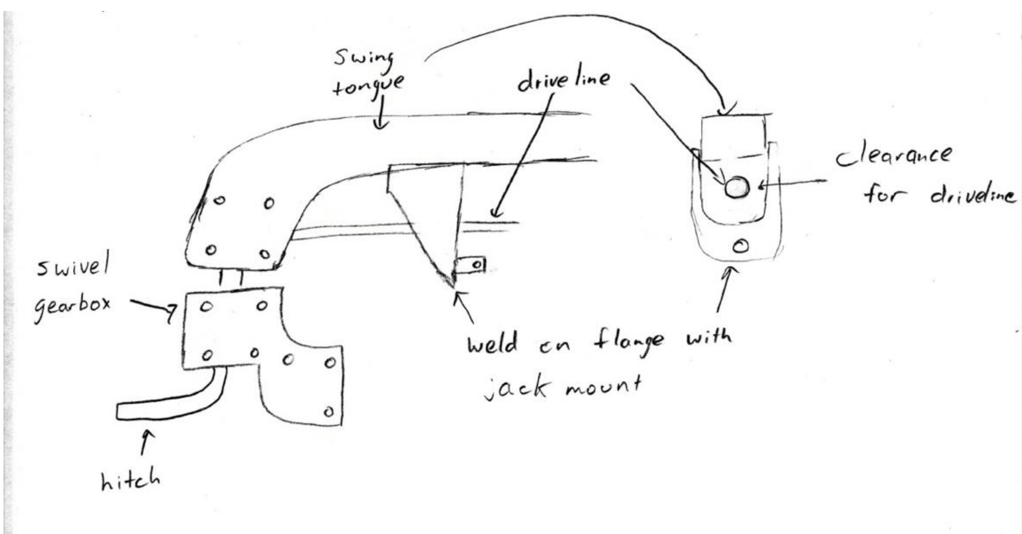


Figure 12: Concept 14 - U-Mount.

Concept 14 allows for the trailer jack to be moved completely off of the swivel gearbox at the front of the swing tongue. Relocation of the jack mount completely removes the rotation of the swivel gearbox, which is the source of instability in current design. This concept features a weld on U-shaped bracket, which is appropriately sized to allow for the driveline to pass between the bottom of the swing tongue and the top of the bracket. Appropriately sizing the gap allows for installation and removal of the driveline when necessary.

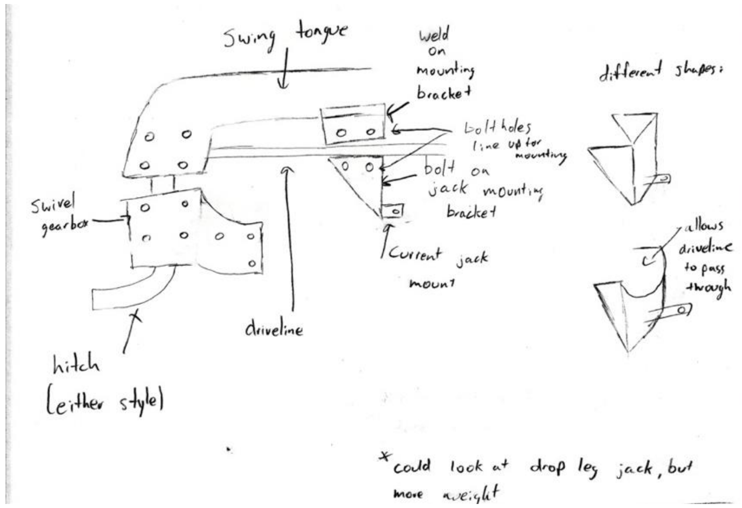


Figure 13: Concept 14.1 – U-Mount Modified.

An improvement of Concept 14 is concept 14.1 which improves the overall idea in terms of serviceability and manufacturability. This concept features two tabs that are welded on to the swing tongue and have bolt holes in the lower portion of each tab. A bracket which has the current jack mount welded on to it, can then bolted into position. The bolt on nature of the bracket improves serviceability and part replacement over Concept 14. A drop leg trailer jack could be investigated for this design as it would reduce the amount of time required to lift the hitch of the header, but at the expense of greater weight.

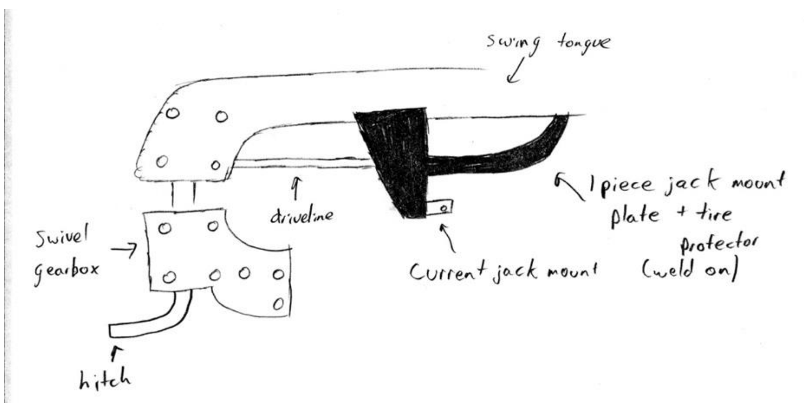


Figure 14: Concept 14.2 – U-Mount and Tire Protector.

Concept 14.2 is another modified version of Concept 14. Due to potential space issues along the header's swing tongue, the weld on bracket is combined with the current tire protector. Combining the tire protector and the jack mount bracket allow for both components to be placed in an overall smaller package. This concept will also help to reduce the overall cost of the components, as two components are being combined into one. With some minor modifications, this concept could be changed to feature a removable jack mount and tire protector.

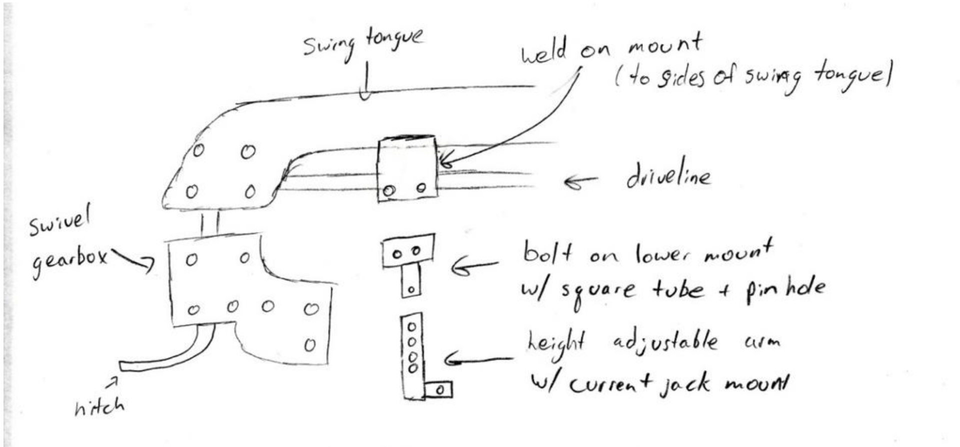


Figure 15: Concept 14.3 - Adjustable Height U-Mount.

Concept 14.3 allows for adjustability when selecting the trailer jack mount height, by using an adjustable height system. The bolt on triangular mounting bracket from Concept 14.1 is replaced with a bolt on mount with a short tube which has a hole drilled through it. The jack mount is welded to a longer tube, which has multiple holes cut through the tube. Both the short tube mount and the longer tube would be sized appropriately to allow for easy movement of the long tube inside the shorter tube. To operate the jack mount, the operator would select the height position of the jack mount by aligning the appropriate hole in the adjustment tube and then inserting a pin. The operator can then attach the jack at their selected height.

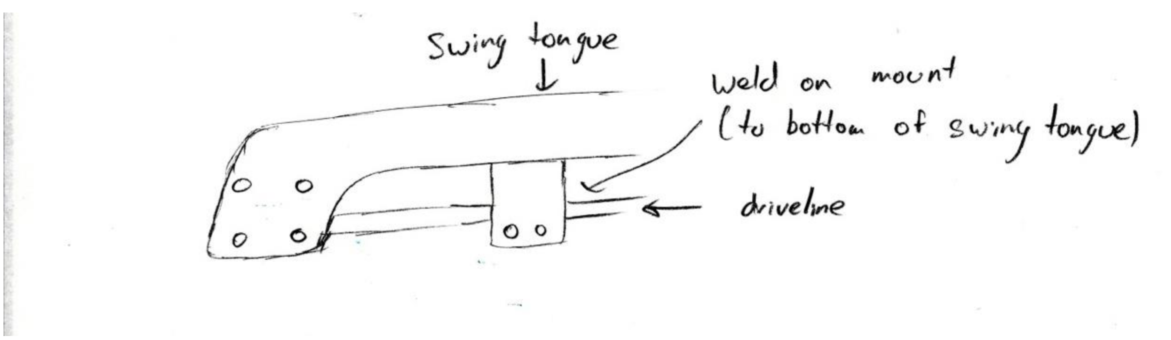


Figure 16: Concept 14.4 - Adjustable Height U-Mount Modified.

Concept 14.4 moves the weld on tabs from the side of the swing tongue to the bottom of the swing tongue. This change means that MacDon does not have to move or resize safety and maintenance decals that are placed on the left hand side of the swing tongue. All other concept features are the same as seen in Concept 14.3.

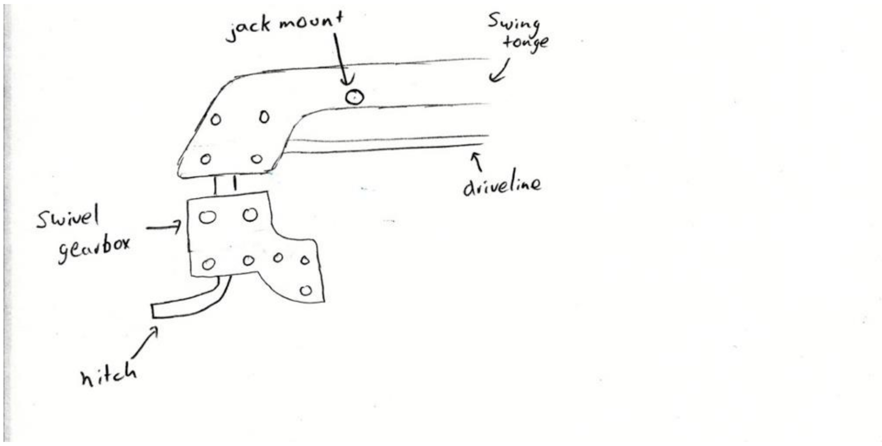


Figure 17: Concept 15 - Swing Tongue Mount.

Concept 15 relocates the trailer jack mount from the left hand side of the lower swivel gearbox weldment to the left hand side of the swing tongue. This was done to remove the rotational movement of the swivel gearbox from the mounting system of the trailer jack. A longer jack would have to be sourced to allow for the increased span that the jack would have to cover.

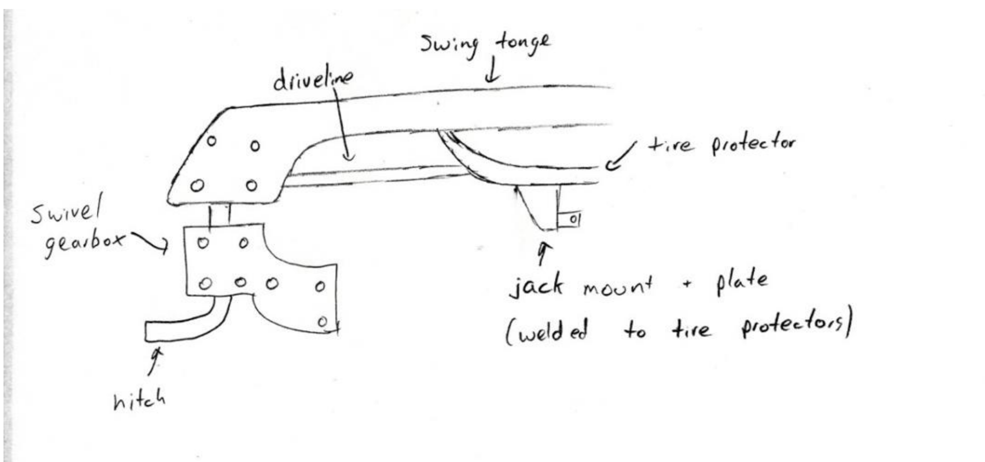


Figure 18: Concept 16 - Tire Protector Mount.

Concept 16 features a triangular bracket that would be welded on to the tire protectors. This location allows the driveline to pass between the bottom of the swing tongue and the top of the trailer jack mounting bracket.

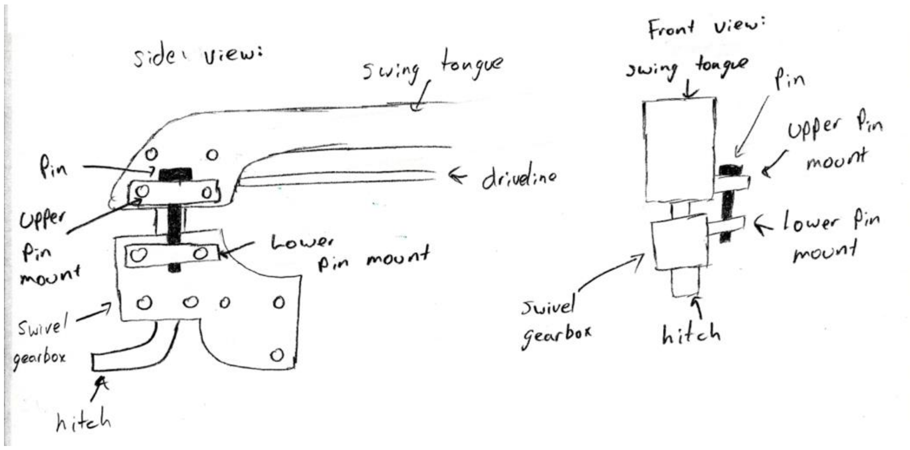


Figure 19: Concept 17 - Locking Pin.

