Letter of Transmittal

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Capstone Design Projects

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FROM:
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RE: Testing Load Cart Redesign Final Design Report

Dr. Labossiere,
The following report on Agriculture Testing Solutions capstone design project is being submitted for your review. This report includes the project definition, concept generation, and a detailed breakdown of the final design. All sections have been written to the best of our abilities, with all referenced material cited.

We would like to thank the Department of Mechanical Engineering, Dr. Sean O'Brien, Dr. Paul Labossiere, Mr. Lorne Grieger, and Mr. Harvey Chorney for the guidance given, and the prospect of available funds for the project.

Thank you and please contact us if you have any further questions.

Sincerely,

Carson Krisko
Email: ____________________

Cole Krisko
Email: ____________________

Ethan Davis
Email: ____________________

Zach Grossart
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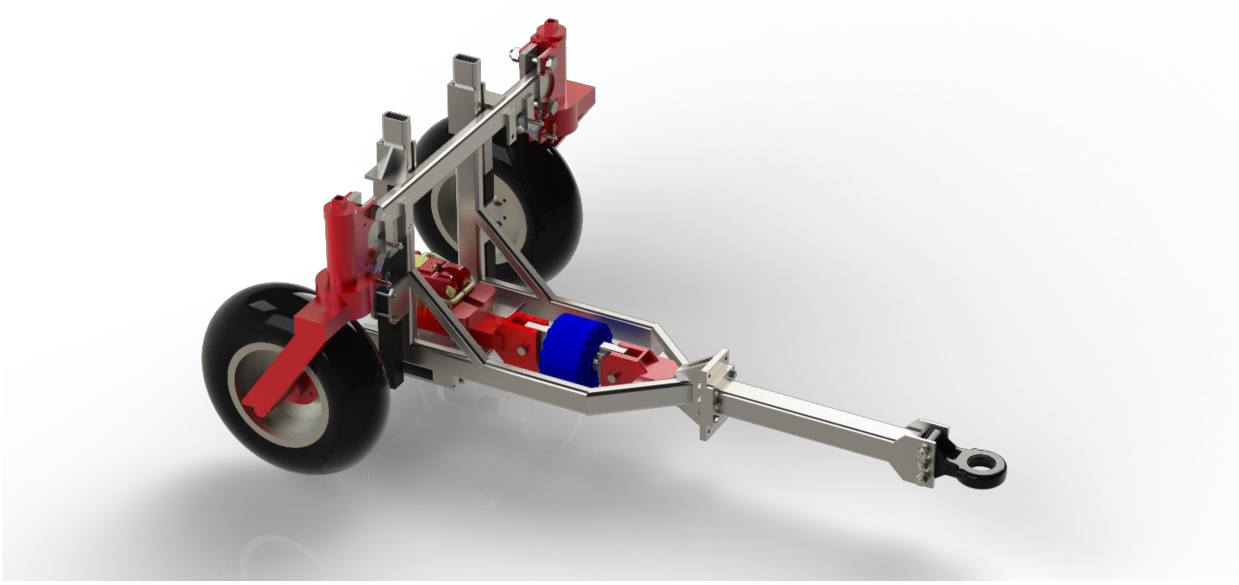

Testing Load Cart Redesign
Final Design Report

Team #4: Agriculture Testing Solutions

MECH 4860 Engineering Design

Prepared For: **Dr. Paul Labossière**

Dr. Sean O'Brien

Mr. Lorne Grieger, P.Eng.

Submission Date: 2018-12-05

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Executive Summary

This report summarizes the project assigned to our team by the Prairie Agriculture Machinery Institute (PAMI), describes the final design, and includes all project deliverables. The project goal is to redesign a testing load cart that is currently used to measure loads applied to a towing machine by an implement. The current load cart has a few major flaws: the load cart wheels cannot track the drawbar of the towing machine through turns, the load cart does not accommodate all necessary hitch categories, and installation and removal of the load cell is time consuming and puts significant strain on the employees.

The main client needs include compatibility with category 3-5 hitches on the front and rear hitch as per ASABE standards, height adjustability of the load cart, the implementation of a PowerPin hitch and two swivel castor wheels, and compatibility with three different load cells. The load cart must be transportable and use standard off-the-shelf materials and parts. The deliverables for the project are Computer Automated Design (CAD) models, preliminary Finite Element Analysis (FEA) of the load cart, a bill of materials, and cost analysis.

Design of the load cart satisfies all needs outlined by the client. The swivel casters have been integrated with and added locking mechanism to allow the load cart to reverse without the wheels interfering with the frame. The casters mount to a horizontal support that extends outside of the frame to allow the casters to rotate 45 degrees. Two weld-on trailer jacks are mounted between the frame and the horizontal support to adjust the height of the rear hitch from 13 to 25 inches. PTFE sheets are used in the rear of the load cart to support a PowerPin hitch while acting as a bearing to mitigate the affect of hitch movement on load cell measurements. The load cell connects the rear hitch to the front of the frame with a vertical installation process to improve operator ergonomics. Once installed vertically, screw adjusters align the load cell connectors with the load cart brackets before the load cell is pinned into place for ease of installation. Two steel pieces from the PowerPin hitch drawbar are repurposed at the front of the frame to strengthen the connection from the load cell bracket to the front hitch extension connection. The front hitch extension is 30 inches long with height adjustability of 2.34 inches both up and down. The front hitch is compatible with the category 3-5 agriculture hitches pre-purchased by PAMI.

The final design complies with all project constraints. The final width and length of the load cart are 6’ 7” by 9’ 5” which fits within the 7’ by 18’ trailer. FEA analysis was conducted on all major load bearing sections of the load cart to satisfy a minimum factor of safety of 2. The load cart design safely tows a draft load of 60,000 lbf with the front hitch extension in the centred position. The maximum load reduces to 30,000 lbf with the hitch extension in the raised position.

The final material costs amount to $5,228 which leaves $4,772 for manufacturing cost including welding, machining, laser cutting, and final component assembly.

A failure modes effect analysis was performed to determine the highest priority risks during operation. The highest risk failures modes are weld failures at the front and rear load cell connections and foreign material striking the load cell during operation. Both failures would cause major damage to the load cell and are not possible to detect before occurrence.

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1 Introduction

Agriculture Testing Solutions has redesigned a testing load cart for Prairie Agriculture Machinery Institute to improve performance and setup time in the field. This report details the final design of the new testing load cart. It will include a summary of the project and concept definition, a detailed final design overview, finite element and failure mode effects analysis. The project definition section will include project background, deliverables, and scope followed by target specifications and constraints. Next, the concept definition section will summarize the load cart design concept generation and selection process. The detailed design section will describe all aspects of the design including integration of all components. This is followed by a stress analysis section including hand calculations and finite element analysis. A section on failure mode effects analysis is included to identify the effect of potential failure modes on the load cart during operation.

2 Background

"Prairie Agriculture Machinery Institute (PAMI) is an applied research, development, and testing company that serves agriculture and other industries in Western Canada" [1]. PAMI's clients have a need for outsourced high-quality, third party testing. These clients rely on PAMI to achieve reliable and accurate test results for their equipment.

The existing load testing cart consists of an A-frame design, two solid axle tire connections, a central load cell mount, front and rear hitch attachments and height adjustability. The old load cart features are displayed in Figure 1 and Figure 2.

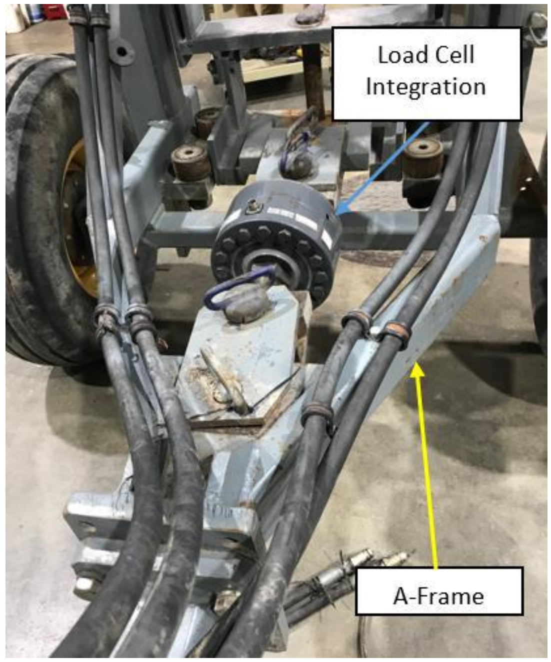


Figure 1. Current load cart with labelled design components (1).

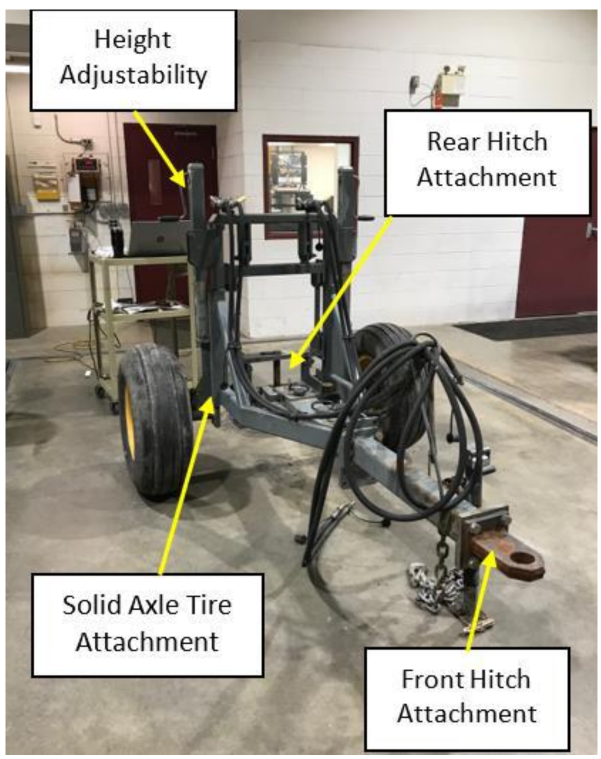


Figure 2. Current load cart with labelled design components (2).

Several major issues exist with the current load cart design. First, the solid axle design does not allow for the wheels to follow the path of the load cart during turns. Second, the installation and removal process of load cells is difficult for the operator. Third, adjusting the load cart for various hitch categories is time consuming and demanding. Finally, there are difficulties in the loading and unloading process with the trailer used for transportation of the load cart.

2.1 Solid Axle Tire Connection

The current load cart tires are fixed on solid axles and cannot swivel to track the trajectory of towing machine. The current tire connection is illustrated in Figure 3.

Figure 3. Solid axle tire connection.

This design causes the load cart wheels to skid during turns and puts a large moment on the axle connection. PAMI has had to repair the part and add gussets to strengthen the connection.

2.2 Main Frame

The current main frame design incorporates the following features: a front hitch, extension beam, an A-frame base, vertical and angled support members, jack mounts, and a rear hitch connection. The current load cart has had an operational life of approximately 15 years. Therefore, all design features incorporated into the current main frame have proven to be effective. The main frame design on the current load is displayed in Figure 4.

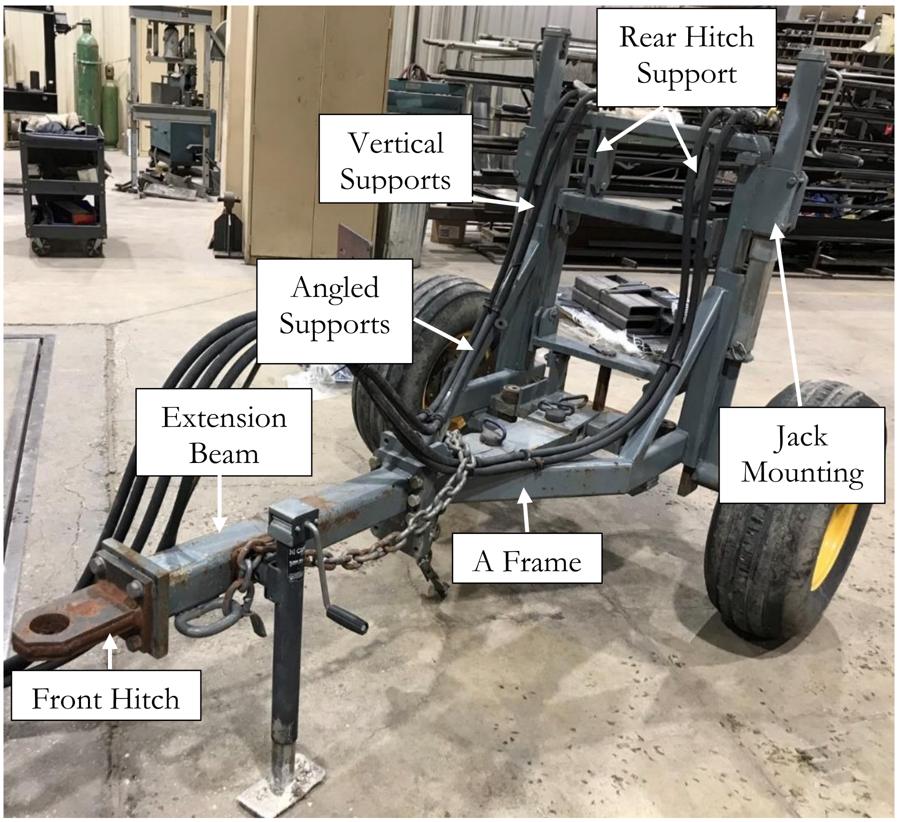


Figure 4. Current load cart frame features.

The starting point for the frame redesign will be based on the main frame features of the current load cart.

2.3 Manual Jacks

The current load cart design features two Fulton square side wind manual crank jacks. The position of both manual jacks on the frame is shown in Figure 5.