Concept 17 has a pin that would be installed between the upper and lower swivel gearbox sections. This pin would be used to lock the rotation of the swivel gearbox. While not shown in the sketch, the current trailer jack mount would be used in the current position. Use of the same jack mount and jack allows MacDon to continue to stock the same service parts. The upper and lower pin mounts would use bolt holes that are currently in the design of the swing tongue and lower swivel gearbox weldment. These holes are used to hold the swivel gearbox in place.

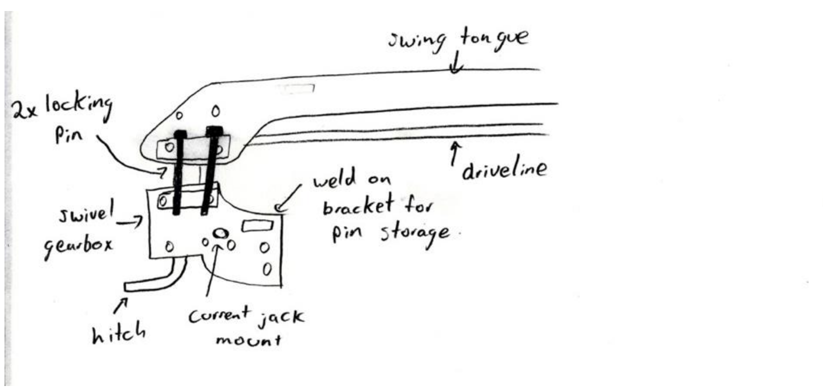


Figure 20: Concept 17.1 - Two Locking Pins.

Concept 17.1 is a slight modification of Concept 17. Instead of using one pin for locking the rotation of the swivel gearbox, two pins would be used. While the pins are shown in the sketch as separate, the pins could potentially be mounted together. Mounting the pins together creates a more efficient mounting system, while also creating a larger part that would be harder to lose for the end user.

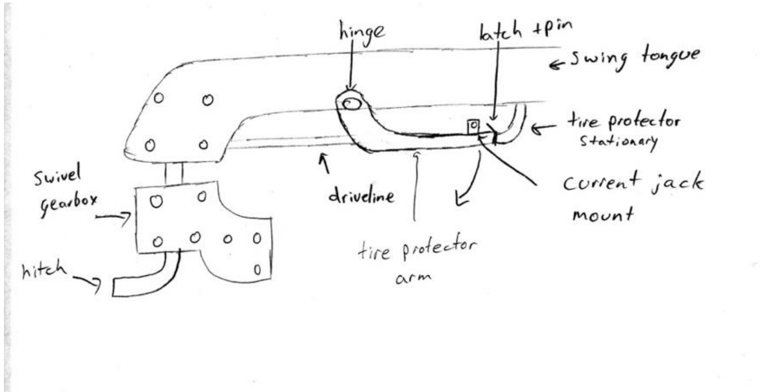


Figure 21: Concept 18 - Rotating Tire Protector.

Concept 18 redesigns the tire protectors that are installed on the swing tongue. A hinge would be placed on one end of the left hand side tire protector and a latch on the other end. The hinge would allow the tire protector to swing down vertically when unlatched. When the tire protector is swung downwards, the jack mount would be at the same height from the ground that the current jack mount is. For this design to function safely, the swing down arm would have to be designed such that the hinge would lock open when the weight of the swing tongue is placed on the jack.

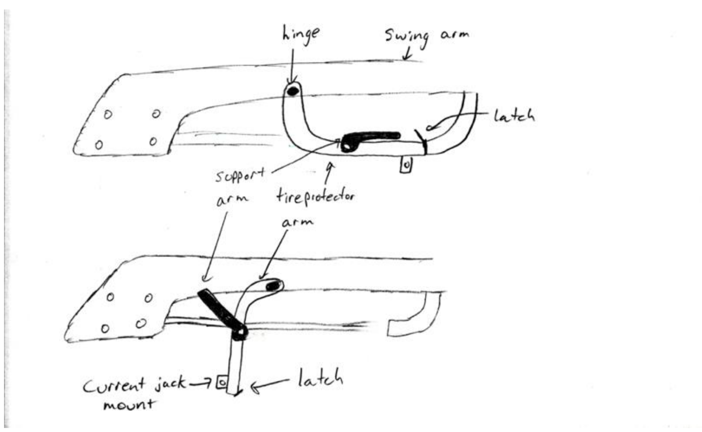


Figure 22: Concept 18.1 - Rotating Tire Protector with Support Arm.

Concept 18.1 is an improvement over Concept 18. This concept adds a support arm to the swinging tire protector. The support arm would be used to reduce the amount of stress on the hinge, while adding an additional angle of support to the jack. The improved design can be seen in the raised and lowered position. Concept 18.2 is the same as Concept 18.1, but instead of the tire protector swinging down vertically, the tire protector would swing outwards at an angle. While increasing the design complexity and therefore cost, the additional support would be worthwhile to have. Concepts 18.3 and 18.4 are the same as Concepts 18.1 and 18.2, but instead of having one swing down arm, there would be two swing down arms, one on each side of the swing tongue.

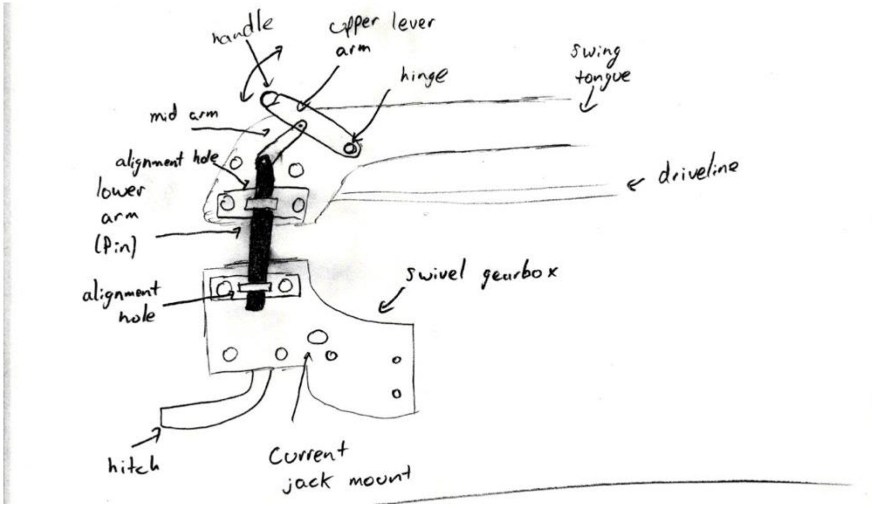


Figure 23: Concept 19 - Hole Punch.

Concept 19 features a mechanism used to move a locking pin up and down. The general concept came from the up and down motion of a paper hole puncher. The pin would be used to lock the rotation of the gearbox, similar to Concept 17. The mechanism itself would most likely be a Scott Russell linkage. Scott Russell linkages allow forwards and backwards linear motion to become up and down linear motion.

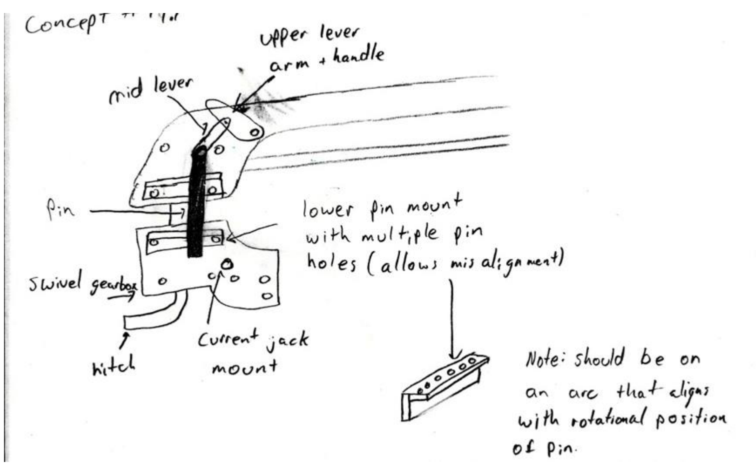


Figure 24: Concept 19.1 - Hole Punch Modified.

Concept 19.1 is a slight modification of Concept 19. This concept is a slight modification because the pin bracket in the lower swivel gearbox would have several holes. These holes would either be placed in an arc or a straight line. The holes in the lower pin bracket allow for rotational misalignment between the upper and lower swivel gearbox.

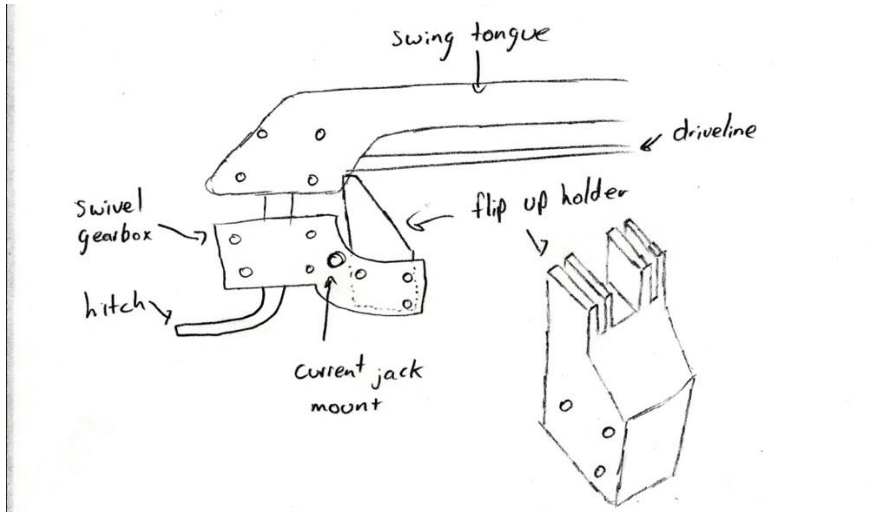


Figure 25: Concept 20 - Flip-Up Fork.

Concept 20 has a fork-like design at the top of the flip up bracket. The tines of the forks would be spaced far enough apart to allow the outer edges of the swing tongue to fit between the tines. Having the swing tongue fitting between the tines would create a mechanical lock that prevents rotation of the swivel gearbox.

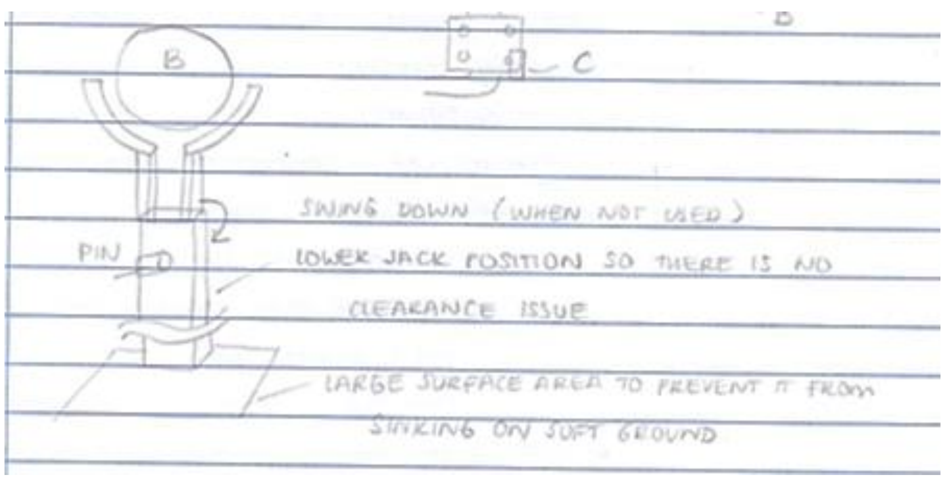


Figure 26: Concept 21.

21. The idea is to have the jack completely underneath the drawbar. The jack stays the same while extra support is added. Swing arms attached on the side of the jack to hold the driveline to limit the gearbox rotation. This will improve stability, but would probably require more time for setup and removal.

Figure 27: Concept 22.

22. Similar to 21. Holes are added to the side-wall of the jack and instead of having swing arms, the support will slide up/down. The height of the arms is adjusted by relocating the spring-loaded pin on whichever holes' location desired.

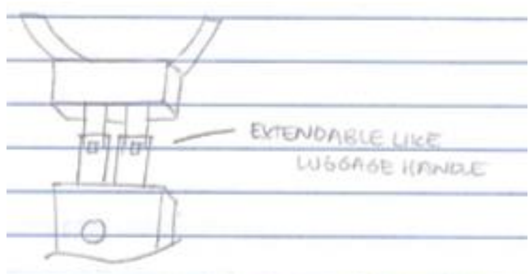


Figure 28: Concept 23.

23. Similar to 21. The support portion above the jack is extendable. The mechanism of the extendable portion will follow that similar to a luggage handle. This will hopefully improve time on during setup and removal.

Figure 29: Concept 24.

24. In this concept, the jack is still pinned to existing stub on the gearbox. But the base is somewhat like a tripod. Since the bottom can extend/fold, storing this jack would be easy. The whole jack would be compact and would only require little setup time and removal as we no longer have to turn the jack handle to crank it up. Having a wide base would also provide a better structural stability.

Figure 30: Concept 25.

25. This concept provides better stability since extra legs are added to both sides of the stand. This is still pinned to the gearbox, but does not require the jack handle to crank it up as the stand will be tall enough.