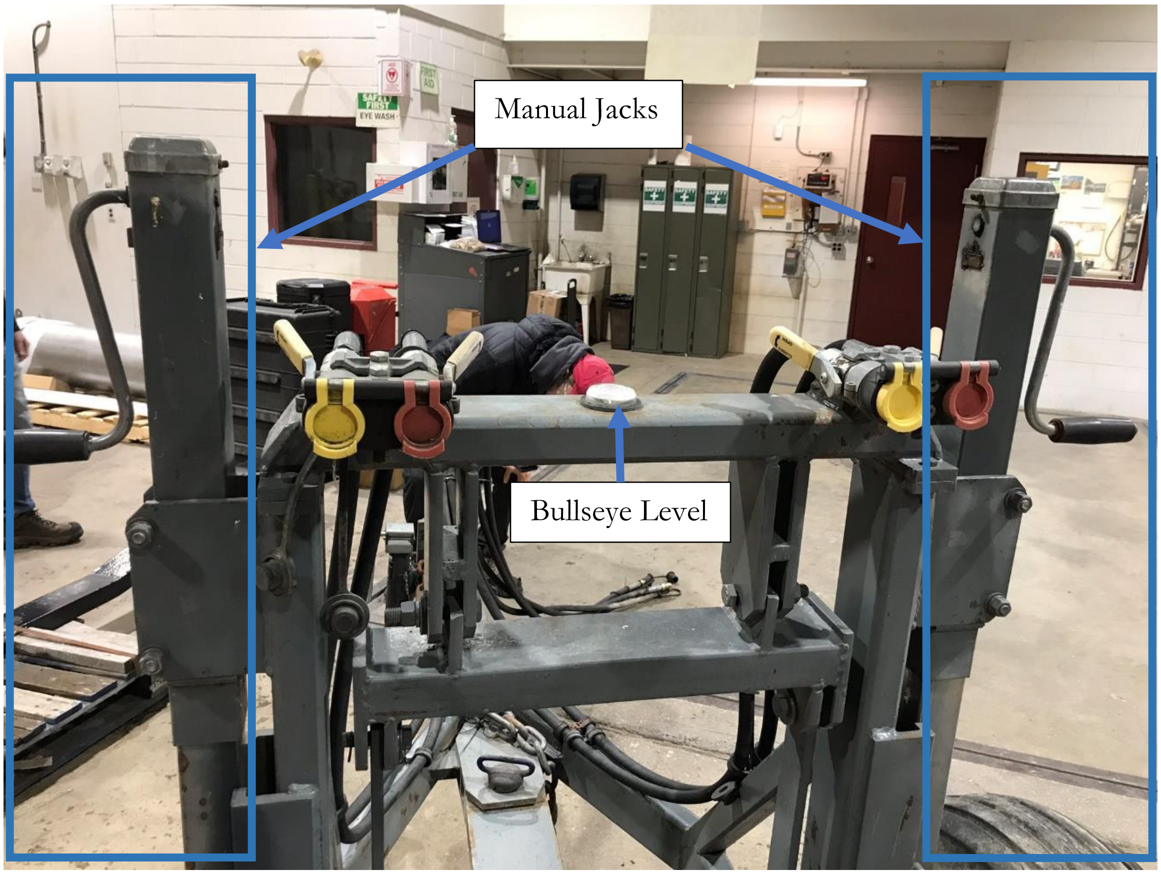


Figure 5. Manual jack positions on frame.

The jacks attach directly to the solid axle tire connections. This allows the height of each jack to be adjusted independently. The bullseye level mounted at the intersection of the centerline of the load cart and the centerline of the manual jacks allows the load cart to be levelled to ensure accurate load cell readings.

2.4 Rear Hitch Connection

The support system for the rear hitch on the current load cart was a hinged support system. The current load cart features cam follower bearings supporting the hinged system. The cam follower bearings at the hitch provide the hinged system with sideways support and prevent metal on metal contact. The hinged system provides the complete vertical support of the rear hitch. A labelled picture of the current load cart rear hitch support system is provided in Figure 6.

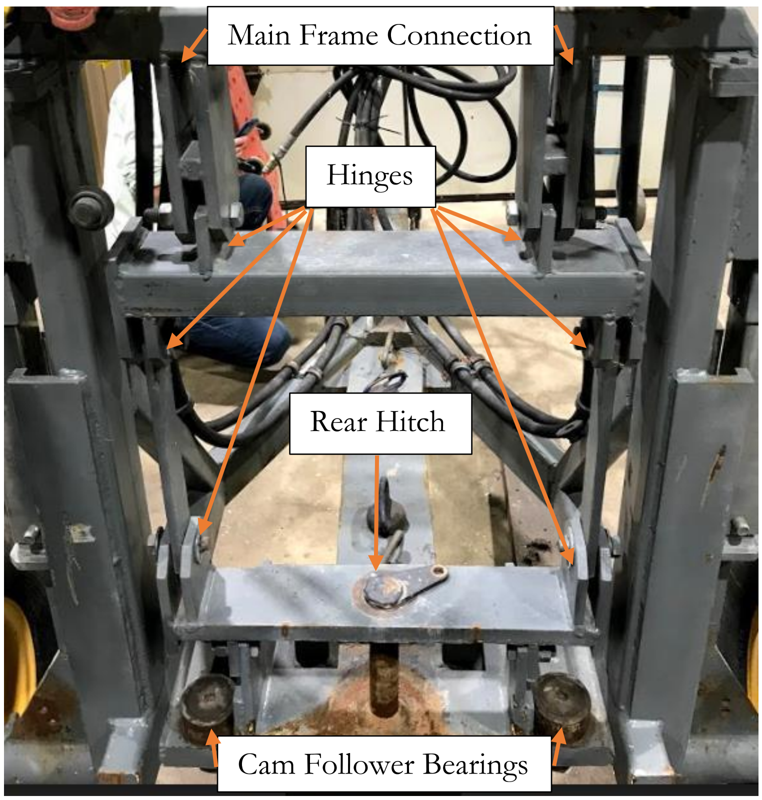


Figure 6. Current load cart rear hitch support system.

The hinged system created problems when installing the load cell as it is not perfectly balanced at the base and requires extra support during the installation process.

2.5 Load Cell Integration

The current load cart design utilizes H-shaped brackets to connect the load cell to the frame. The H-shaped brackets can be observed in Figure 7 and Figure 8. The load cell connection is pinned to one H bracket attached to the main frame and the other H bracket is pinned to the rear hitch. Load cell installation with these brackets is difficult and time-consuming due to very tight clearances with the load cell connection.

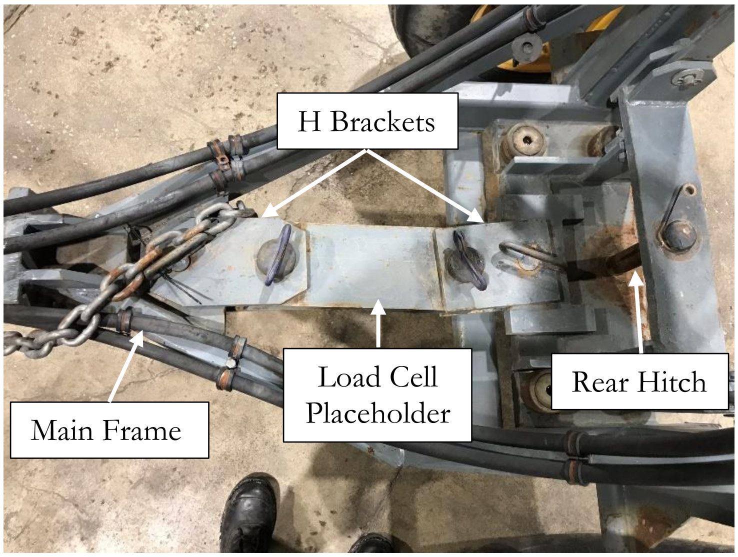


Figure 7. Current load cell integration method.

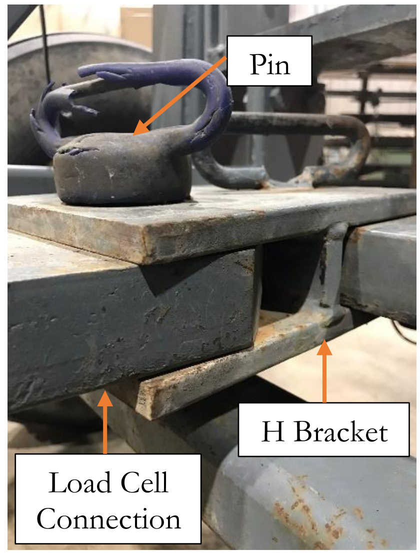


Figure 8. Side view of H bracket.

The lack of space between the main frame and the H brackets present another difficulty for installing the load cell as the connector piece must be installed into the H bracket horizontally.

2.6 Front Hitch Connection

The current load cart uses a customized front hitch to connect with the tow vehicle. This front hitch connection can be observed in Figure 9.

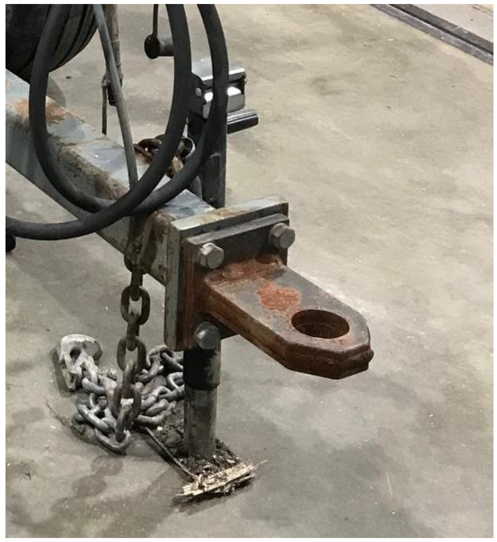


Figure 9. Previous load cart front hitch connection.

The current front hitch is not a standardized hitch configuration and does not allow for the integration of pre-purchased standard category 3 to 5 agriculture hitches.

2.7 Additional Design Features

The current load cart has several additional design features to improve safety and allow the implement and tractor to be linked. These are a safety chain, front hitch height adjustment, and hydraulic system extensions.

2.7.1 Safety Chain

A requirement per ANSI/ASAE S338.5 is to include a safety chain between a tow vehicle and the towed agriculture field equipment. The current load cart design includes a safety chain that is attached at the front of the extension beam. This is shown in Figure 10.

Figure 10. Safety chain attachment current load cart.

2.7.2 Front Hitch Height Adjustment Jack

The current load cart design includes a jack that attaches at the front of the extension beam, shown in Figure 10. The jack is used to adjust the height of the front hitch when hooking up to the tow machine drawbar and supports the load cart while in storage. The current load cart uses a weld-on trailer jack with a flat plate at the foot. This does not aid in supporting the load cart during unloading from the trailer for transportation.

2.7.3 Hydraulic System

The current hydraulic system consists of four 14 ft hydraulic hoses, quick coupling inserts to insert into the towing machine hydraulic system, clamps, and two quick disconnect adapters. The hydraulic system is outlined in Figure 11.

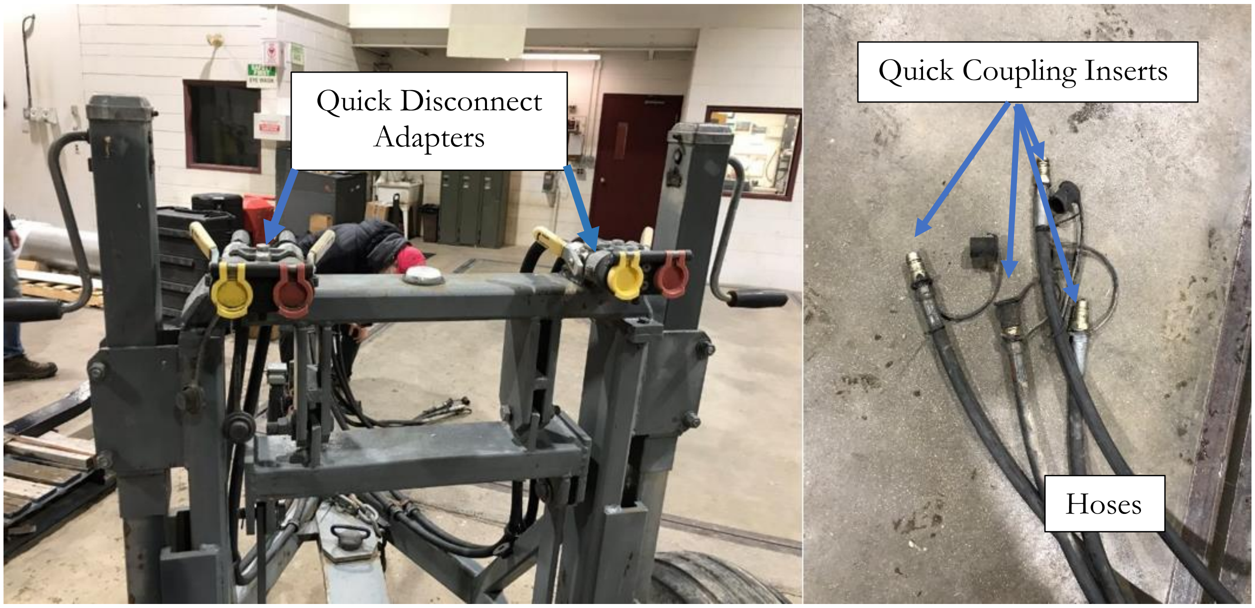


Figure 11. Hydraulic system components.

2.8 Test Setup

One possible load cart test setup consists of a towing machine and a cultivating implement connected to the front and rear hitch of the load cart, respectively. The goal of these tests is to compare the loads applied to the towing machine to the slip of the towing machine relative to the ground. The layout for testing is shown in Figure 12 where the drawbar is referred to as the towing machine hitch. The towing machines and implements used for testing vary in size, weight, and type of machine.

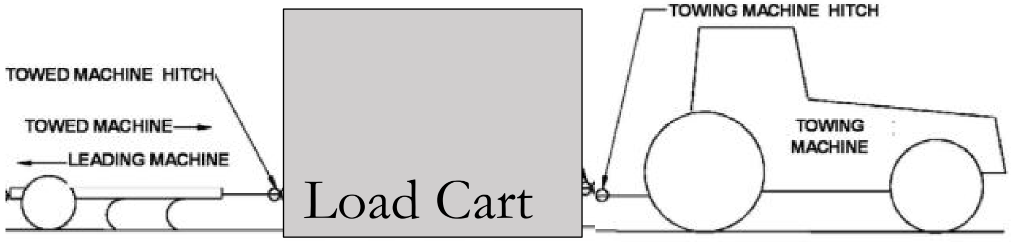


Figure 12. Load cart implemented in complete test setup. [2]

An example of a cultivating implement used for testing is shown in Figure 13. The testing load cart utilizes a load cell to track the force applied by the implement to the drawbar of the towing machine. The load is varied on the drawbar by increasing and decreasing the draft of the implement. “Draft is the total required force parallel to the direction of travel to pull the implement. Both functional draft (soil and crop resistance) and draft due to rolling resistance of the implement are included” [3]. On the cultivator implement the draft is increased by lowering the cultivator shovels further into the ground and subsequently the draft is lowered by raising the shovels.

Figure 13. Example of a cultivating implement used in the test setup. [4]

2.9 Problem Statement

Prairie Agriculture Machinery Institute (PAMI) requires a load testing cart that can accurately measure draft loads on a diverse range of towing machines and implements. The load cart must be able to follow the drawbar accurately and connect to drawbar hitches of category 3-5 at the towing machine and implement hitches.

The current solid axle load cart design does not allow for the wheels to follow the path of the load cart during turns. Adjusting the load cart for various hitch categories requires the removal and install of four bolts at the front and takes considerable time. There are several additional complications that exist with the current load cart including transportation of the load cart and the installation and removal process of the load cells.

Continuing with the current design will lead to inconsistencies in draft loading measurements and difficulties in the use of the system for PAMI employees. Agriculture Testing Solutions is tasked with designing a new load cart equipped with the following features:

- Compatible to category 3-5 drawbar hitches at the towing machine and category 3-5 at the implement
- Accurate tracking of the drawbar
- Ergonomic design for load cell installation

3 Project Definition

The following section presents a summary of the overall project definition. The project objectives are defined including the client deliverables and the design components that are considered outside the scope of the project. Next, a list of target specifications is outlined and given a numerical metric based on aspects of the design. Finally, the complete list of constraints and limitations of the project are presented.

3.1 Project Objectives

The objective of this project was to design a load cart that can accommodate a larger range of towing machines and implements with better load cell measurement accuracy than the current load cart design. This was achieved by meeting the following objectives:

- Integration of pre-purchased client parts and test equipment
- Increased field-testing performance
- Compatibility with a wide range of towing machines and implements
- Compliance with relevant standards outlined by PAMI and ASABE (American Society of Agricultural and Biological Engineers)
- Improved test setup time and ergonomics

The completed project deliverables required by the client are as follows:

- Computer Aided Design (CAD) models of the load cart
- Preliminary Finite Element Analysis (FEA) of the load cart
- Bill of materials

- Preliminary supplier and cost information
- Failure Modes Effect Analysis (FMEA)

The following design tasks were excluded from the scope of the project:

- The implementation of vertical load cells.
- The consideration of apparatuses other than a load cart for applied load measurements between the vehicle drawbar and the loading implement.

3.2 Target Specifications

The target specifications for the load cart are all the quantifiable parameters that were required for the successful final design. These parameters were split into two sections, client needs and metrics. The client needs were written based on requirements and necessary design features from the client. The metrics were then created from the client's needs and were given respective units of measurement with quantities where applicable. Both the needs and metrics are prioritized by importance on a scale of 1 through 5, with 5 being the most important.

3.2.1 Client Needs

The team defined several client needs that the load cart design had to satisfy. All the needs were directly defined by the client as being important for the success of the project. The 1-5 rankings allowed the team to prioritize the most important aspects of the design. The client's needs, priority rankings, and a description documenting the reasoning for each need are listed in TABLE I.

TABLE I:
SUMMARY OF CLIENT NEEDS

#	Client's Need	Priority	Description
1	The load cart can be transported on a trailer.	5	The client needs to transport the load cart using a trailer to various locations for different test conditions.
2	The load cart passes all necessary ASABE standards.	5	The load cart requires a safety chain across the tow machine hitch per ASABE standards. The hitch category and pin sizes are also defined by ASABE standards.
3	The load cart is compatible with different sizes of load cell.	5	20,000, 50,000 and 100,000 lb load cells provided by the client must be compatible with the load cart.
4	The load cart is height adjustable for varying drawbar heights.	5	The load cart must be height adjustable to accommodate drawbar heights ranging from 13-24 inches.
5	The load cart wheels follow the direction of travel of the load cart.	5	The load cart needs wheels that follow the motion of the cart to obtain accurate load cell measurements.
6	The load cart does not interfere with the towing machine or implement.	5	For the load cart to perform as intended in the field, the cart must not interfere with the tow machine or implement.
7	Load bearing designs must satisfy FEA simulations with a minimum Factor of Safety of 2.	5	To ensure the design will not fail the client requires a minimum safety factor of 2. Initial calculations will be done by hand, followed by a more detailed approach using Finite Element Analysis.
8	Load cell installation is ergonomic and accessible from outside of the load cart.	4	The load cell installation process is important for the client. Installation must be a quick ergonomic process done in the field.
9	The design results in accurate load cell measurements.	4	The load cell needs to be kept in line with the applied load from the implement to ensure accurate measurements.
10	The load cart has a front and rear hitch compatible with drawbar categories from 3-5 as per ASABE standards.	3	The front and rear hitch need to be versatile to allow the load cart to test various towing machines. The lower category hitch compatibility is no longer required.

#	Client's Need	Priority	Description
11	The load cart has four sets of standard pioneer quick couplers for hydraulics.	3	The load cart adds to the distance between the towing machine and implement and requires hydraulic hose extensions.
12	The load cart accurately measures compressive loads during braking.	2	The load cart has a potential future use measuring compressive loads for brake testing.

3.2.2 Metrics

Each client need from Section 3.2.1 corresponds to one or more technical specifications. The technical specifications were used to determine how well the client's needs were satisfied at the end of the project. Each technical specification consists of a metric and a value. TABLE II outlines the metrics with descriptions for the load cart redesign. The importance of each metric was assigned a rank (1-5) where 5 is the most important, and a unit used to measure the metric.

TABLE II:
LOAD CART METRICS AND UNITS

Metric #	Metric	Importance	Units	Min Value	Max Value
1	Maximum load cart width and length: Maximum allowable length and width to ensure the load cart can fit on a trailer pre-purchased by PAMI.	5	ft	—	Width: 7 Length: 18
2	Safety Chain can be implemented: The load cart must pass all necessary ASABE standards.	5	Pass/ Fail	—	—
3	Load Cart is Compatible with Three Different Load Cells: The load cart is compatible with PAMI's current 20,000 lbs, 50,000 lbs, and 100,000 lb load cells.	5	Pass/ Fail	—	—

Metric #	Metric	Importance	Units	Min Value	Max Value
4	Minimum and Maximum Adjustable Hitch Height: The maximum and minimum height the hitch must reach.	5	Inches	13	24
5	Caster Wheels can be Implemented: Pre-purchased Bourgault swivel caster wheels must be able to be integrated into the redesign.	5	Pass/ Fail	—	—
6	PowerPin can be Implemented: Pre-purchased PowerPin must be able to be integrated into the redesign.	5	Pass/ Fail	—	—
7	Clearance of the load cart: The minimum distance the load cart can be from any part of the tow machine while turning on a 50° angle.	2	Inches	3	—
8	Factor of Safety: The minimum factor of safety for all load cart components is 2, although a safety factor of 3 is desired.	5	Pass/ Fail	Minimum.: 2 Desired: 3	—
9	Load cell installation is ergonomic and accessible from outside the load cart: The load cell must be easily installed and removed by an employee from outside the cart in the field.	5	Pass/ Fail	—	—
10	Accurate load cell readings: Keeping the load cell in line with the applied load from the implement ensures accurate measurements of the load applied to the towing machine.	4	Pass/ Fail	—	—
11	Load Cart is Compatible with Category 3-5 Hitches: The height adjustability of the load cart is one of the metrics that will allow the load cart to be compatible with category 3-5 hitches. Additionally, attachment pin ranges from 1.25-2.75 inches.	5	Inches	1.25	2.75

Metric #	Metric	Importance	Units	Min Value	Max Value
12	Mass of the Trailer: The maximum mass of the load cart is governed by the current trailer PAMI has purchased. The maximum load that the trailer can support is 5000 lbs.	2	lbs	—	5,000
13	Four Quick Couplers are Implemented: The load cart adds to the distance between the towing machine hitch and the implement and requires hydraulic hose extensions.	5	Pass/ Fail	—	—
14	Maximum applied load: The maximum load applied to the load cart is dictated by the largest load cell PAMI will be implementing.	5	lbs	—	100,000

3.3 Constraints and Limitations

Based on the specific scope of this project, ten limiting constraints were identified by the team. The list of constraints and limitations in this project are summarized in TABLE III. The description in the right-hand column of TABLE III justifies why the constraint was included, and any information that needed to be considered in the load cart redesign.

TABLE III:
LIST OF CONSTRAINTS AND LIMITATIONS

#	Constraint	Value	Description
1	Project Timeline	14 weeks	• Determined by MECH 4860 course. • Internal schedule developed to meet both client and class deliverables.
2	Project Budget	$10,000	• The project budget determined by the customer is $10,000. This includes materials and labour. The project budget will limit the design options for the load cart.

#	Constraint	Value	Description
3	Incorporate Bourgault swivel casters	Fixed	• The Bourgault swivel casters were pre-purchased by PAMI to incorporate in the design. • The swivel casters do not have CAD models or manufacturing drawings
4	Fit on a 7ft x 18ft trailer	N/A	• The maximum length and width of the trailer provide a dimensional constraint.
5	Incorporate three load cells	Fixed	• Three load cells (20,000, 50,000 and 100,000 lb) were pre-purchased by PAMI to incorporate in the design. • Incorporating the load cells restricts the consideration of smaller and lighter options for measuring the load.
6	Incorporate PowerPin	Fixed	• The PowerPin was pre-purchased by PAMI to incorporate in the design.
7	No interference with the rear wheels of the towing machine while turning	N/A	• The turning radius of towing machines that are tested with the cart will affect the overall shape and dimension of the cart.
8	Source material from local vendors	Fixed	• The use of local vendors will limit the types of materials and variety of parts that can be implemented on the load cart.
9	Source materials in standard sizes	Fixed	• Standard size materials will limit the design process an could cause areas of over-engineering due to excess material.
10	Minimum factor of safety of 2 for the design.	Fixed	• A factor of safety of 3 is optimal however a factor of safety of 2 is acceptable to the client to reduce cost.

4 Concept Generation

For initial concept generation the design problem was separated into six sections that were combined into a final design. These design aspects were front hitch integration, frame geometry, load cell integration, PowerPin integration, caster wheel mounting, and height adjustability. Certain concept selections were dependent on other sections, meaning that one concept selection narrowed down the feasibility of concepts for another section.

4.1 Concept Weighting Criteria

The two main concept weighting criteria were manufacturability and cost, they used in all designed sections of the load cart for concept selection.

- **Cost:** Cost is a crucial factor to consider for selection criteria. The total project budget is $10,000 for all materials and manufacturing. Cost will be considered in all comparison matrices throughout this concept development and selection sections.
- **Manufacturability:** The manufacturing of the load cart will be completed by PAMI in house. Therefore, it is important to be conscious of the manufacturing capabilities of PAMI's shop. The load cart components should use basic manufacturing processes where possible. Manufacturability will be considered in all comparison matrices throughout the concept development and selection sections.

Other weighting criteria were used for select sections of the load cart included:

- **Strength**
- **Weight**
- **Compatibility with pre-purchased parts**
- **Ease of use**

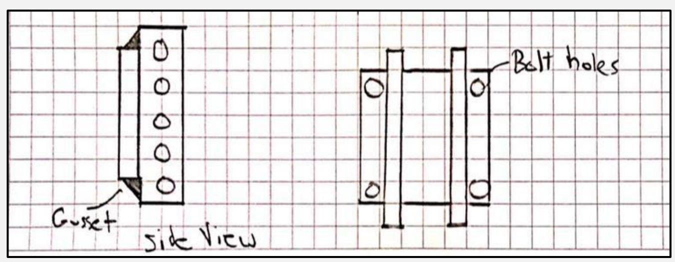
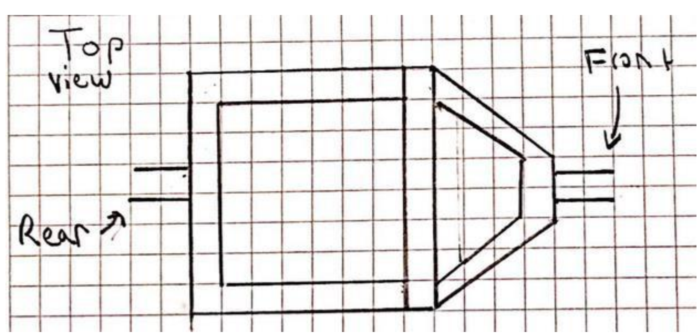
- **Compatibility with other sections of the load cart**

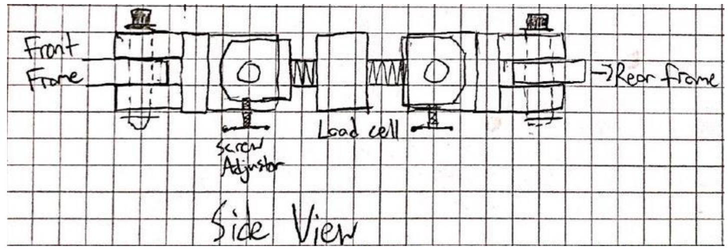
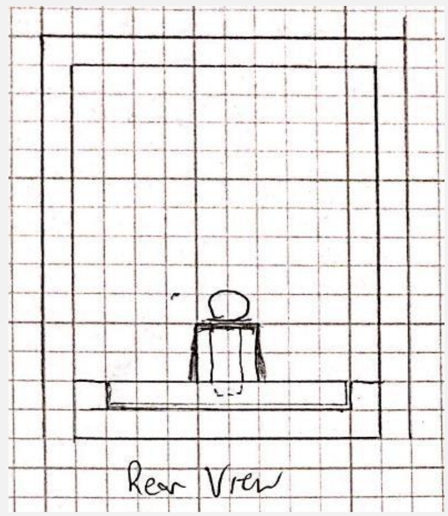
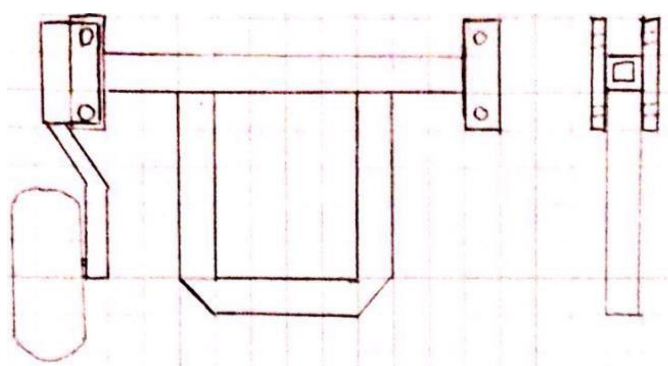
These criteria are described in detail for each section of the load cart they pertain to in Appendix A.

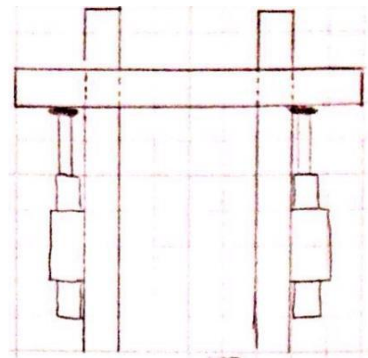
4.2 Overall Concept Selection Results

A detailed concept generation, weighting and scoring process can be found in Appendix A. This process resulted in the final concepts selected for the six main aspects of the design. This includes the front hitch integration, frame, load cell integration, PowerPin integration, caster attachment and height adjustability. The results of the selection process for each design aspect are presented in TABLE IV.

TABLE IV:
FINAL CONCEPT SELECTION SUMMARY

Design Aspect	Selected Concept	Image of Concept
Front Hitch Attachment	Multi-hole design	
Frame	A-frame	

Design Aspect	Selected Concept	Image of Concept
Load Cell Integration	Vertical Install	
PowerPin Integration	Bearing Plate	
Caster Attachment	Horizontal Beam	

Design Aspect	Selected Concept	Image of Concept
Height Adjustment	Outer Guide Beam	

The six selected concepts are combined into a testing load cart in the detailed design phase.

5 Detailed Design

The following section outlines each component of the redesigned load cart. The components are split up into eight different sections: swivel casters, main frame, horizontal support, manual jacks, PowerPin, PTFE bearings, load cell integration, and front hitch connection. The eight sections of the load cart are shown in Figure 14 and Figure 15.