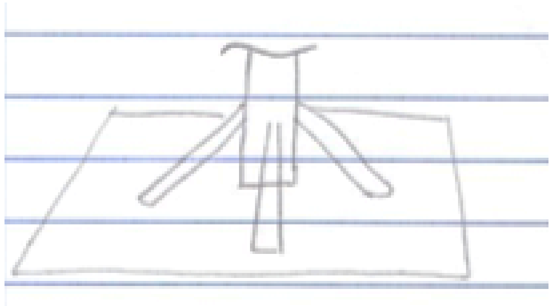


Figure 31: Concept 26.

26. We can use the 2-point hitch stand since the drawbar also has the same 3 holes at the back of the gearbox. It will be more time efficient during setup and removal since the stand is built-in. However, extra legs could be added to the stand for better stability.

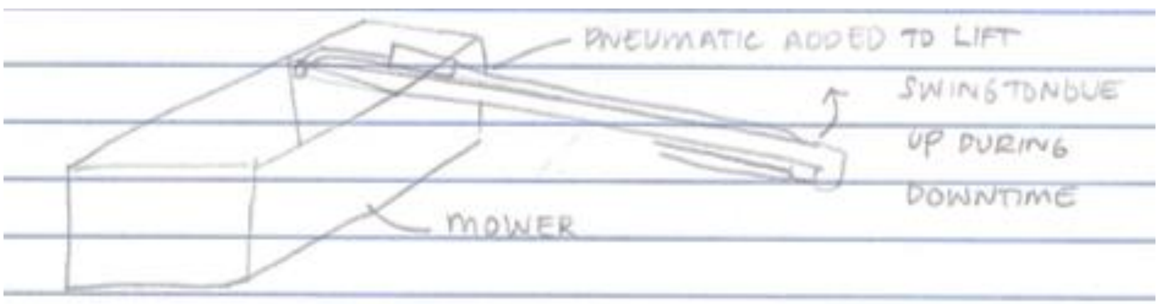


Figure 32: Concept 27.

27. This concept uses a pneumatic system that could be added to the back of the header by the mower. The idea is to have the pneumatic engaged as the hitch gets detached, so it will lift up the whole swing tongue.

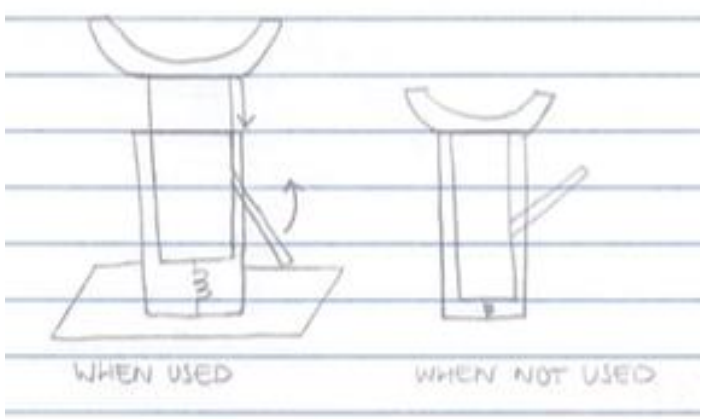


Figure 33:Concept 28.

28. In this concept, we have a spring loaded support arm and a lever to compress the spring for when the stand is not used. The support arm will go to the bottom of the driveline.

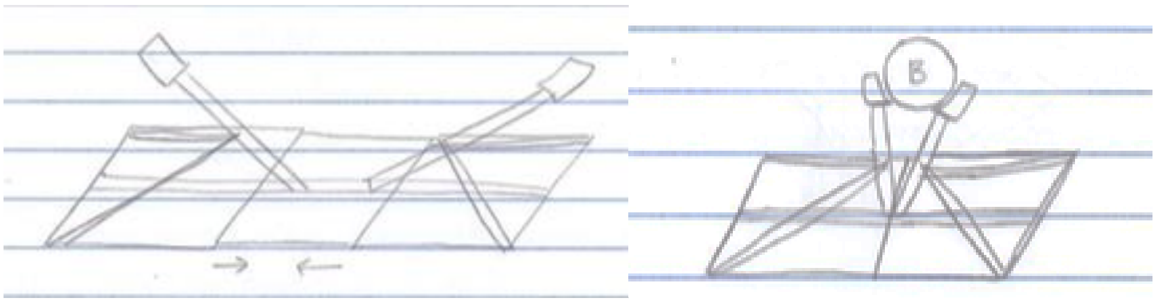


Figure 34: Concept 29.

29. This concept resembles that of a keyboard base. When not used, the stand will be folded down and compact. When used, the base will get smaller as the support arms get closer to each other and get attached to the driveline.

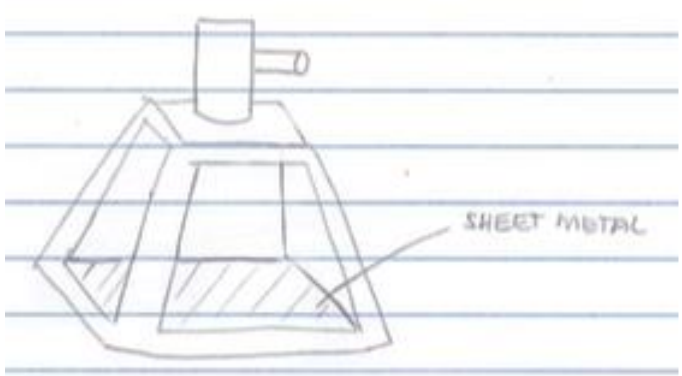


Figure 35: Concept 30.

30. The idea is to have a stand tall enough to give the hitch a lift. The stand will be connected by pin support to the gearbox. However, more structural stability can be achieved since this concept has a wider pyramid base.

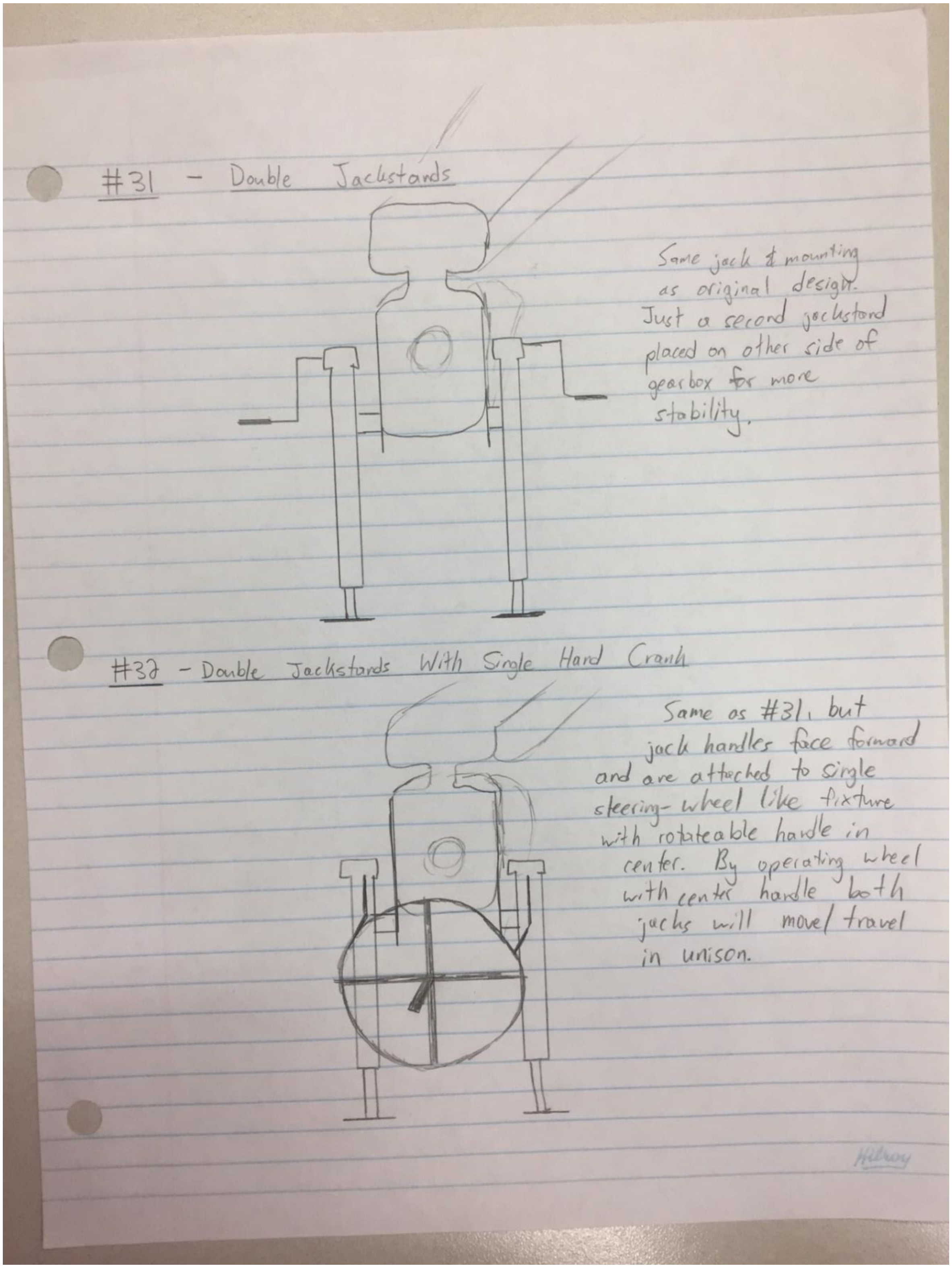


Figure 36: Concepts 31-32.

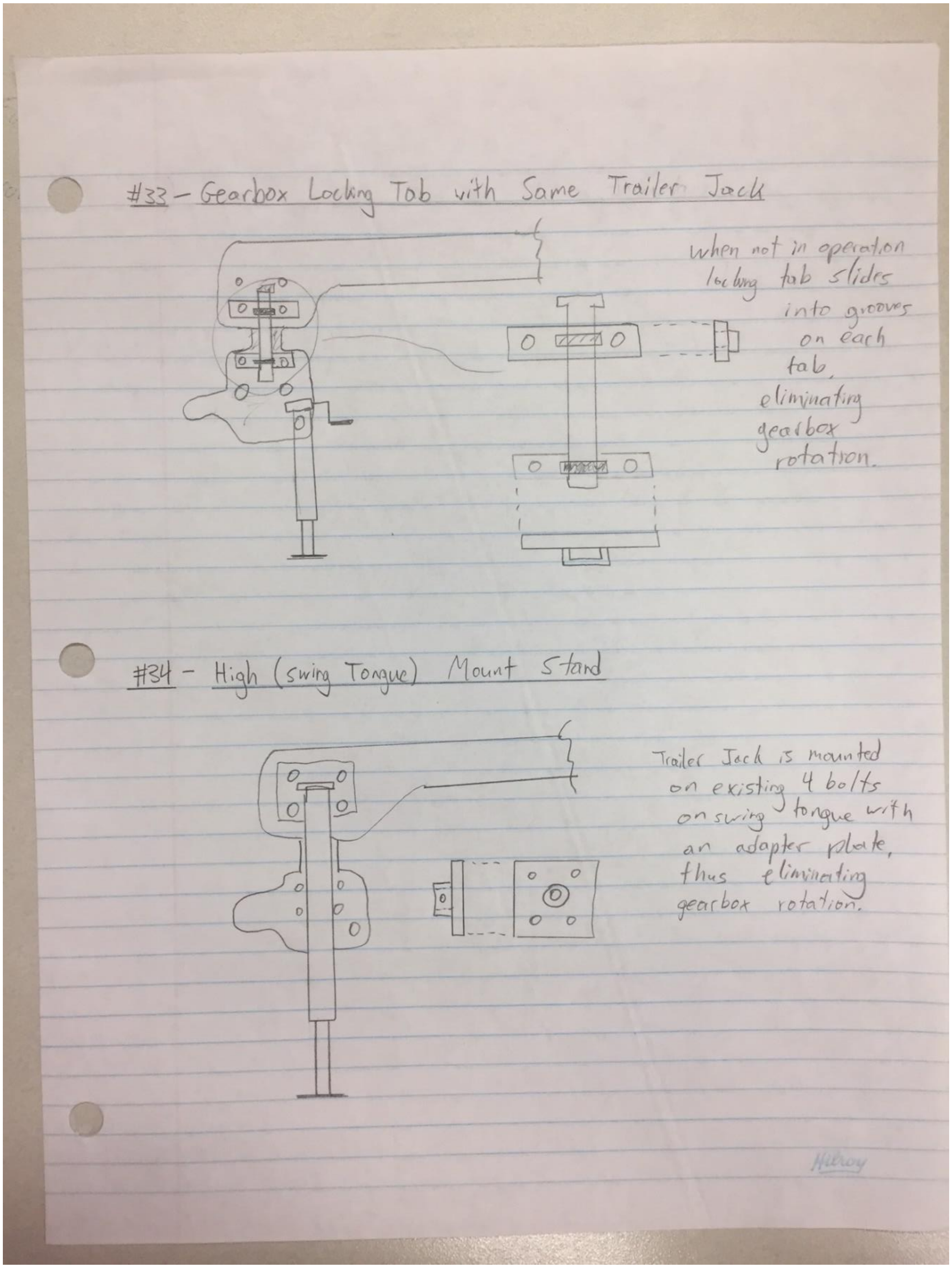


Figure 37: Concepts 33-34.

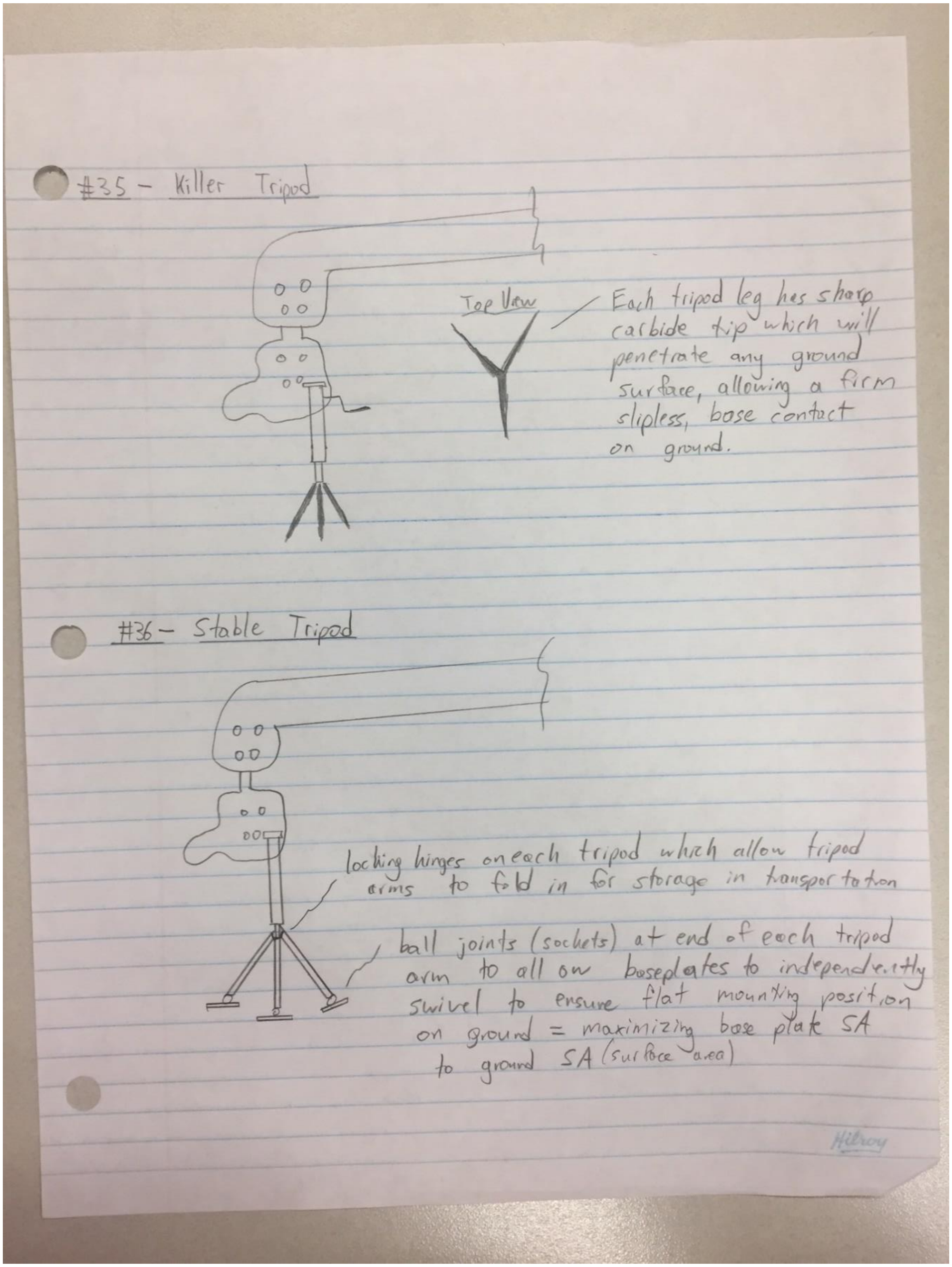
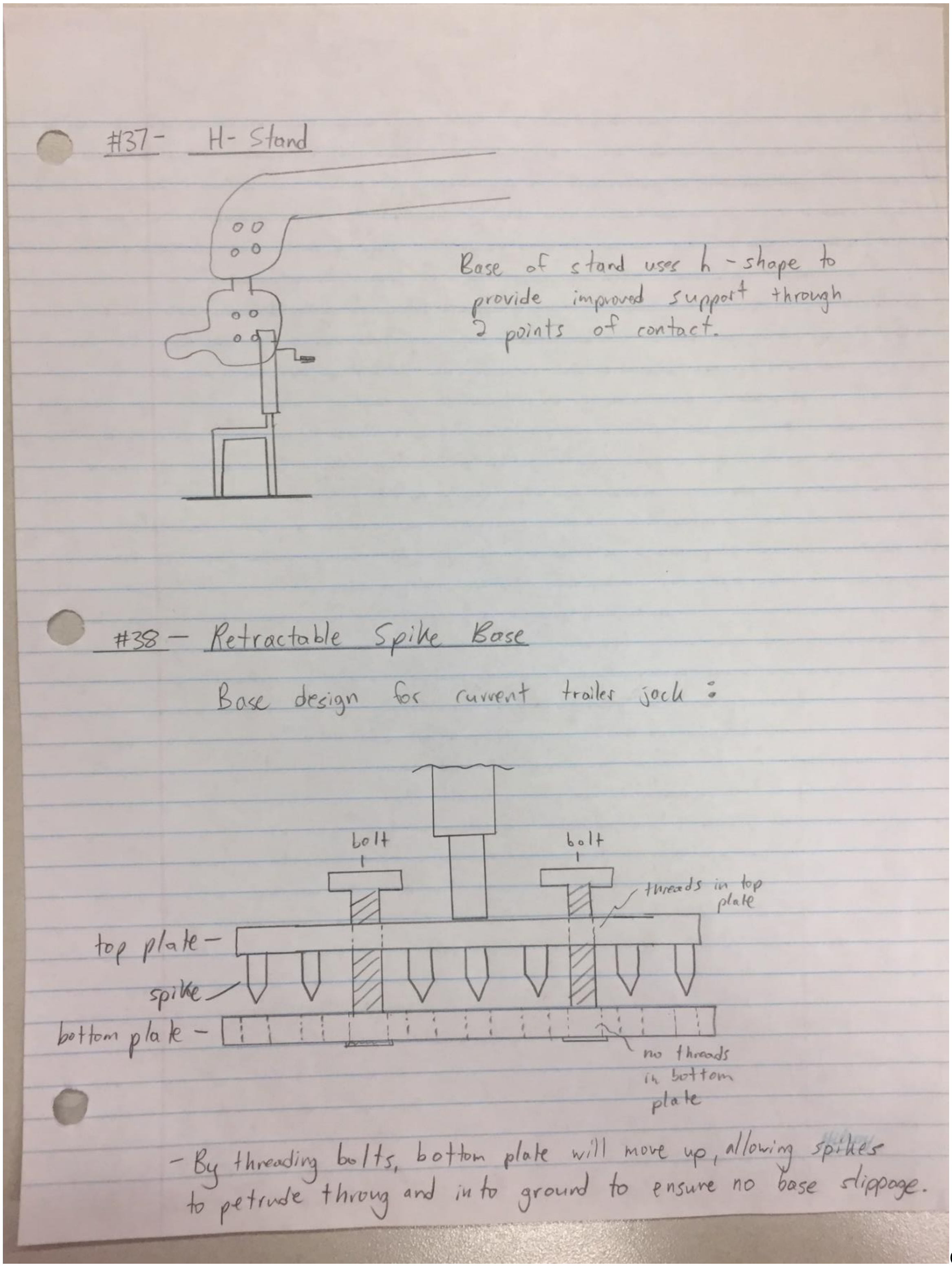


Figure 38: Concepts 35-36.


#37 - H-Stand
Base of stand uses h-shape to provide improved support through 2 points of contact.
#38 - Retractable Spike Base
Base design for current trailer jack :
bolt
bolt
threads in top plate
top plate -
spike
bottom plate -
no threads in bottom plate
- By threading bolts, bottom plate will move up, allowing spikes to petrude throug and in to ground to ensure no base slippage.
dffd

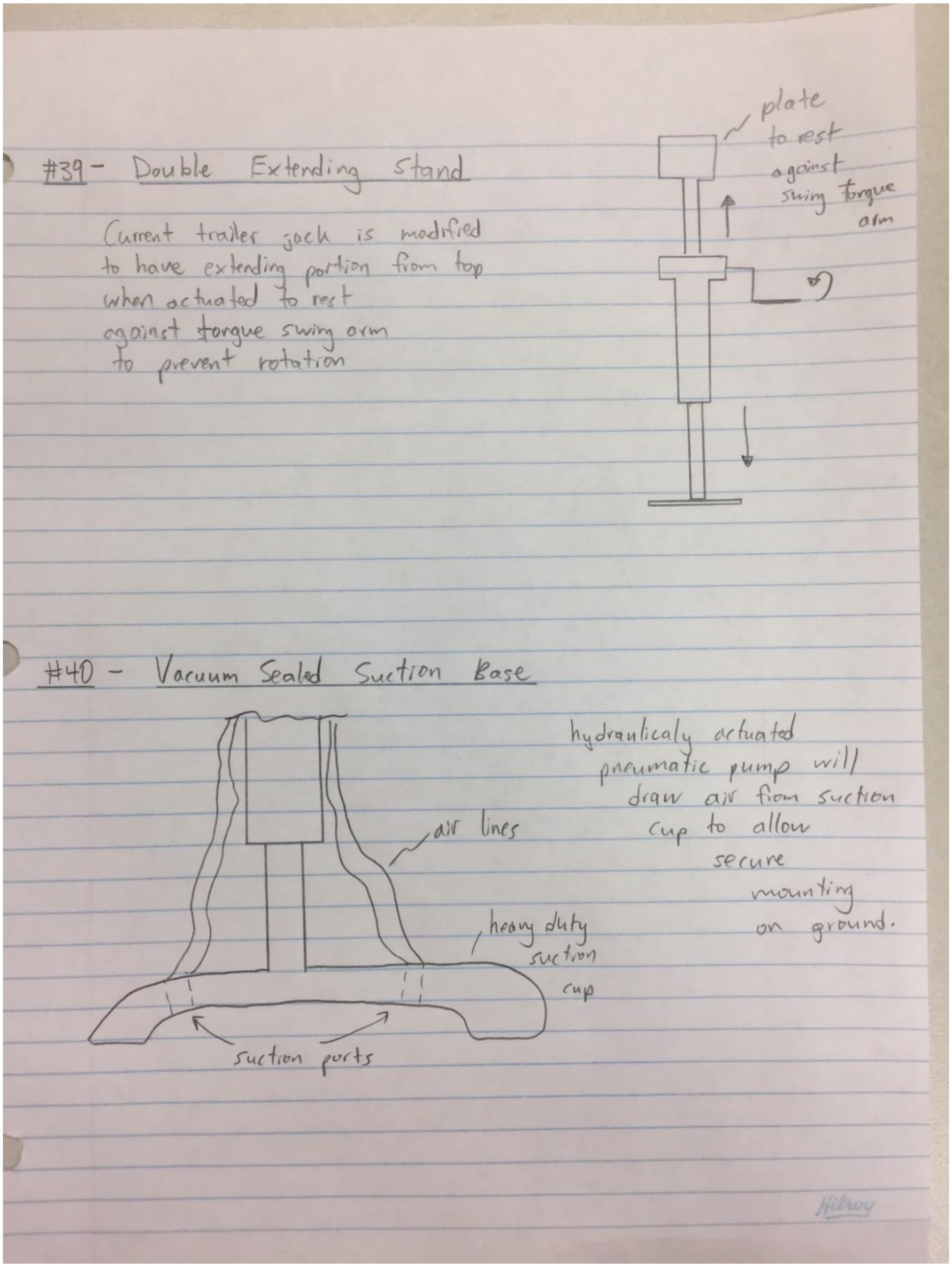


Figure 39: Concepts 39-40.

A.3.0 Concept Generation

The purpose of the concept generation section of this project will be to produce an approximate description of the technology, working principles and form of the final design products. This will include a concise description of how the products will satisfy the customer needs utilizing sketches with accompanying textual descriptions. In order to produce these design concepts, a modified five step methodology was utilized to effectively generate conceptual designs. This five step method included initially clarifying the problem, performing internal research, external research, a systematic concept exploration and the final step being reflecting on the process.

A.3.1Clarifying the Problem

Before any brainstorming or actual concept generation took place, the first step was to re-clarify the problem, ensuring a broad project description to not limit creativity. After reviewing the client needs, metrics, and the project limitations and constraints, our group decomposed and simplified the problem. We identified the overall problem that needed addressing as the instability of the current jack stand design. It was identified that this problem could be solved by either directly improving the stability of the rotary header jack stand, or by eliminating the gearbox rotation, hence indirectly increasing stability. It was also important to note that a combination of these two solutions may also be a viable option. With a solid clarification of the problem at hand, the group could now move onto the internal search step of the conceptual design phase.