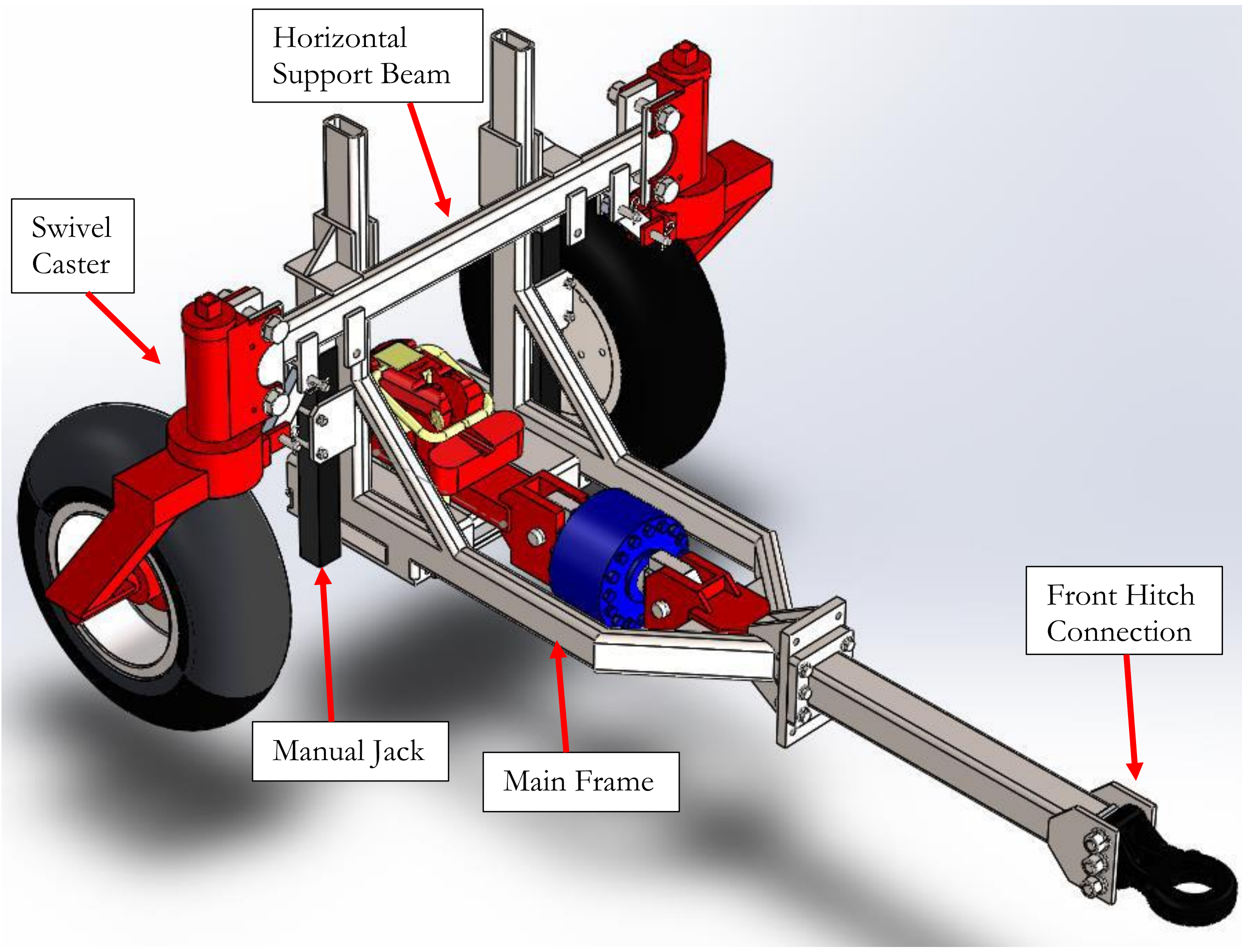


Figure 14. Overview of final design (1).

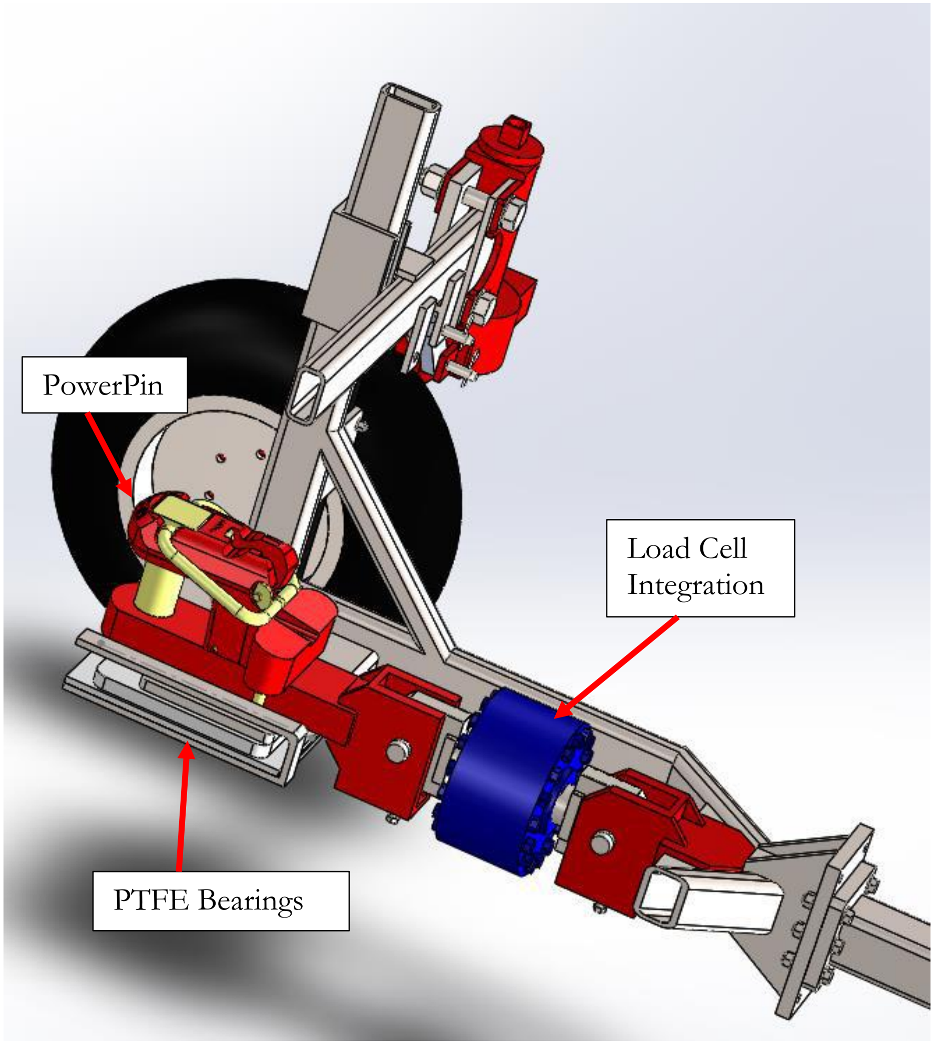


Figure 15. Overview of final design (2).

For each section, the details of the design features will be explained and displayed using 3D models, and model renders. The constraints applicable to each design feature will be outlined, along with a description of the needs met by the design feature.

5.1 Swivel Casters

PAMI has purchased two Bourgault caster wheel assemblies that must be mounted onto the load cart as per client need #5. This section describes how the dimensions and motion of the casters will affect other design components due to their size and freedom of motion. Design alterations made to the casters to prevent interference with the load cart during operation are also explained.

5.1.1 Background and Constraints

The two pre-purchased Bourgault casters are approximately 40 inches tall with 30-inch diameter wheels installed. The casters have three main components, the mounting bracket, pivot point, and wheel connection illustrated in Figure 16.

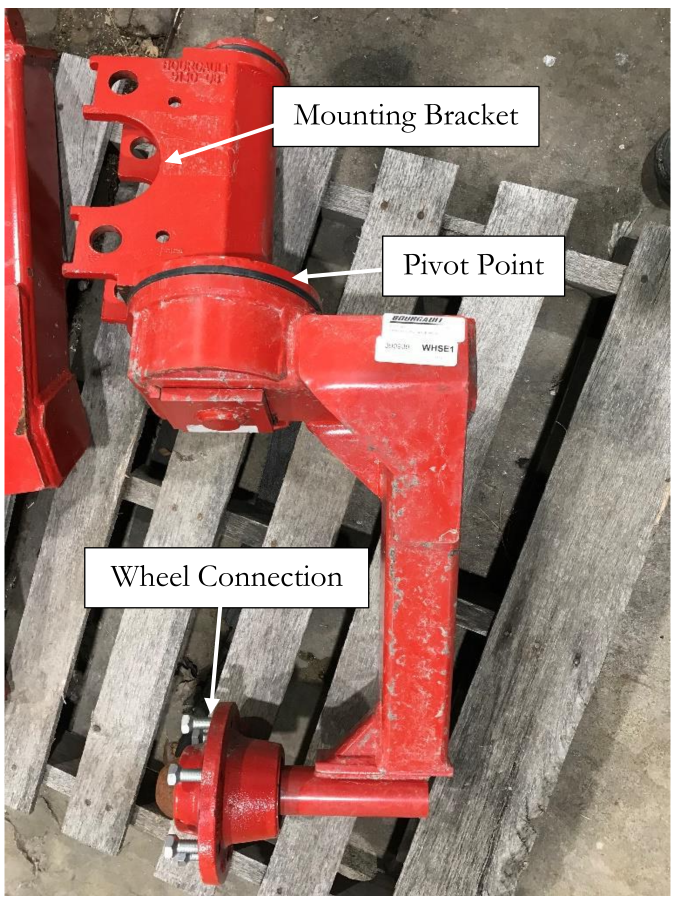


Figure 16. Swivel caster components.

Using the swivel casters will allow the wheels to follow the motion of the load cart. This eliminates the previous issue of the load cart wheels skidding during turns.

A radius of 24 inches from the pivot point of the caster is needed to avoid interference between the wheel and the load cart frame or implement. This is shown in Figure 17 with a top view of a 90-degree rotation of the caster wheel. The casters must be mounted away from the frame due to the clearance required by the caster wheels.

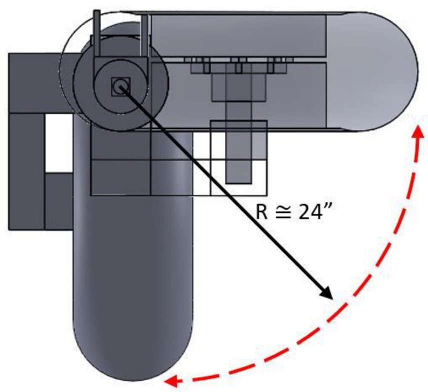


Figure 17. Required radius of clearance to avoid interference with casters.

The swivel casters were integrated keeping in mind the following constraints:

- **Constraint #3:** Design must incorporate Bourgault swivel casters

5.1.2 Design Features

The rotation of the wheels must be restricted to avoid interference with the implement or load cart due to the large clearance required to rotate 360 degrees. Figure 18 shows a top view of the design with the wheels rotated to the maximum allowable angle of 45 degrees while moving forward. The frame dictates the maximum allowable angle for load cart turns and does not require any added mechanisms to stop the motion of the casters.

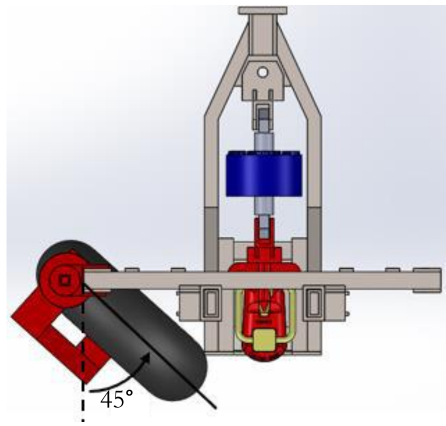


Figure 18. Maximum turning angle of swivel casters.

Interference with the frame is problematic and is most likely to occur when backing up the load cart. The caster wheels naturally swivel backwards when the load cart moves forward as shown in Figure 19 A. When the load cart begins moving backwards in Figure 19 B, the caster interferes with the cart before reaching the final position in Figure 19 C.

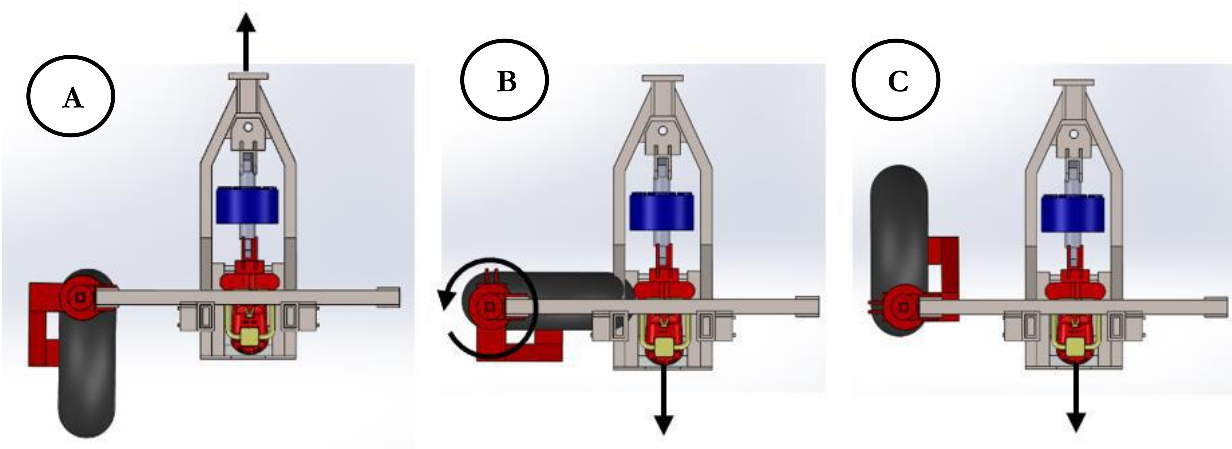


Figure 19. Caster orientation with the load cart moving forward (A) then backward (B and C).

A locking mechanism has been implemented to lock the caster in the position shown in Figure 19 A. This locking mechanism prevents the casters from reversing direction when backing up the load cart. The two positions of the locking mechanism are displayed in Figure 20. The locking mechanism consists of a 1" thick bar with a slotted hole of 0.75-inch diameter on either end. In

Figure 20 A, the bar is pinned to a 0.5” thick tab which is welded onto the beam. This allows the caster to rotate freely. Figure 20 B shows the caster wheel in the locked position with the locking mechanism connected between the fixed horizontal support and the rotating section of the caster wheel. The connection is made with a pair of tabs welded on the caster wheel assembly.

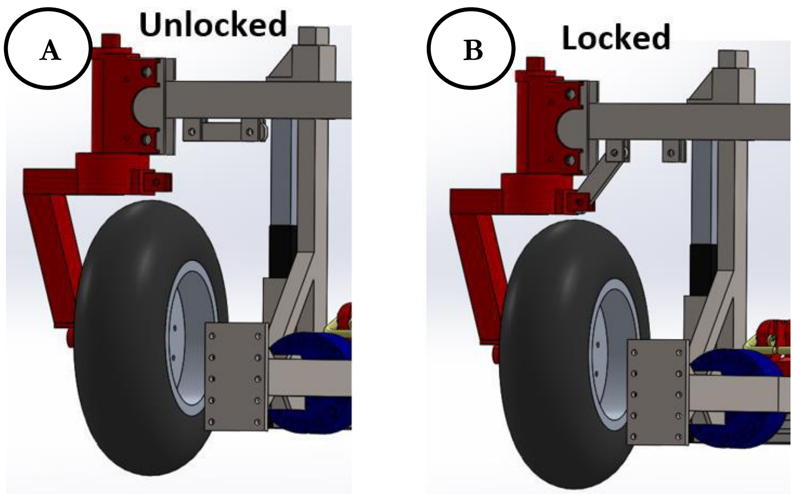


Figure 20. Caster locking mechanism in the unlocked and locked or stow-away position.

The CAD models include all details necessary to properly weld the square tabs onto the casters. The CAD models will be submitted to the client at the end of the course.

5.1.3 Design Assessment

The new swivel casters directly address several needs outlined by the client.

- **Need 5:** The installation of swivel casters allows the load cart wheels follow the direction of travel of the load cart.
- **Need 7:** The new design aspects of the swivel caster assembly satisfy a minimum safety factor of 3 in FEA simulations and will be outlined in Section 8.

5.2 Horizontal Support

The horizontal support allows the swivel casters to connect to the frame with adequate clearance while maintaining the height adjustability of the load cart. The main constraints of the horizontal support are discussed in this section. Following constraints, the three defining features of the support are split into the following sections: the horizontal beam, caster mounting, and support sliders. These three features are displayed in Figure 21.

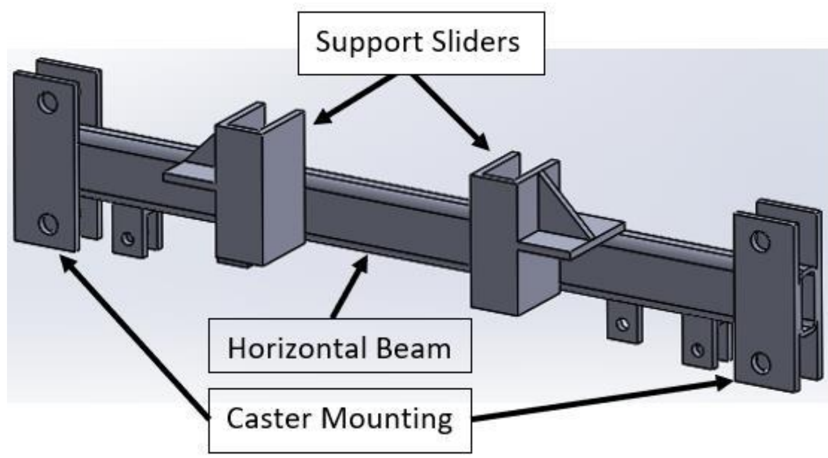


Figure 21. Horizontal support features.

5.2.1 Background and Constraints

The main constraint for the horizontal support is integrating the swivel casters (Constraint #3) while keeping adequate clearance between the casters and the frame. The new design allows the swivel casters to rotate a maximum of 45 degrees in either direction.

Another limitation with the caster mounting is ensuring the downwards load from the implement at the rear hitch is applied in front of the wheels. Having the load in-between the front hitch and the wheels will guarantee all forces transferred to the towing machine hitch are downwards. This imitates forces from regular operation of an implement.

5.2.2 Design Features

Rectangular Beam

A hollow rectangular tube was selected for the horizontal beam. The tube has a height of 4 inches, width of 2 inches, and wall thickness of 3/8th of an inch. This sizing was chosen as it is common and can also be used for the main frame tubing. By using one size, only one long span of tubing needs to be purchased and cost will be reduced.

The horizontal beam stretches a total of 55 inches for 16.5 inches of clearance on both sides of the 22-inch-wide frame. This distance allows the caster wheels to rotate 45 degrees in both directions. The beam is also mounted in front of the two vertical supports. This orientation of the beam works in conjunction with the main frame to withstand the bending moment in the beam created by the drag forces on the wheels while the load cart is moving. The setup for the horizontal beam is shown in Figure 22.

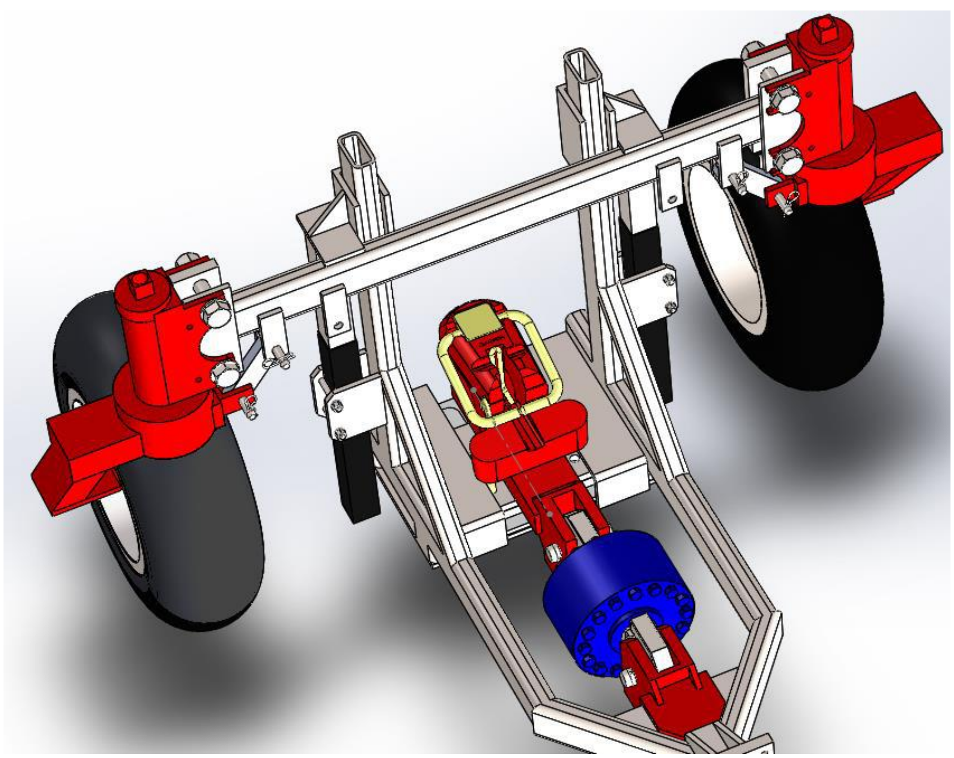


Figure 22. Horizontal beam position on frame.

Caster Mounting

Bolts were used to mount the casters to the rectangular beam. As shown in Figure 23, one 0.5-inch thick plate, and one 1-inch thick plate will be welded on to either end of the rectangular beam with holes that match with the holes on the top of the casters. The caster mounting brackets are slid around the two plates and two bolts are inserted through the holes and fastened with nuts on the opposite side to hold the caster in place.

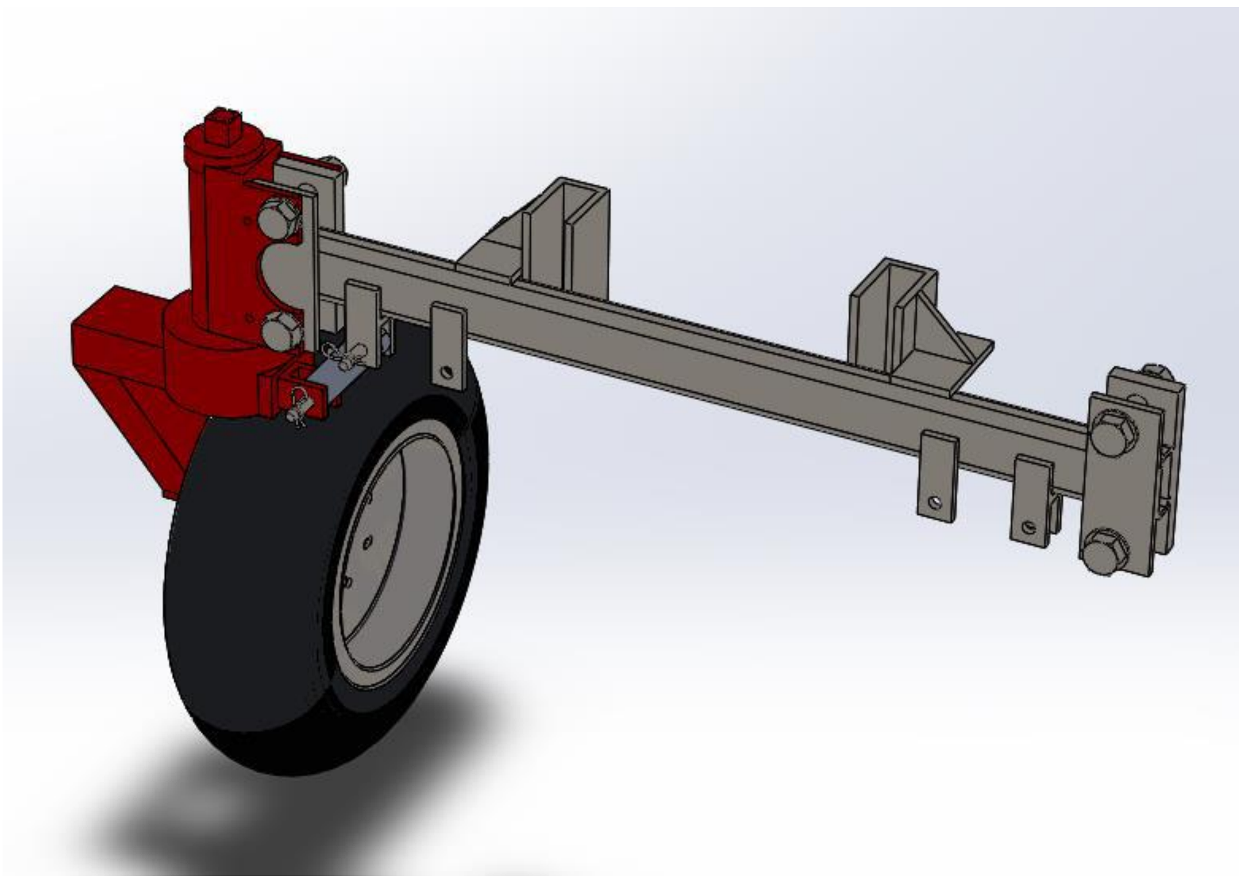


Figure 23. Caster mounting layout.

With the casters and wheels added the total width of the load cart from the outside of each caster is 6 feet 7 inches, 5 inches less than the maximum 7-foot width of the trailer.

Beam Height Adjustment

To allow the frame to move up and down, a custom rectangular tube is welded on to the horizontal beam that fits around the vertical supports. This tube is made from 0.5-inch steel plating with a height of 8 inches and is shown in Figure 24.

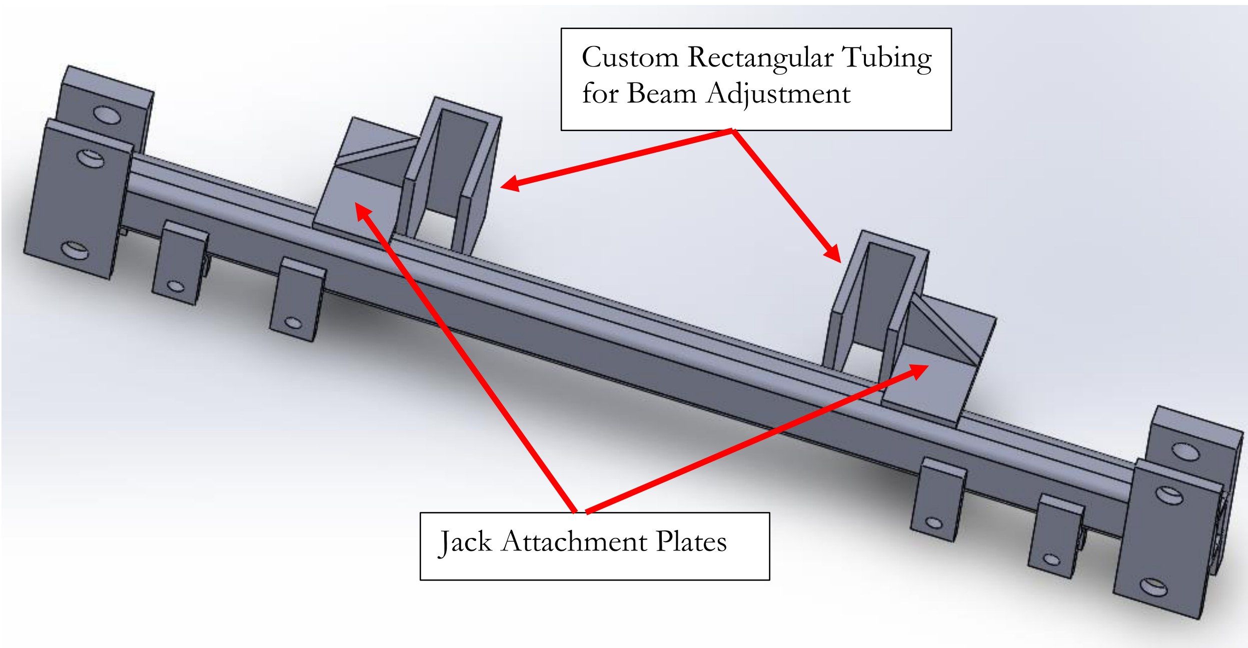


Figure 24. Horizontal beam height adjustment features.

Figure 24 also shows a square plate welded to the side of the custom tube and on top of the horizontal beam. This plate is then welded to the foot of the jack which can adjust the frame vertically. The plate has a reinforcing gusset for added support against vertical loads. Additionally, the jack support was placed strategically at the outside of the vertical support to avoid interfering with the frame or the power pin.

5.2.3 Design Assessment

- **Need #4:** The horizontal support beam has support sliders that mount to manual jacks. The jacks move the horizontal support beam relative to the vertical supports. This permits the load cart to have height adjustability.

- **Need 7:** All load bearing aspects of the horizontal support assembly satisfy a minimum safety factor of 3 in FEA simulations and are outlined in Section 8.

5.3 Main Frame

All design details related to the main frame will be provided in this section. This will include a description of corresponding needs, constraints and background information, followed by a detailed description of all five major design components of the main frame. Finally, a summary of the complete main frame design will be presented.

5.3.1 Background and Constraints

Many of the general main frame features on the current load cart have been incorporated into the main frame redesign. This includes the front hitch extension connection, A-frame base, vertical members with angled supports, and similar jack mounts. The rear hitch connection has been modified in the main frame redesign to incorporate a PowerPin rear hitch attachment.

The new main frame was designed to accommodate the following constraints:

- **Constraint #4**: Fit on a 7ft x 18ft trailer
- **Constraint #5**: Incorporate three load cells
- **Constraint #6**: Incorporate PowerPin
- **Constraint #7**: No interference with the rear wheels of the towing machine while turning
- **Constraint #8**: Source materials in standard sizes
- **Constraint #10**: Minimum factor of safety of 2 for the design

5.3.2 Design Features

The main frame consists of six main components as labeled in Figure 25. An A-frame is used at the base with a connection point for the front hitch extension where the front two frame members meet. On the top of the base frame at the rear of the load cart are two vertical supports, each with angled supports for added lateral strength. The vertical supports also have mounting points for jacks that are further explained in this section. Below the base frame at the rear is the mounting system for the PowerPin.