A.3.2 Internal Search

The first step of the internal search was to individually generate conceptual designs. Each group member was tasked with the quantitative goal of individually developing 10 concepts. The idea behind this was that a large pool of potential concepts would be generated, with little to no bias or influence from other team members. As expected, a good variety of varying designs were developed. Each of these 40 designs is included in Appendix A: 40 Initial Concept Generation Results. Next, collectively as a group we used the “Gallery Method”. This method was performed by each group member sketching their 10 concepts on a large classroom whiteboard. The 40 corresponding whiteboard sketches can be seen in the following four figures:

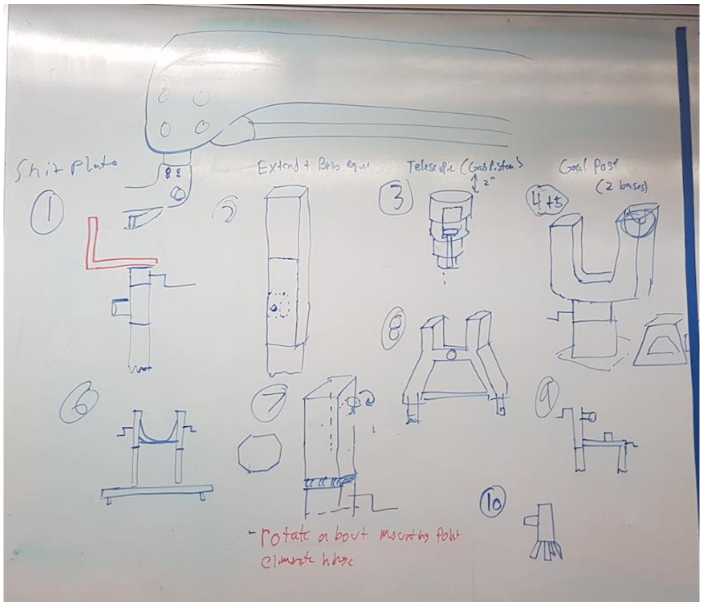


Figure 40: Whiteboard Sketch of Concepts 1-10.

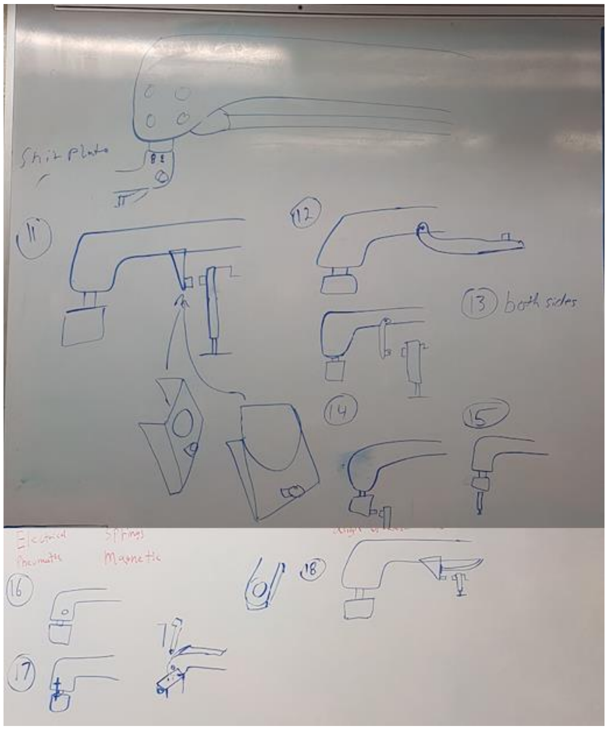


Figure 41: Whiteboard Sketch of Concepts 11-20.

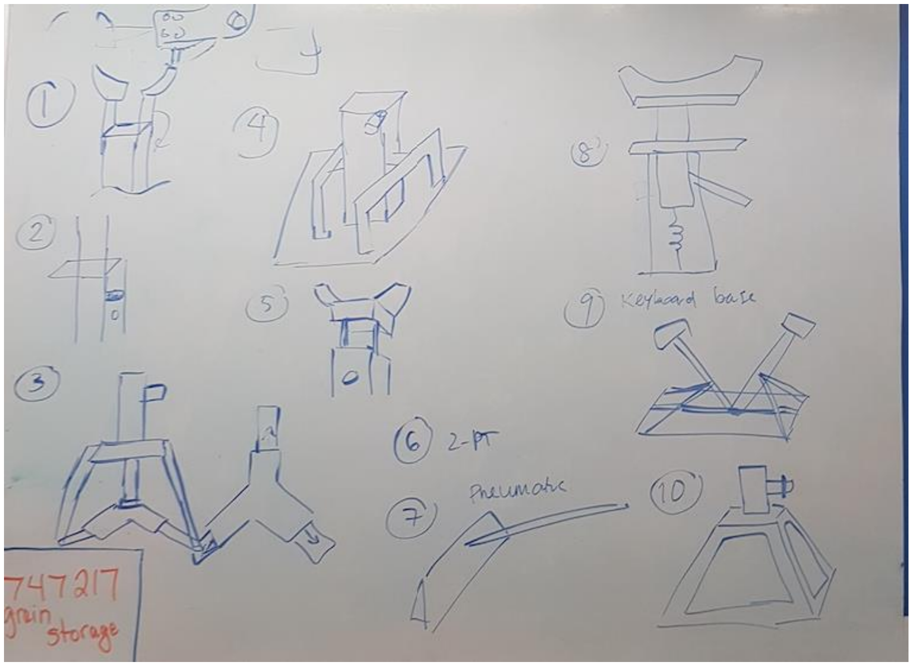


Figure 42: Whiteboard Sketch of Concepts 21-30.

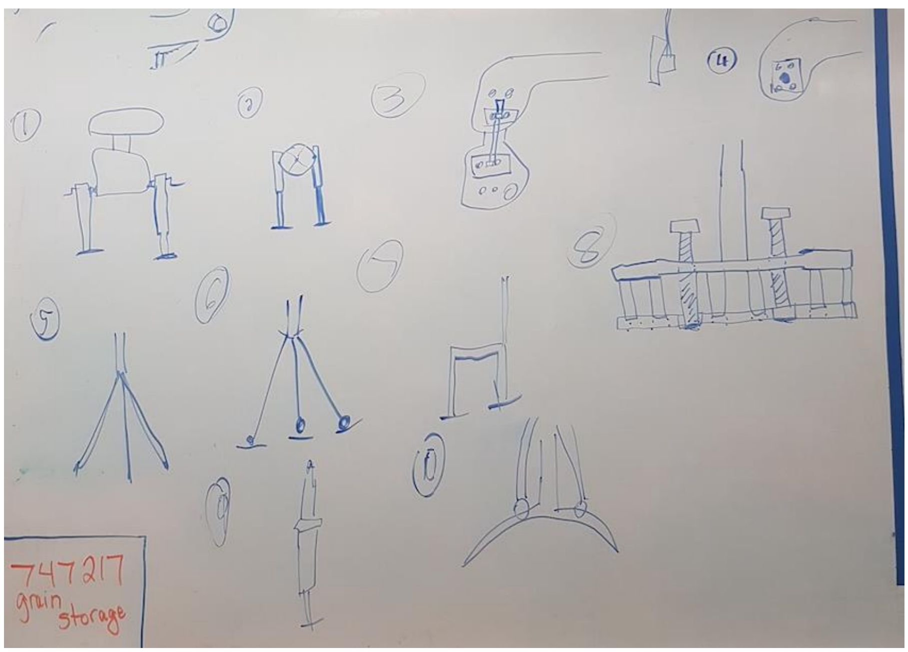


Figure 43: Whiteboard Sketch of Concepts 31-40.

After the ten designs of each group member were sketched on the whiteboard, the corresponding group member gave a brief explanation of his/her concepts, followed by any questions other group members had regarding their designs. Finally, potential design changes and improvements were also mentioned which helped to generate potentially new ideas from the original concept.

Upon completion of the “Gallery Method”, our design team switched internal search focuses onto another concept generation method known as SCAMPER. The acronym stands for substitute, combine, adapt, modify, put, eliminate, and reverse. Each of these terms relates specifically to how each individual term could be applied to the concepts to help generate new conceptual designs. In order to simplify the use of the SCAMPER method, we looked at all the initial concepts and broke them down into a form of a concept combination table which was drawn up on the whiteboard as seen below in Figure 44.

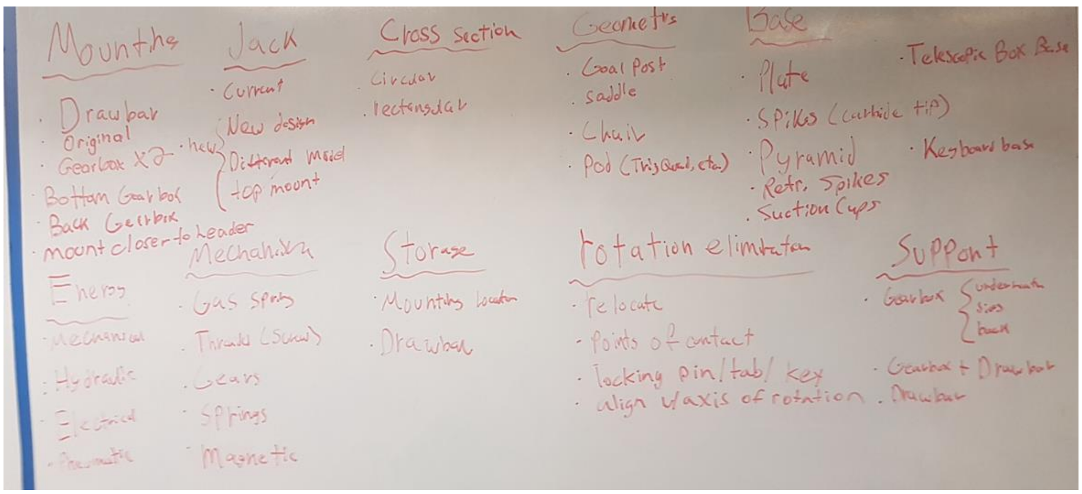


Figure 44: Whiteboard Concept Combination Table.

This concept combination table consisted of ten classifications of concept features, each with multiple variations of accomplishing that specific feature. A clearer visual summary of this whiteboard combination table can be seen below in the

TABLE I.

TABLE I: CONCEPT FEATURE CLASSIFICATIONS AND ACCOMPANYING MEANS

Mounting Location	Jack	Jack Cross Section	Geometry	Base	Energy	Jack Mechanism	Storage	Rotation Elimination	Support
Original	Current	Circular	Goal Post	Plate	Mechanical	Threaded	Current Location	Relocate	Underneath Gearbox
Swing Tongue	New Design	Rectangular	Saddle	Spikes	Hydraulic	Gas Piston	Swing Tongue	Points of Contact	Gearbox Sides
Gearbox (both sides)	Different Model		Chair	Pyramid	Electrical	Gears		Locking Pin/Tab/Key	Gearbox Back
Back of Gearbox			Post (Triangular)	Retractable Spikes	Pneumatic	Spring		Align with Rotational Axis	Gearbox & Swing Tongue
Closer to Header			Single Post	Suction Cup		Magnetic			Swing Tongue
				Telescopic Box					
				Keyboard					

As a group, an extensive amount of time was allotted to run concepts through the developed concept combination table, allowing us to develop advantageous design changes. This was accomplished by first determining which features of the concept classifications the design we were critiquing had, and then choosing other possible combinations that may work with this design. In addition to this, the SCAMPER method was also utilized to look for any possible modifications which may have been overlooked using the combination table. In the conclusion of utilizing these two combined methods, we were able to develop ten designs that the team collectively felt would be good to move forward with. A table of the initial top ten concepts generated is listed below in

TABLE II.

TABLE II: TOP 10 INITIAL CONEPTUAL DESIGNS

1	Current jack and plate attachment
2	Tripod stand
3	Swing tongue plate and jack
4	Mount current jack off plate on back of gearbox
5	Mount jack off swing tongue
6	Swing tongue and gearbox locking mechanics
7	Folding tube jack
8	Current jack with corded pin and bracket
9	Back of gearbox lock-in mount plate
10	Back of gearbox lock-in mount plate and bar (clamp)

Through the utilization of individual brainstorming, the “Gallery Method”, SCAMPER, and a combination table, ten initial concepts were developed. These ten initial concepts generated though the internal search portion of this project can now be carried forward into the external search portion of this project. The 10 concepts are pictured in the following figures.

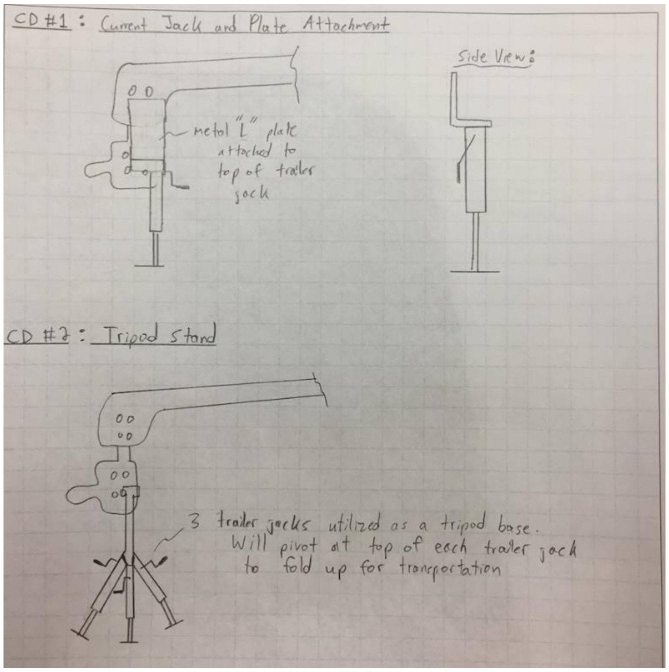


Figure 45: Conceptual Designs 1 and 2.

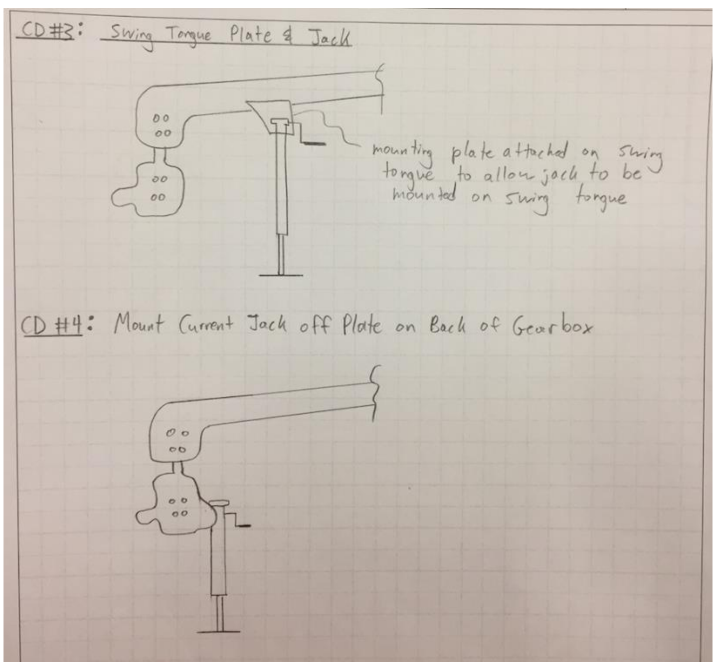


Figure 46: Conceptual Designs 3 and 4.

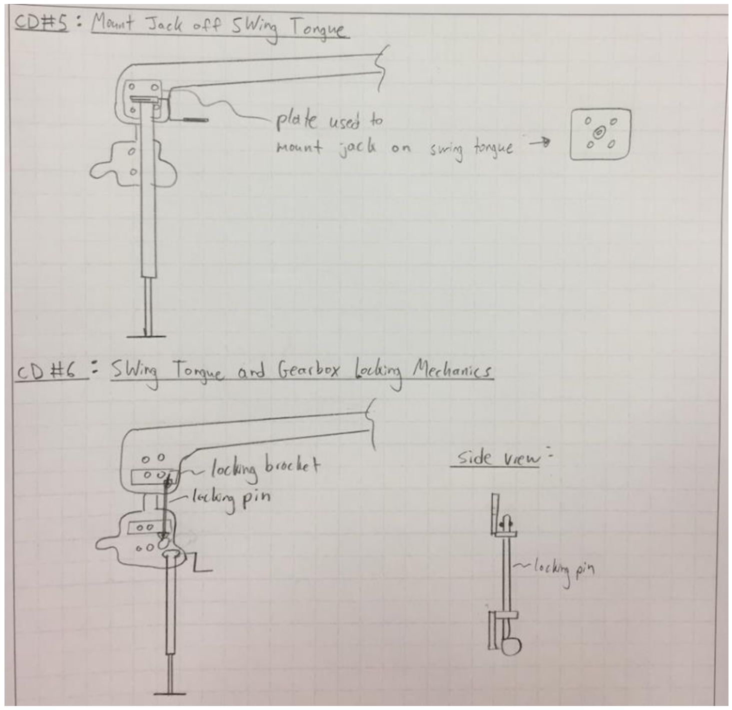


Figure 47: Conceptual Designs 5 and 6.

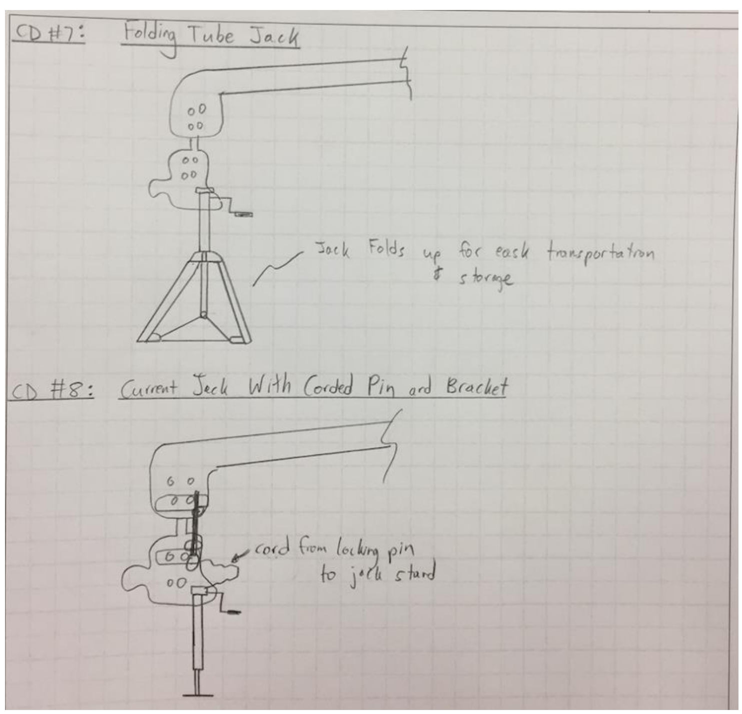


Figure 48: Conceptual Designs 7 and 8.

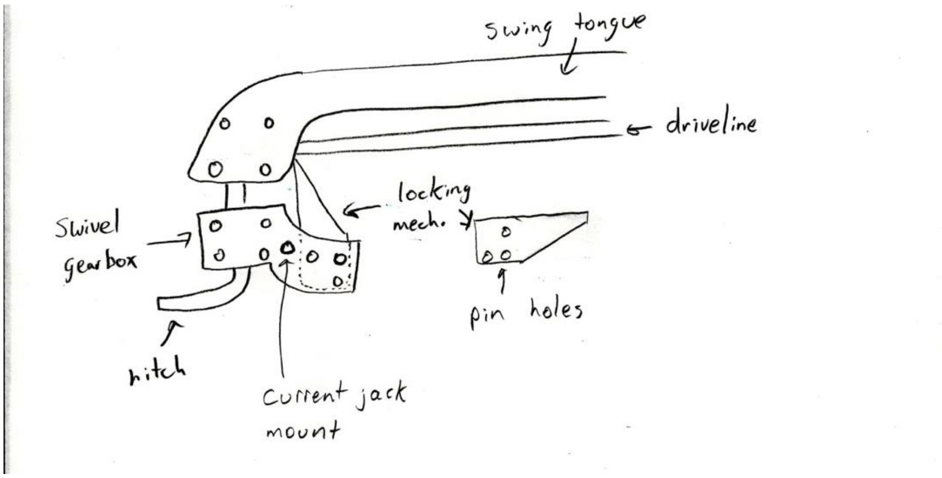


Figure 49: Conceptual Design 9.

Conceptual Design 9 makes use of already existing holes in the lower swivel gearbox weldment. On the rear of the weldment, there are three holes which are currently used for pinning the 2-point hitch stand in the appropriate location. This concept uses a flip up locking mechanism which includes the current jack mount. When in the upwards position, the locking mechanism fits between the sides of the swing tongue behind the upper swivel gearbox, thereby preventing rotation. When in the lowered or storage position, the locking mechanism sticks out behind the lower swivel gearbox. The pin used to hold this bracket in position would most likely be the pin that is currently used on the 2-point hitch stand, but the pin would have to pass further analysis to ensure that it would not fail.

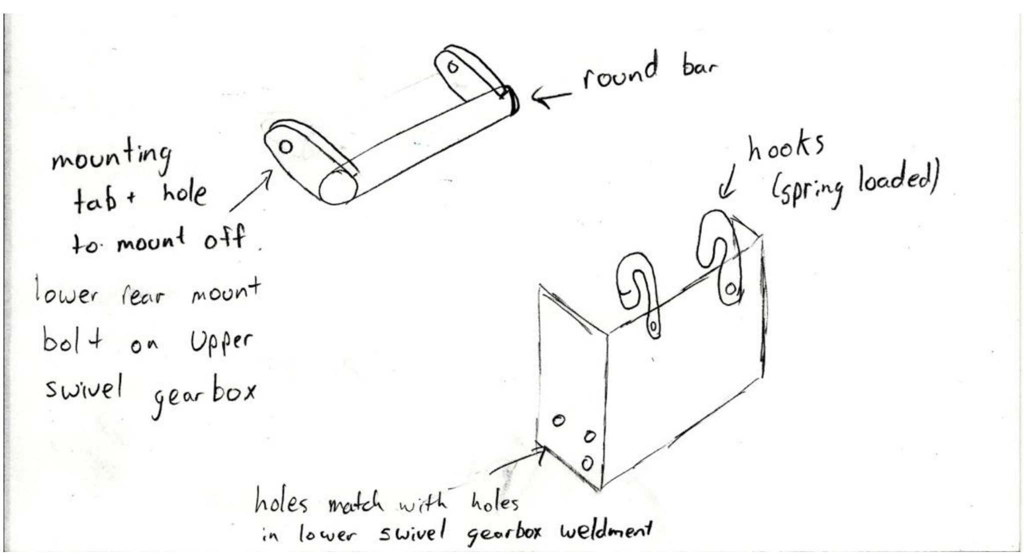


Figure 50: Conceptual Design 10.

Conceptual Design 10 consists of a bar that would be mounted off the lower rear mounting bolts for the upper swivel gearbox. A similar flip up bracket as the bracket in conceptual Design 9 would be placed in the lower swivel gearbox weldment. The flip up bracket for this concept has hooks that flip over the bar, thereby locking the rotation of the swivel gearbox.

A.3.3.External Search

The external search portion of the conceptual design generation focused on important topics such as lead users and experts, patents, literature, and benchmarking. To a certain degree, some of this external search has already be completed as many of the applicable patents, technical standards and competitor's products were already researched and thoroughly discussed in the external research section of this report. The goal moving forward with this section was therefore to apply the researched information to ensure no infringement on applicable patents, technical standard compliance and possibly incorporating aspects of competitors' designs. Once this was achieved, further external research and conceptual development was performed with our client.

Once it was confirmed collectively as a team that the ten initial concepts generated were free from patent infringement and achieved technical standard compliance, a meeting with our client was arranged. The purpose of this meeting was to ensure firstly that the designs were headed in a direction that the client hoped to see, and secondly to receive feedback from the client as he had received valuable feedback from many of his clients on the shortcomings of the current design. The client had no major issues with any of our designs and also did not want to limit our creativity, so we were informed to continue developing our conceptual designs in the manner currently being undertaken. With the completion of the external search complete, our team could move onto the next phase of the conceptual generation.

A.3.4.Reflecting on the Process

The systematic exploration section of the conceptual generation process was technically included within the internal search section. This is due to the fact that the combination table utilized is technically considered a systematic exploration, however our design team felt it would be more effective to utilize the combination table in the internal search to better develop designs prior to meeting with our client. This will ensure effective use of both parties time. As a result, the final reflecting on the process section of the concept generation process could now be moved forward with.

As an engineering design team we feel that we started really strong by generating a good pool of initial concepts. However, moving into the combining phase of the generation we recognized we could have done better. It proved very difficult to combine some designs since they may only prevent gearbox rotation, while others may only directly improve stability. This resulted in essentially two types of solutions, with others lying in the middle ground. With that being said, we still can not think of a better way of developing concepts that could have been utilized, however better design documentation through the phases would have helped to better illustrate this process.

A.3.5.Concept Generation Summary

Through the use of a modified approach of the five step method, 10 conceptual designs were successfully generated to move forward with into concept analysis and selection. This was accomplished by initially clarifying the problem and decomposing it into two main sub categories. Next, an extensive internal research was completed by utilizing individual brainstorming, the “Gallery Method”, SCAMPER and finally a concept combination table which was a method of a systematic exploration. The external search included utilizing previously performed external research results on patents, technical standards and competitor’s offerings to tweak our concepts before presenting them to our client for initial feedback. Finally, the process was reflected on to understand our shortcomings and how our concepts may require further refinement still.

A.4.0 Guard Bar Mesh Convergence Stress Results

The first mesh applied to the guard bars resulted in a max stress obtained of 56.77 MPA as seen in figure

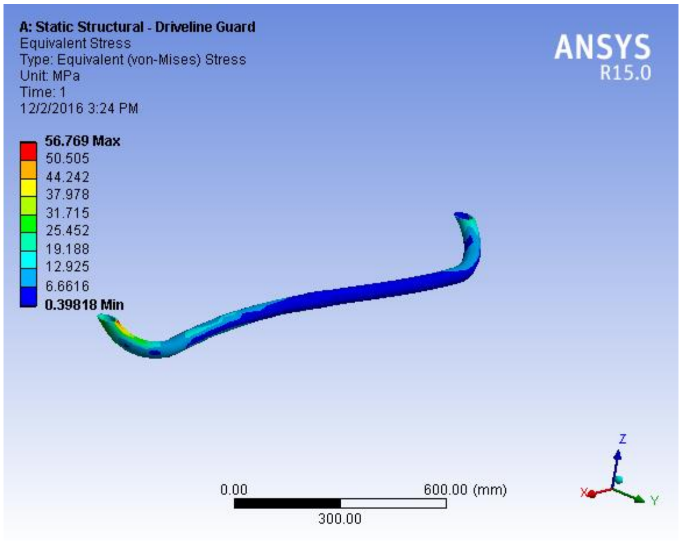


Figure 51: Guard bar iteration one stress results.

The second iteration of the guard bar analysis saw mesh refinement along the edge of the circular tubing's top flat face. This gave the stress distribution seen in Figure 52, with a max stress of 89.417 MPA.

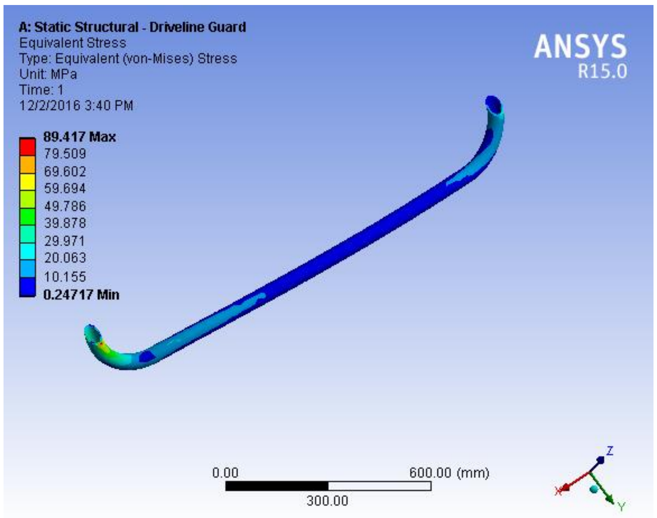


Figure 52: Guard bar iteration two stress results.