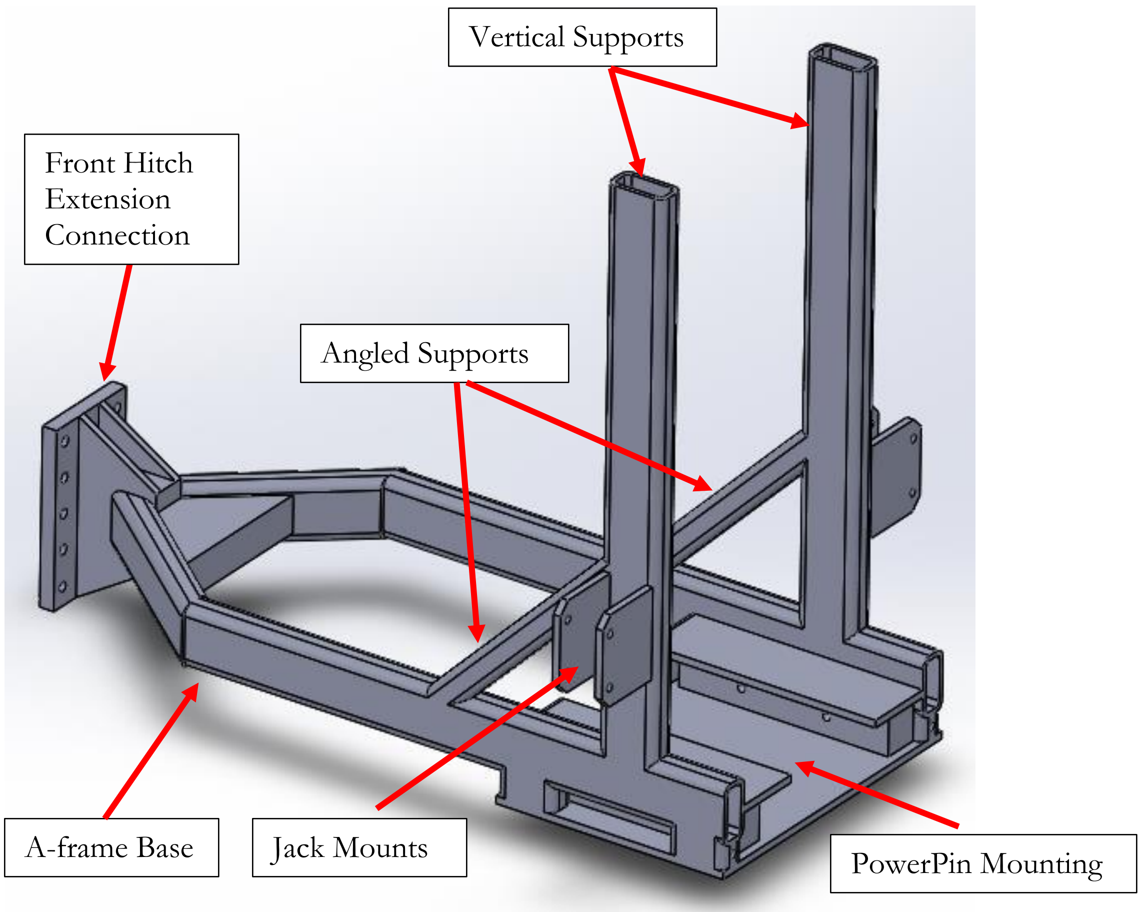


Figure 25. Redesigned load cart main frame features.

Front Hitch Extension Mounting Plate

The front mounting plate attaches to the front hitch extension beam. The front hitch extension beam provides distance between the tow vehicle and the load cart to prevent contact. The extension beam from the current load cart design is unchanged and has been integrated into the re-design. The extension beam attaches to the mounting plate on the main frame displayed in Figure 26.

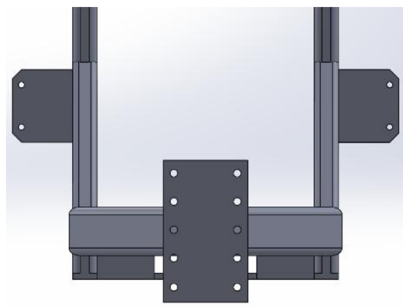


Figure 26. Front hitch extension mounting plate.

The main frame mounting plate consists of a bolt pattern with twelve 5/8" holes that allow height adjustability. From the middle position the beam can be adjusted 2.34" higher or 2.34" lower. The extension beam is 30" long with plates welded on the front sides and back. The back plate is welded perpendicular to the cross section of the beam. The back plate allows the beam to attach to the main frame mounting plate. The beam is bolted to the mounting plate using six 5/8" bolts. The front plates are welded to the sides of the beam at the opposite end of the rear plate. These plates have holes drilled in them to integrate the front hitch assembly. The entire extension beam is displayed in Figure 27.

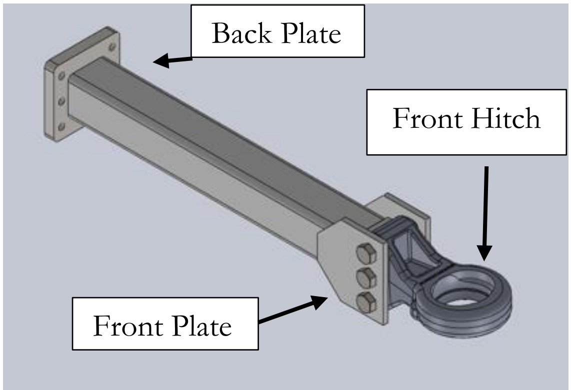


Figure 27. Extension beam assembly.

The detailed dimensions of the extension beam and main frame mounting bracket are outlined in the CAD models that will be submitted to the client at the end of this course.

A-Frame Base

The A-frame shape replicates the basic shape of the current load cart design. The A-frame shape incorporated into the final main frame design is displayed in Figure 28.

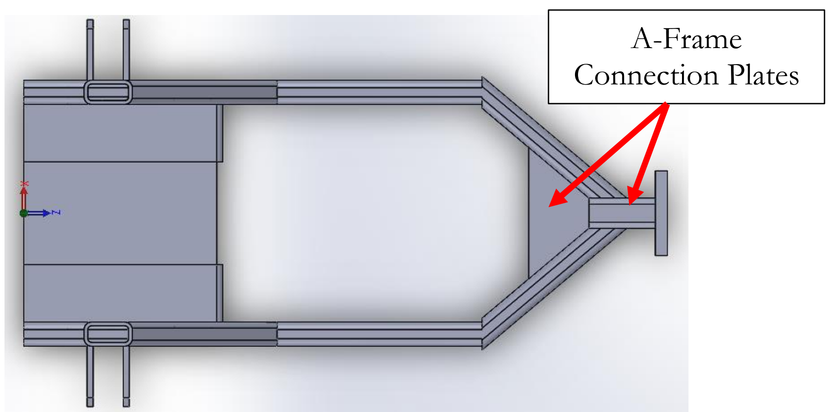


Figure 28. A-frame design.

The A-frame connection plates shown in Figure 28 are made from repurposed steel from the PowerPin drawbar. The PowerPin drawbar is 5” wide and 2.5” thick. A piece of the drawbar was manufactured to fit into each position. The interior of the frame has a width of 18” for the installation of the load cell. This spacing allows for an ergonomic load cell installation. The frame members consist of 2" x 4" square steel tubing that is 3/8" thick. The detailed dimensions of the A-frame are provided in the CAD models that will be submitted to the client at the end of this course.

Vertical Supports

There are two vertical supports at the rear of the frame, as displayed in Figure 29. The vertical beams are used to support the horizontal support beam. The horizontal support beam is height adjustable and moves relative to the vertical supports. The height of the vertical supports is determined by the minimum and maximum height requirements of the drawbar defined by ASABE standards for the 3 hitch categories.

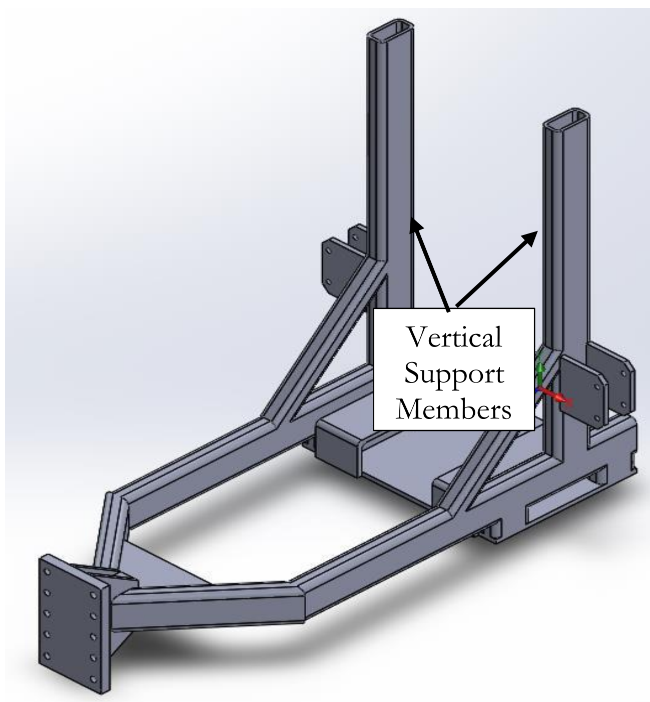


Figure 29. Vertical support beams with caster wheel attachment beam.

The vertical supports are made from 2" x 4" square tubing with a 3/8" wall thickness. The total height of each vertical support is 36.5". The detailed dimensions of the vertical support beams are provided in the CAD models that will be submitted to the client at the end of this course.

Angled Supports

Angled supports were welded to the base frame and vertical supports shown in Figure 30. These angled members provide additional support for the caster wheel attachment beam. The angled support member length was set by the minimum and maximum height requirements of the drawbar per ASABE standards. The maximum height of the load cart is 26" and occurs when the caster wheel attachment beam contacts the angled supports.

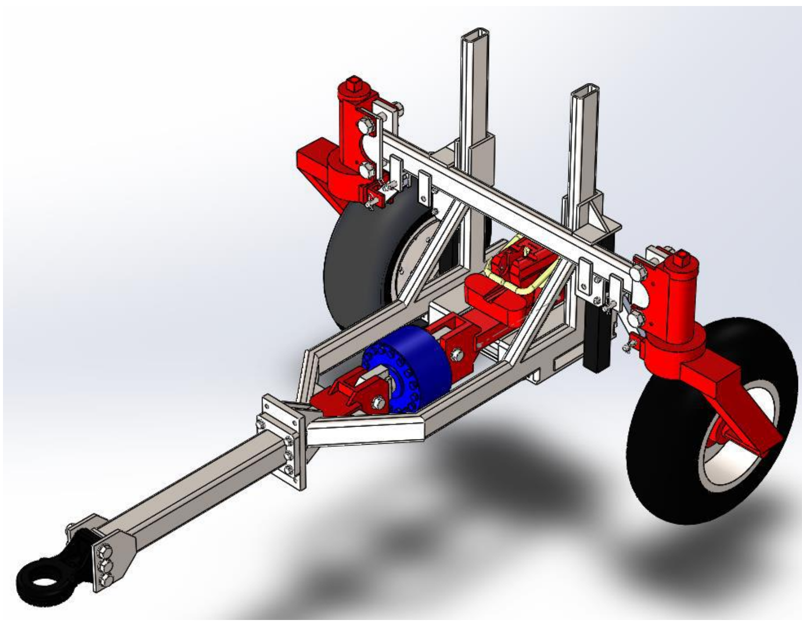


Figure 30. Position of caster wheel extension beam for maximum load cart height with angled supports.

The range of motion of the caster wheel attachment beam limited the attachment point of the angled support to a height of 12 inches up the vertical support member. The angled supports are made from 2" x 2" square tubing with a 1/4" wall thickness. The total length of each angled support is 17". The detailed dimensions of the angled support beams are provided in the CAD models that will be submitted to the client at the end of this course.

Jack mounts

Mounting plates for the jacks are positioned on the outside of the vertical support beams. The jacks are used to control the height of the load cart and are mounted using the mounting plates. The jacks are positioned upside down relative to the standard jack orientation. Figure 31 displays how the jacks are bolted to the mounting plates on the vertical supports. A detailed breakdown of the jack mount design is found in Section 5.4.

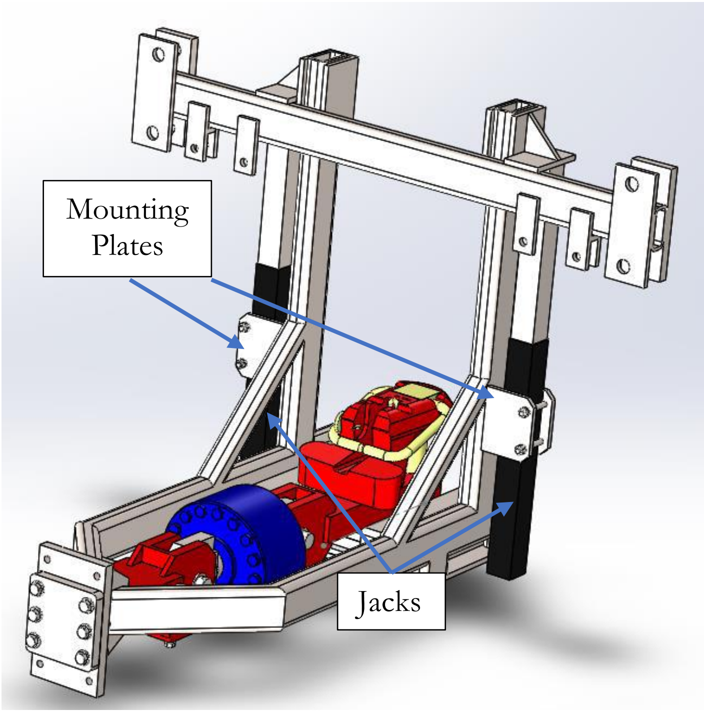


Figure 31. Main frame jack mounting assembly

The detailed dimensions of the jack mounts and jack are provided in the CAD models that will be submitted to the client at the end of this course.

PowerPin Mounting

The PowerPin is mounted at the rear of the load cart and is supported by a half inch thick steel plate. To permit space for the entire PowerPin assembly, the steel plate is welded onto four square tube members that are 1.75 inches below the bottom of the main frame. The PowerPin support plate is displayed in Figure 32. A detailed breakdown of the PowerPin mounting design is found in Section 5.5.

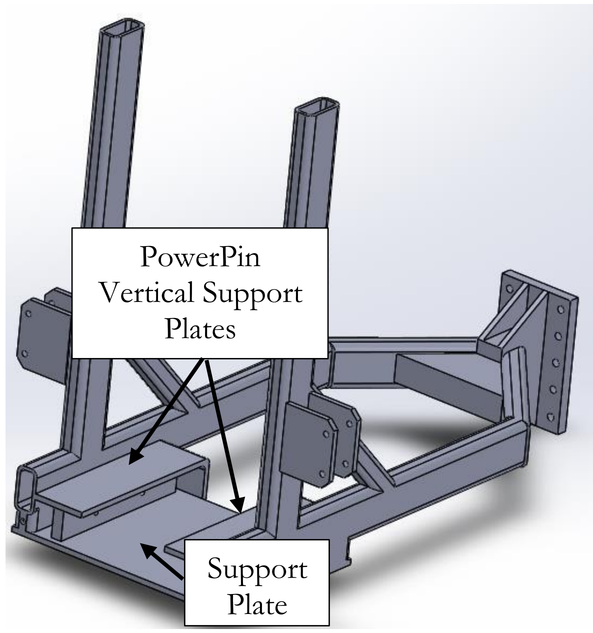


Figure 32. PowerPin mounting area on main frame

The PowerPin vertical support plates are welded to the main frame. These plates are installed to support the PowerPin assembly from vertical loading caused at the connection to the implement. The detailed dimensions of the PowerPin mounting sections of the main frame are provided in the CAD models that will be submitted to the client at the end of this course.

5.3.3 Design Assessment

The new frame directly addresses five of the needs outlined by the client.