A.5.0 Meeting Target Specifications

This section details how the V-support meets the target specifications laid out by the team at the beginning of the design project. The marginal and ideal values for each target specification were created through discussion with the client and based on sound engineering principles. For each target specification, a marginal and ideal value was created. If the design met the marginal technical specification, the design would pass.

The V-support utilizes the same jack as the current stand design. Using the same jack in the proposed design is an important aspect of the design to MacDon. This is because the addition of a second jack would increase the number of part numbers in MacDon's system, increase customer confusion over which jack to order and increase MacDon's logistical footprint.

The jack mount structure is compatible with both the drawbar and 2-point hitch styles. The mounting location of the V-support and jack is further from the decal locations on the swing tongue than the ideal values of 14 mm. As the V-support frame requires the driveline guards to be extended, the design is not backwards compatible. The driveline guards had to be extended to prevent damage to the tractor pulling the header. Therefore the V-support is not backwards compatible with units that have previously sold.

The newly added stationary V-support bracket weighs 3.8 kg, which is 3.0 kg lighter than the ideal weight of 6.8 kg. The low weight added to the header means that increases in manufacturing and shipping costs will remain low. Due to the use of the same jack in the teams design as the original design, the team knows that the weight of the jack, which is a component that an operator might have to move by hand, has been approved by MacDon.

The stand meets the requirements of the relevant standards that the team found, thereby confirming that the stand is technically compliant.

The V-support mounting location improves the stability of the hitch in all conditions, as the weight of the swing tongue is applied downwards through the rigidly mounted V-support and jack stand. Based on an FEA analysis of the V-support, the design supports the ideal load of 16680 N and has a factor of safety on this load of 2.91. In addition to the increased stability created by the change in loading conditions, the V-support frame has been numerically analyzed via FEA to confirm that side loads up to 1920 N applied at the base of the jack will be supported by the V-support. The V-support has been analyzed for fatigue failure based on 7000 cycles. 7000 cycles was chosen as it represents 2 load cycles a day for 350 days/year for 10 years. Under this fatigue case, the support structure does not fail. The durability of the jack stand will increase as the stand will be loaded more closely to the ideal loading the jack was intended to support.

The jack stand is easy to operate in all positions and has reduced set up time compared to the original design. In the original design, it was possible for the jack to become angled to a position that prevented the operator from turning the jack handle to raise or lower the jack. This caused the operator to have to find a suitable method to straighten the jack so that the handle could be used. The new design prevents this unwanted rotation. Therefore the jack handle can be used throughout its entire range of motion. Additionally, there is space around this range of motion to account for hand clearance so that the operator is not inconvenienced while lifting or lowering the jack.

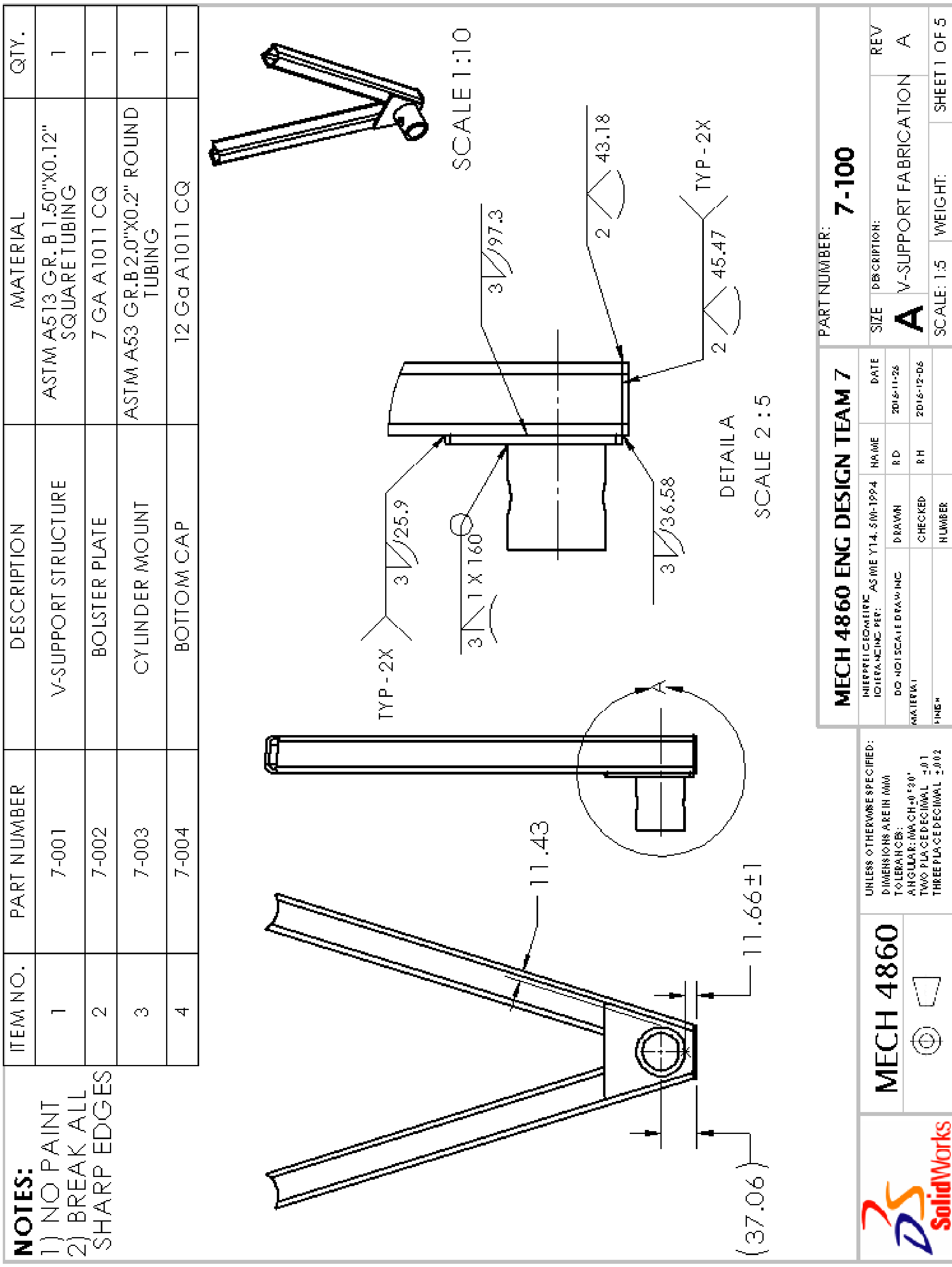
As the tube and pin connection used to mount the jack have not changed, the current storage location can continue to be utilized. Due to the positioning of the V-support, the jack is closer to the storage mount on the swing tongue. The improved proximity between the jack mount and jack storage locations reduced the amount of time required to disassemble and store the jack and will increase the likelihood that the jack is stored it the correct location. The V-support is maintenance free and the jack is reused, therefore the only tool required for maintenance is a grease gun. The jack stand is easily accessible from all angles for maintenance on the V-support structure and does not block access to any current service points on the header. All components of the jack and jack stand are as easily replaceable in the V-support design as the equivalent components in the original design. Due to these factors, the required maintenance for the V-support structure and jack meets the ideal value of 30 seconds.

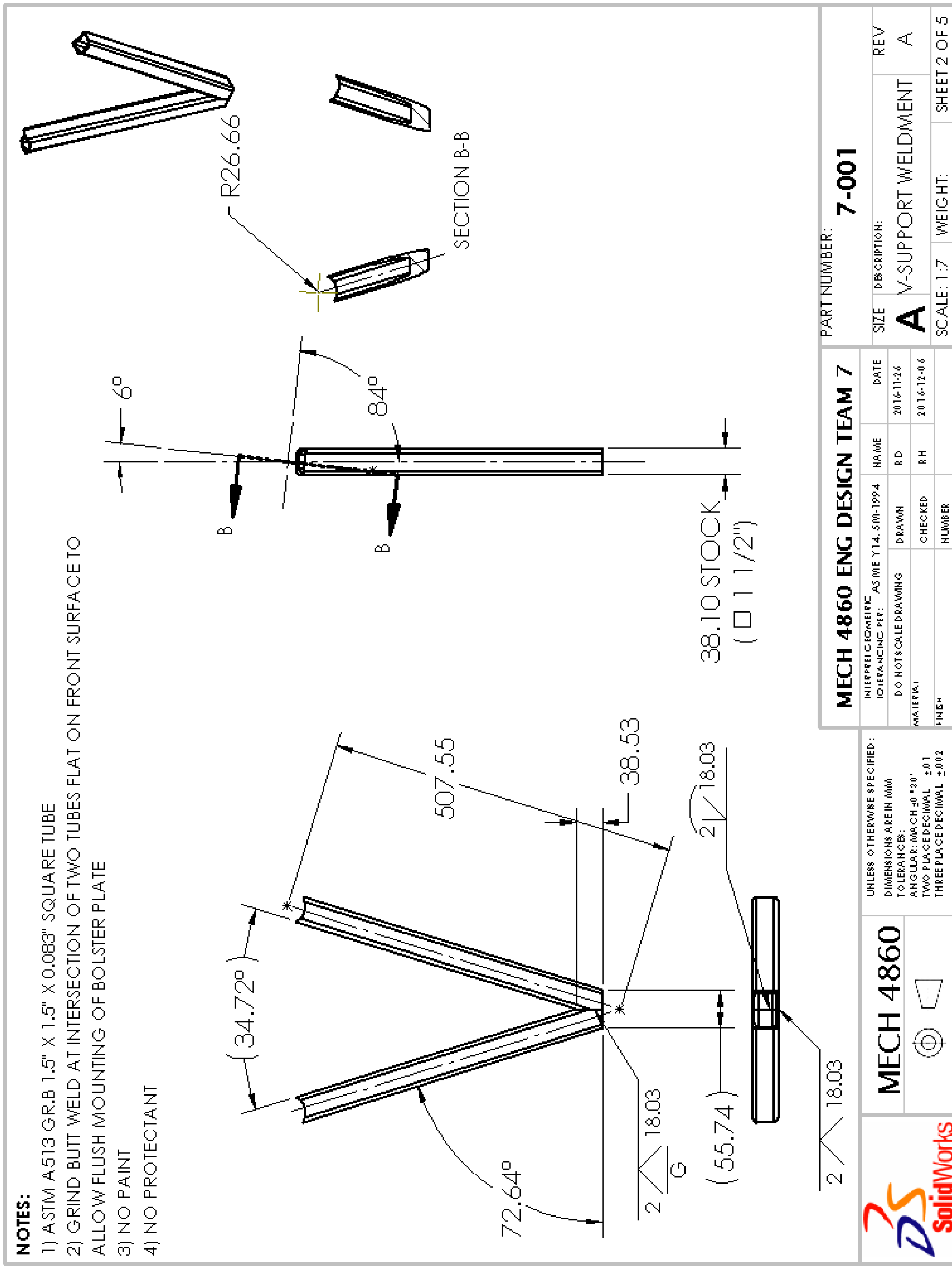
The improved stability of the hitch and the ability for the hitch to rotate for minor adjustments reduces the effort of operators to attach a tractor to the hitch. This reduced effort means that 1 operator can more easily attach the header to the tractor.

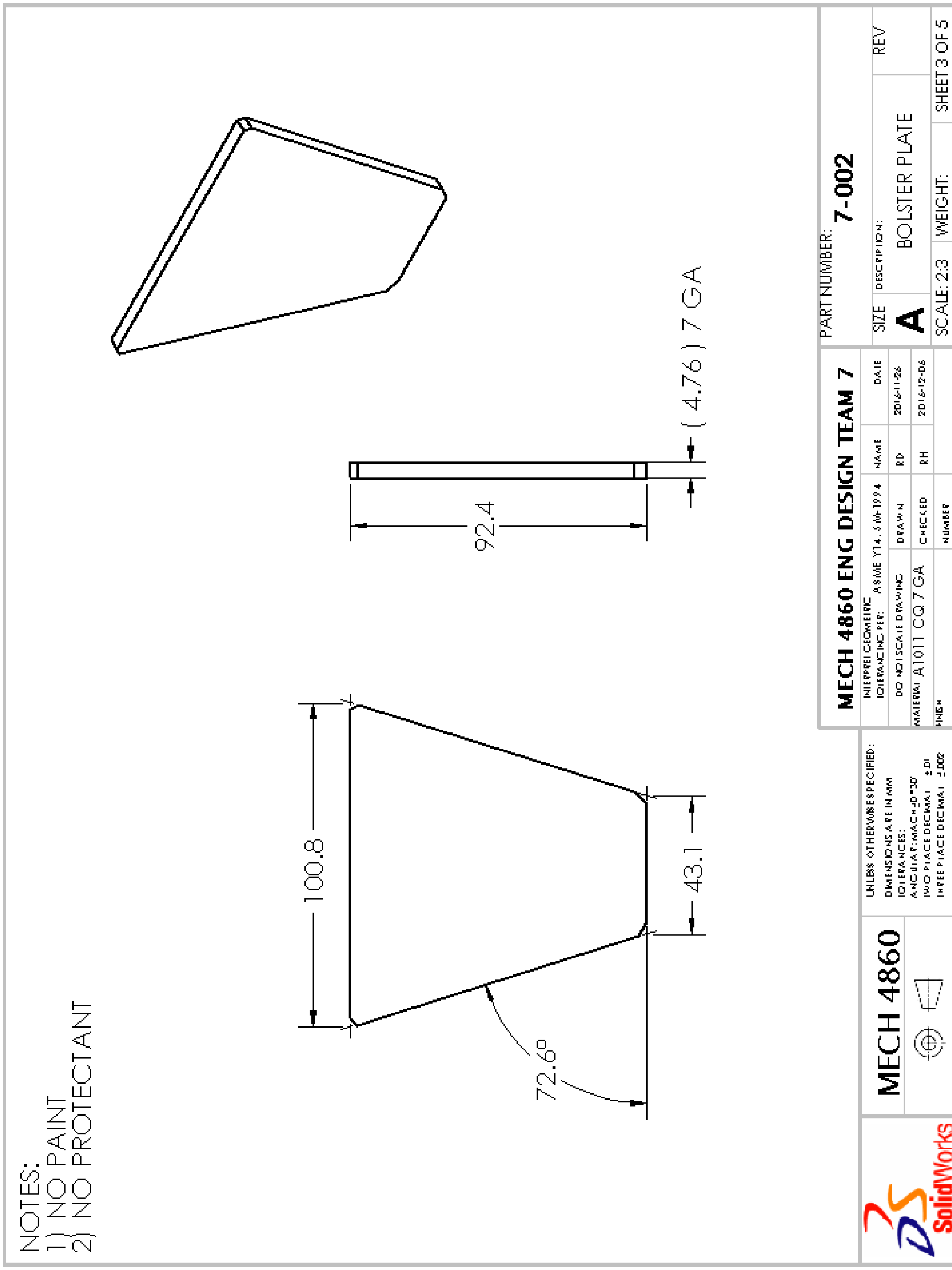
The team was provided a budget of $100CAD for the design. The jack costs $50 CAD and should be included in this cost. The designed components have a manufacturing and fabrication cost of $80.27 CAD, which raises to $130.27 CAD after the jack is added to the cost.

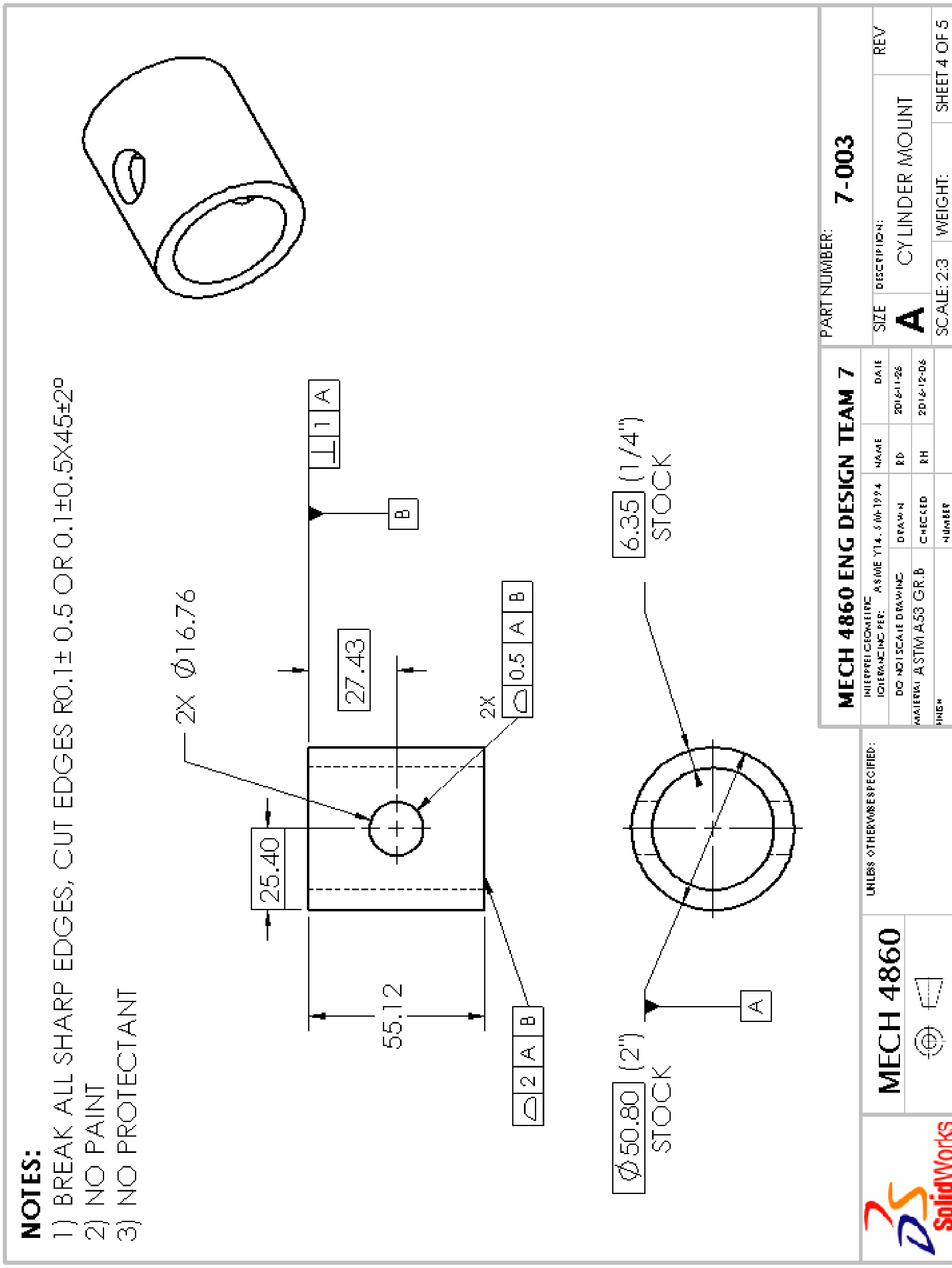
The team's design meets all of the needs of the client and all of the target specifications except for the cost and backwards compatibility needs. The team recognizes that MacDon's production costs will be significantly less than the prices used for the cost breakdown, thereby reducing the part costs. In terms of backwards compatibility, the client has expressed support for extending the driveline guards to move the V-support away from a position where accidental contact with tractor tires can occur.

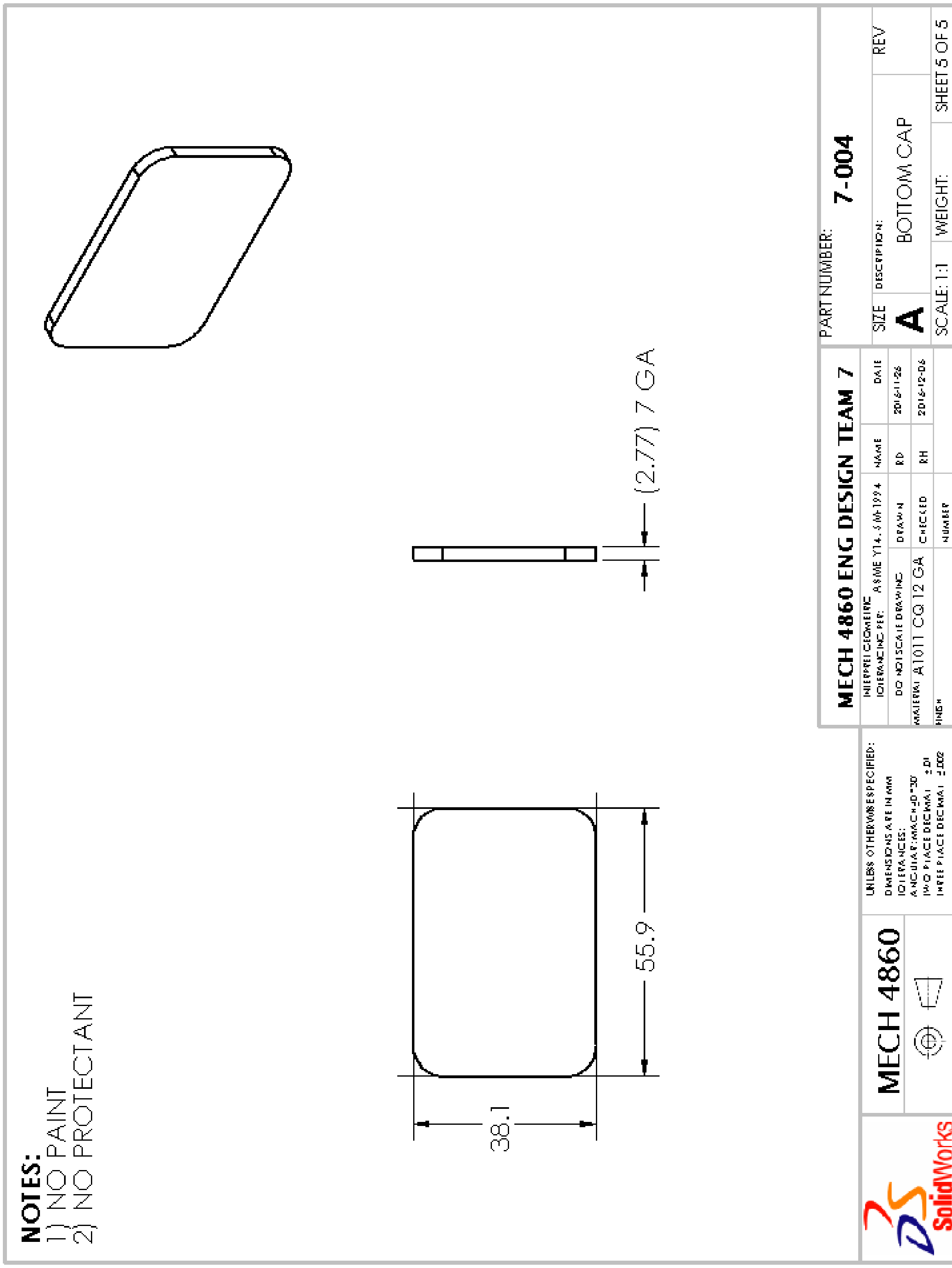
A.6.0 Technical Drawings




NOTES:
1) ASTM A513 GR.B 1.5" X 1.5" X 0.083" SQUARE TUBE
2) GRIND BUTT WELD AT INTERSECTION OF TWO TUBES FLAT ON FRONT SURFACE TO ALLOW FLUSH MOUNTING OF BOLSTER PLATE
3) NO PAINT
4) NO PROTECTANT
72.64°
(34.72°)
507.55
38.53
18.03
(55.74)
R26.66
SECTION B-B
6°
84°
B
38.10 STOCK
(□ 1 1/2")
MECH 4860 ENG DESIGN TEAM 7
PART NUMBER:
7-001
DESCRIPTION:
V-SUPPORT WELDMENT
SIZE
A
REV
A
SCALE: 1:7
WEIGHT:
SHEET 2 OF 5
NAME
DATE
DRAWN
RD
2016-11-26
CHECKED
RH
2016-12-06
DO NOT SCALE DRAWING
MATERIAL
FINISH
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
MECH 4860
SolidWorks


NOTES:
1) NO PAINT
2) NO PROTECTANT
100.8
43.1
72.6°
92.4
(4.76) 7 GA
MECH 4860 ENG DESIGN TEAM 7
PART NUMBER:
7-002
DESCRIPTION:
BOLSTER PLATE
SIZE
A
REV
SCALE: 2:3
WEIGHT:
SHEET 3 OF 5
ASME Y14.5M-1994
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
RD
2016-11-26
CHECKED
RH
2016-12-06
MATERIAL A1011 CQ 7 GA
FINISH
NUMBER
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
MECH 4860
SolidWorks


NOTES:
1) BREAK ALL SHARP EDGES, CUT EDGES R0.1± 0.5 OR 0.1±0.5X45±2°
2) NO PAINT
3) NO PROTECTANT
2X Ø16.76
25.40
27.43
55.12
2X
0.5 A B
1 A
B
2 A B
Ø50.80 (2")
STOCK
A
6.35 (1/4")
STOCK
UNLESS OTHERWISE SPECIFIED:
MECH 4860
MECH 4860 ENG DESIGN TEAM 7
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5M-1994
DO NOT SCALE DRAWING
MATERIAL ASTM A53 GR.B
FINISH
NAME
DATE
DRAWN
RD
2016-11-26
CHECKED
RH
2016-12-06
NUMBER
PART NUMBER:
7-003
SIZE
A
DESCRIPTION:
CYLINDER MOUNT
REV
SCALE: 2:3
WEIGHT:
SHEET 4 OF 5
SolidWorks


NOTES:
1) NO PAINT
2) NO PROTECTANT
38.1
55.9
(2.77) 7 GA
PART NUMBER:
7-004
REV
SIZE
A
DESCRIPTION:
BOTTOM CAP
SCALE: 1:1
WEIGHT:
SHEET 5 OF 5
MECH 4860 ENG DESIGN TEAM 7
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5M-1994
NAME
DATE
DRAWN
RD
2016-11-26
CHECKED
RH
2016-12-06
DO NOT SCALE DRAWING
MATERIAL A1011 CQ 12 GA
FINISH
NUMBER
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: MACH ±0°30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.002
MECH 4860
SolidWorks

Works Cited

[1] New Holland, *New Discbine Center-Pivot Disc Mower-Conditioners,* New Holland: New Holland, 2013.

[2] T. S. Spade, "New Holland Discbine 313," 23 September 2015. [Online]. Available: https://s-media-cache-ak0.pinimg.com/originals/f3/1e/77/f31e7725b26a2b282d043096ae1f526c.jpg. [Accessed 13 November 2016].

[3] T. S. Spade, "New Holland Discbine 313," 25 September 2015. [Online]. Available: https://s-media-cache-ak0.pinimg.com/originals/c7/ee/07/c7ee07ea4ff7cc7dc46c26513818d316.jpg. [Accessed 13 November 2016].