- **Need #3:** The load cart frame was designed so that the three different load cells can be installed ergonomically into the frame without interference.
- **Need #4:** The front hitch extension mounting plate on the main frame allows independent height adjustment of the front hitch extension. Additionally, the jack mounts allow manual jacks to be mounted to the frame to adjust the height of the load cart wheels.

- **Need #6:** The front hitch extension beam provides distance between the tow vehicle and the load cart to prevent contact.
- **Need #7:** The length and width of the load cart frame minimizes the likelihood of interference with the towing machine and implement.
- **Need #8:** All load bearing aspects of the frame satisfy a minimum safety factor of 2 in FEA simulations and is outlined in Section 8.
- **Need #9:** The main frame shape is similar to the current load cart design and does not present any accessibility issues for the installation of the load cell.

5.4 Manual Jacks

Two manual weld-on trailer jacks will be used in the redesign for adjusting the load cart height similar to the current load cart design. The method of mounting the jacks to the frame and the specifications of the jacks are outlined in this section.

5.4.1 Background and Constraints

A mechanism must be incorporated into the new design to achieve the load cart height adjustability requirements. These requirements are defined by the client to be a minimum of 15 inches and a maximum of 23 inches, measured from the ground to the mounting location on the hitch.

5.4.2 Design Features

A jack is mounted on the outside of each vertical support with the sleeve of the jack on the bottom and the foot of the jack pointing up. A labeled image of the selected jack is displayed in Figure 33.

[redacted]

Figure 33. Labeled Powerfist 5,000 lb weld-on drop leg trailer jack. [5]

The foot of the jack is welded to a plate that connects to the horizontal beam allowing the jack to raise and lower the load cart. The jack is positioned to exceed the maximum and minimum frame height necessary to achieve the desired height adjustability. Figure 34A shows the jack at the minimum position and Figure 34B shows the jack maximum position.

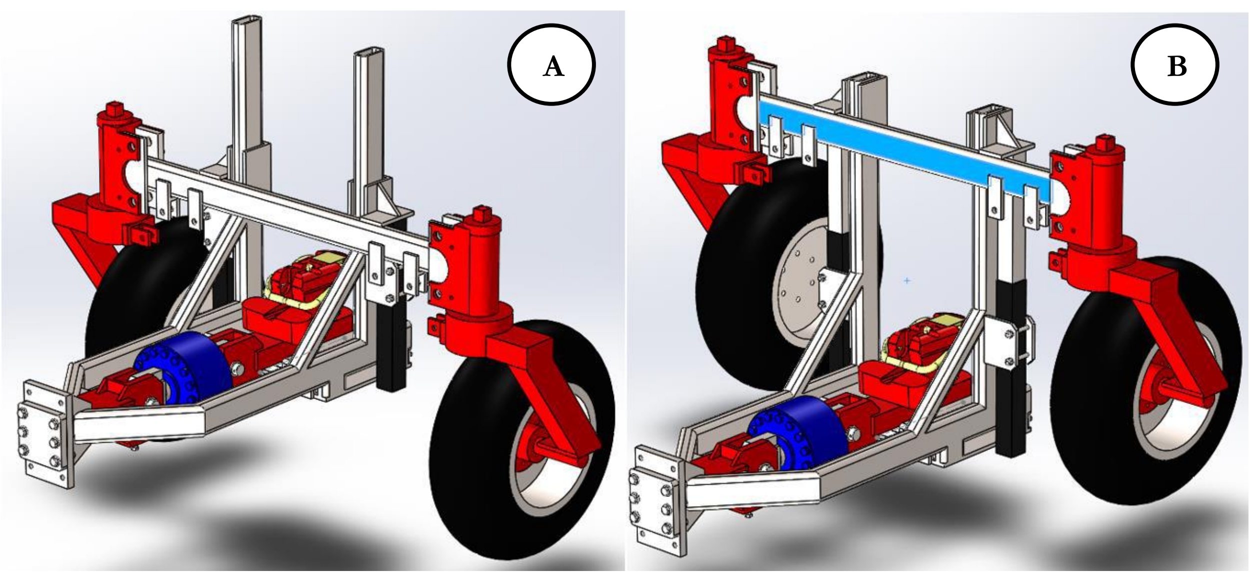


Figure 34. Load cart maximum and minimum height positions

The method of fixing the jack to the load cart is the same method used on the previous load cart design. Two plates are welded to the frame and the jack is inserted between the plates. Two bolts are then used to clamp the plates onto the jack. To prevent the jack from sliding, two steel tabs can be welded onto the jack above the plates to counter any vertical forces on the jacks. The setup of this mounting system is shown in Figure 35.

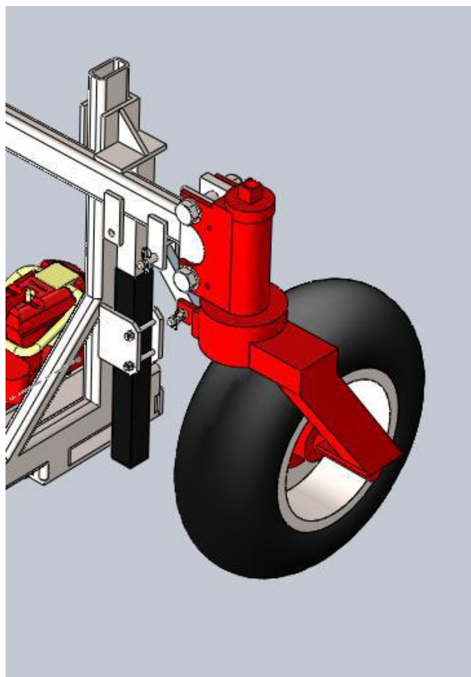


Figure 35. Jack mounting system setup.

The jack selected is a Powerfist 5,000 lb weld-on drop leg trailer jack. The specifications of the jack are displayed in TABLE V.

TABLE V:

WELD-ON DROP LEG TRAILER JACK SPECIFICATIONS

Minimum Height	**23.5 inches**
Maximum Height	52.5 inches
Maximum Load Capacity	5,000 lbs
Handle	Top wind

The jacks allow the height of the load cart to be adjusted between of 13" and 26", exceeding the clients requirements.

5.4.3 Design Assessment

The integration of manual jacks directly addresses the following need specified by the client.

- **Need #4:** The installation of manual jacks allows the load cart to be height adjustable. This feature is used to accommodate varying tow vehicle drawbar heights.

5.5 PowerPin (Rear Hitch)

PAMI purchased a PowerPin drawbar system for the rear hitch of the load cart to accommodate category 3-5 hitches. This system connects to an implement at the rear of the load cart and transfers draft loads to the load cell.

5.5.1 Background and Constraints

The PowerPin drawbar that PAMI purchased to be implemented into the load cart redesign is shown in Figure 36.

Figure 36. PowerPin drawbar.

The PowerPin drawbar provides a safer system for connecting the load cart to an implement. The hitch point is designed to automatically drop the pin into place when the implement hitch triggers a mechanical lever in the clevis portion of the PowerPin. PAMI has allowed the team to modify the drawbar so that it will fit in the design space of the load cart, but the clevis and automatic pin dropping system were kept intact.

The rear hitch point is located in the same position as the previous load cart to maintain the same hitch location that has worked for PAMI in previous tests.

The new rear hitch design was also restricted by the following constraints:

- **Constraint 2:** Project budget
- **Constraint 5:** Incorporate three load cells
- **Constraint 6:** Incorporate PowerPin
- **Constraint 8:** Source material from local vendors
- **Constraint 9:** Source materials in standard sizes
- **Constraint 10:** Minimum factor of safety of 2 for the design

5.5.2 Design Features

The rear hitch connection PowerPin drawbar system consists of four major components, the PowerPin, a PTFE sheet, a steel base plate, and the load cell connection point illustrated in Figure 37.

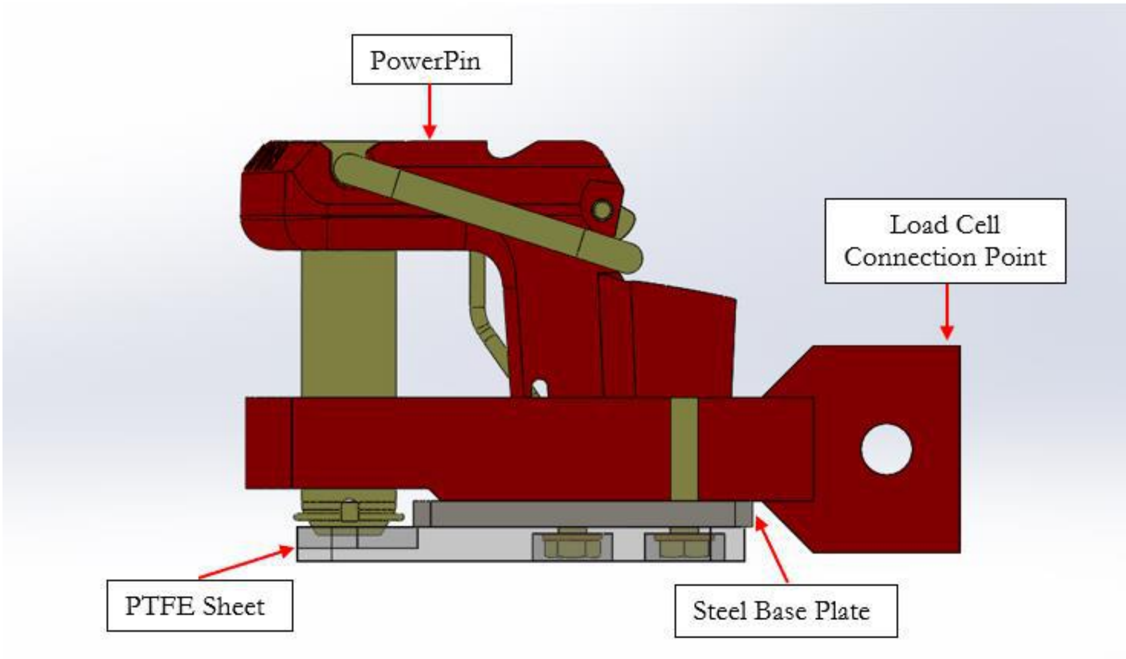


Figure 37. Labelled rear hitch connection.

Three major modifications must be made to the PowerPin drawbar purchased by PAMI for the design. First, the drawbar is cut just behind the clevis section of the PowerPin, 18" from the end to allow it to fit on the small frame of the load cart and accommodate a system to mount the three different load cells. The location of this cut is shown in Figure 38.

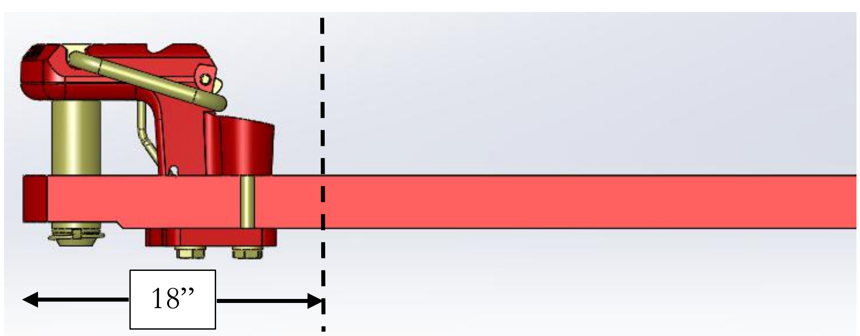


Figure 38. First cut location on the PowerPin drawbar.

The second change is the inclusion of the load cell connection point, which allows for vertical installation of the load cell, which then connects directly to the rear hitch. Figure 39 shows how the PowerPin integrates with the load cell.

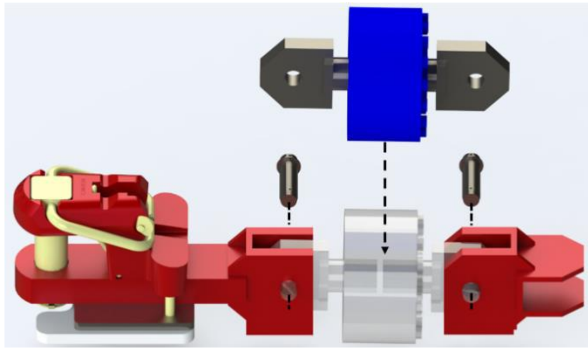


Figure 39. PowerPin load cell integration.

The original bottom plate on the PowerPin was switched for a custom steel base plate that can distribute the vertical load experienced by the PowerPin more evenly over the PTFE. The steel plate is 1" thick with a length and width of 9.84" and 10.5", respectively. Finally, the base of the PowerPin clevis was modified to incorporate a 1" thick, 11.5" x 13" PTFE bearing sheet. Both plates accommodate the original bolts used by the PowerPin clevis, with holes drilled through both as well as machined areas to accommodate the bolt heads and main rear hitch pin in the PTFE sheet shown in Figure 40.

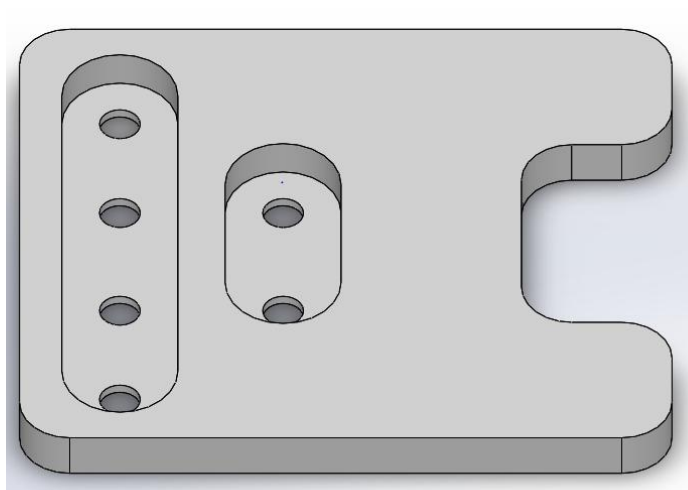


Figure 40. PTFE sheet machined sections.

These features ensure that the bolt heads and main rear hitch pin will not contact the PTFE on the main frame.

5.5.3 Design Assessment

The new rear hitch point design addresses four of the client needs:

- **Need 3:** The load cell connection point on the PowerPin drawbar system is designed to attach to three different sizes of load cell.
- **Need 8:** All load bearing aspects of the PowerPin assembly satisfy a minimum safety factor of 2 in FEA simulations and are outlined in Section 8.
- **Need 10:** The PowerPin load cell connection point allows the load cell to maintain alignment with the drawbar for accurate measurements.
- **Need 11:** The rear hitch is compatible with drawbar categories 3-5 as per ASABE standards.

5.6 Rear Hitch Support (PTFE Bearings)

One of the areas of the load cart that was not constrained by pre-purchased parts was the support system for the rear hitch. The team investigated the current load cart support and other methods for hitch support. The PTFE was selected due to its low coefficient of friction at high compressive loads. PTFE is a proven method for bearing applications in the agriculture industry and integrates well into our project needs. The selection of PTFE bearings was a suggestion from the client based on experience in the agriculture industry.

5.6.1 Background and Constraints

Due to the robust nature of the PowerPin drawbar and the need to accommodate category 3-5 towing machines the hinged design was ruled out in the redesign. The PTFE bearing plate design prevents the hitch from rotating, or moving vertically, improving the installation and removal of the load cell.

The new design was limited by the following constraints:

- **Constraint 2:** Project budget
- **Constraint 6:** Incorporate PowerPin
- **Constraint 8:** Source material from local vendors
- **Constraint 10:** Minimum Factor of Safety of 2 for the design

5.6.2 Design Features

Two different sizes of PTFE sheets were used for the design of the bearings. 0.5" thick sheets were used on the main frame to enclose the PowerPin Plate, while a 0.5" thick PTFE sheet was used beneath the PowerPin base plate. This design results in PTFE sliding on PTFE, reducing friction preventing any metal on metal sliding.

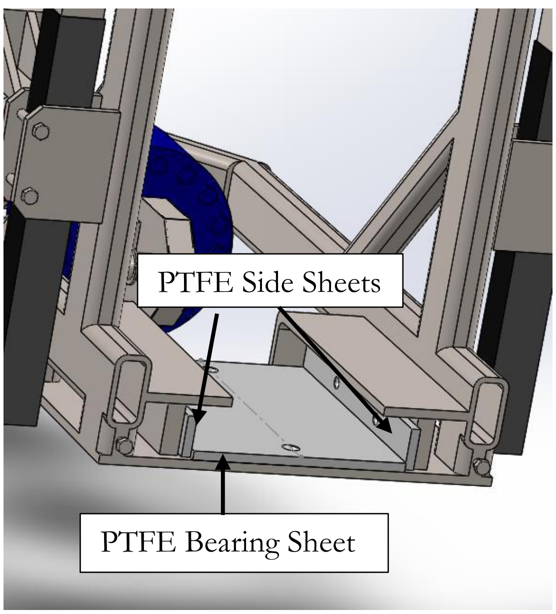


Figure 41. PTFE sheets directly on the main frame.

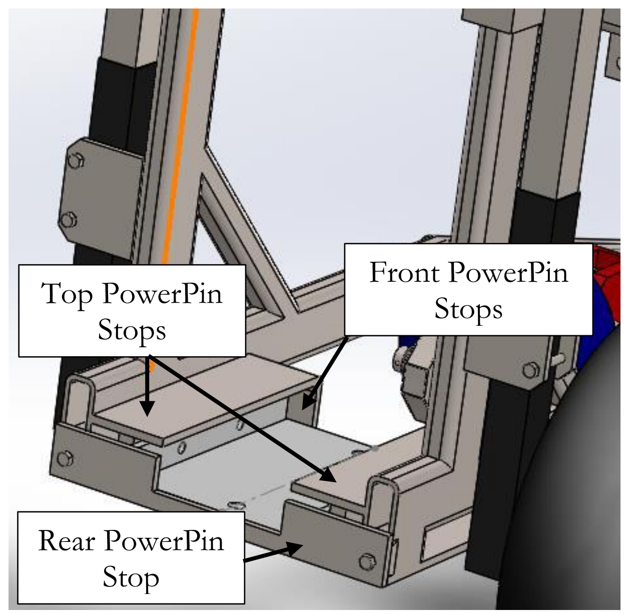


Figure 42. Stops for PowerPin rear hitch.

The two PTFE sheets that protect against sideways motion are bolted directly onto the frame, while the main PTFE bearing sheet will be bolted to the frame at the front and rear. The PowerPin will fit in the channel and will be restricted to a short range of motion by the front and rear PowerPin stops to allow for the load cell install. Vertical motion is constrained by the PTFE bearing sheet and top PowerPin stops, while transverse motion is prevented by the PTFE side sheets. The rear PowerPin stop is bolted on to the main frame to allow the PowerPin or PTFE to be removed or replaced if necessary.

5.6.3 Design Assessment

The new rear hitch support design addresses three of the client needs:

- **Need 8:** All load bearing aspects of the rear hitch support satisfy a minimum safety factor of 2 in FEA simulations and is outlined in Section 8.
- **Need 9:** The PTFE sheets are linear bearings that allow the PowerPin to have a small range of motion. This motion allows adjustment to assist in the load cell installation process.
- **Need 10:** The rear hitch support keeps the PowerPin aligned with the drawbar leading to accurate measurements.

5.7 Load Cell Integration

PAMI has pre-purchased three load cells with load capacities of 20,000 lbs, 50,000 lbs, and 100,000 lbs. This section will describe the corresponding background information, client needs, and design constraints to accommodate all three load cells into the load cart. Additionally, the vertical installation and screw height adjustment features will be explained.

5.7.1 Background and Constraints

The three load cells that will be used for the redesigned load cart can be observed in Figure 43.

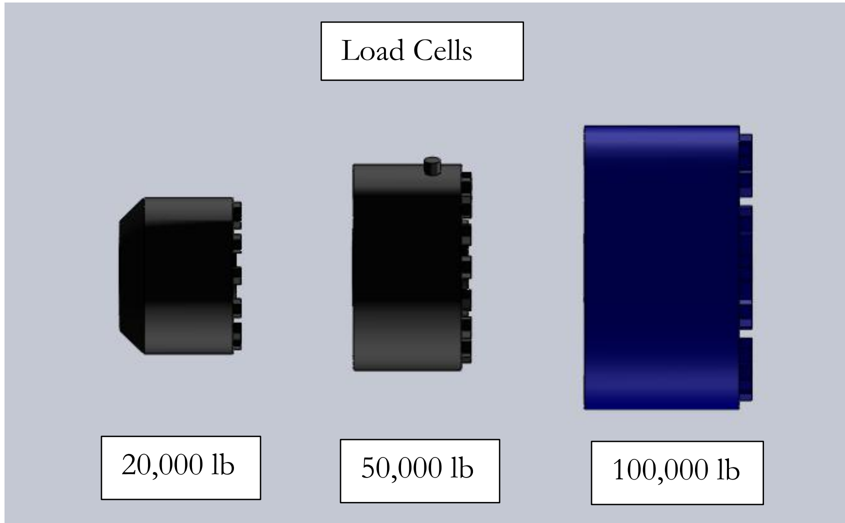


Figure 43. 20,000lb, 50,000lb and 100,000lb load cells.

The design of the new load cell integration is limited by the following constraints:

- **Constraint #5:** Design has to incorporate three different sized load cells
- **Constraint #9:** Minimum factor of safety of 2 for the design

5.7.2 Design Features

The final load cell integration design consists of two brackets that are fastened to the load cell connectors with two separate pins. The load cell integration is shown in Figure 44. More detail will be provided for the two-major load cell integration features: vertical installation and screw height adjustment.

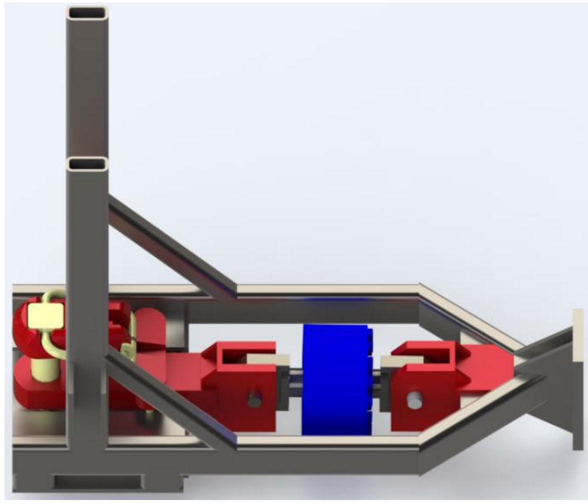


Figure 44. Vertical load cell integration.

Vertical Installation

The new load cell installation utilizes a vertical load cell implementation. The vertical load cell installation is illustrated in Figure 45.

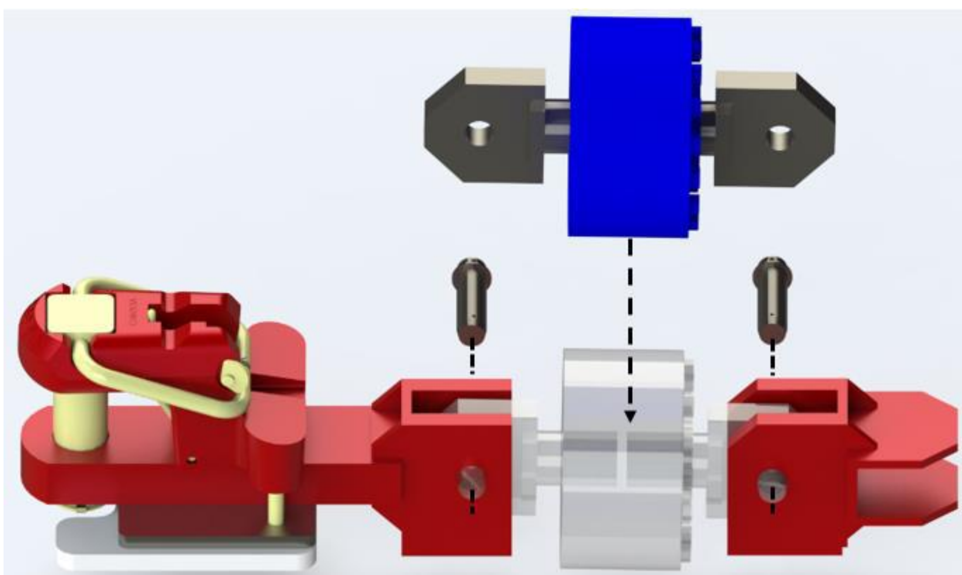


Figure 45. Load cell installation process.

The vertical load cell installation uses gravity to allow for the load cell connector to slide into the bracket easily. The team set a clearance of +/- 0.1875 inches between the load cell connectors and the bracket for adjustability when lining up the holes to insert the pin.

Load Cell Connection Screw Height Adjuster

Screws were implemented in the bottom of both new brackets to adjust the height of the load cell connector. The load cell connector is dropped into the bottom of the bracket and the screw adjuster is used to align the hole in the connector with the holes in the bracket. The adjuster is a screw that is threaded into a tapped hole in the bottom of the support bracket. The screws are ½" x 1 3/4" long. The adjuster screws can be observed in Figure 46.

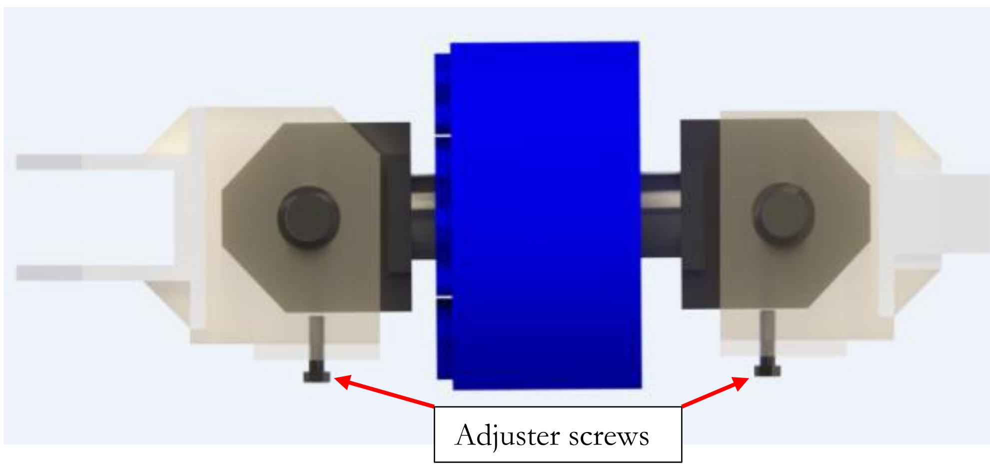


Figure 46. Bottom view of load cell installation.

In the current load cell installation process lining up the holes to pin the assembly together is difficult. The redesign simplifies the installation process using a vertical installation technique. The lack of space between the frame and the brackets is no longer an issue. The detailed dimensions for the components of the load cell integration are provided in the CAD models that will be submitted to the client at the end of this course.

5.7.3 Design Assessment

The design of the new load cell integration addresses the following needs:

- **Need #3:** The load cell brackets are designed to fit three different sizes of load cells.
- **Need #8:** All load bearing aspects of the load cell integration satisfy a minimum safety factor of 2 in FEA simulations and are outlined in Section 8.

- **Need #9:** The process of vertical load cell installation allows the load cell to be placed into the brackets with ease. Proper clearance between the load cell and the brackets gives the user an ergonomic installation.
- **Need #10:** The load cell mounting brackets allow the load cell to maintain alignment with the drawbar leading to accurate measurements.

5.8 Front Hitch Connection

PAMI has purchased two agriculture hitches that must connect to the front of the load cart. This section will describe the corresponding background information, client needs, and the design constraints to allow for the accommodation of category 3-5 power pin hitches. Additionally, the process of converting between different hitch categories will be explained.

5.8.1 Background/Constraints

The three pre-purchased power pin hitches connect to category 3, 4, and 5 drawbars. The hitches are attached through two and three bolt connections as observed in Figure 47 and Figure 48.

Figure 47. Category 4 and 5 hitches. [6]

Figure 48. Category 3 hitch. [6]

The design of the new front hitch connection is limited by the following constraints:

- **Constraint #8:** Source materials in standard sizes
- **Constraint #9:** Minimum Factor of Safety of 2 for the design

The front hitch connection design considers all relevant needs and constraints to achieve a final design that satisfies the client. The front hitch connection's design features are outlined in the next section.

5.8.2 Design Features

The final front hitch connection design consists of a tapered plate with three 1" vertical holes, that are welded to the outside of the extension beam. The final front hitch connection is displayed in Figure 49.

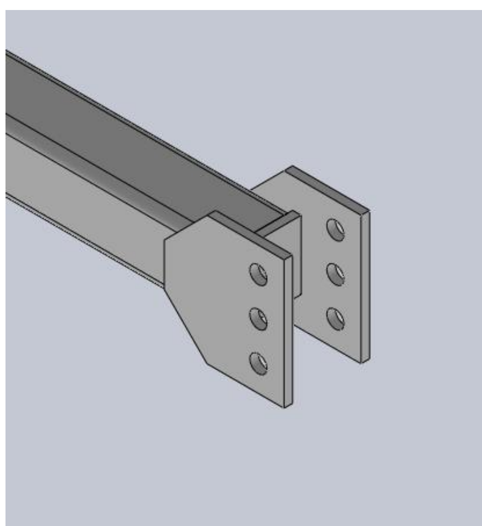


Figure 49. Final front hitch connection.

The hitch connection allows for comfortable conversion between category 3-5 hitches.

Category Conversion

Converting between category 3, 4 and 5 hitches is straightforward with this design allowing for a high level of versatility. The first step to convert between categories is to undo the bolted connections and remove the hitch. Second, the desired category is bolted on in the correct position. Category 3-5 hitches in their installed positions are shown in Figure 50 and Figure 51.

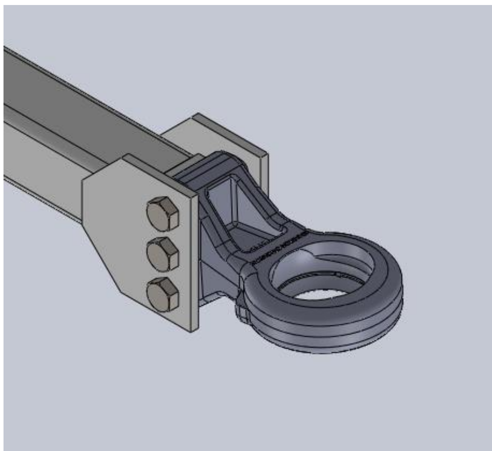


Figure 50. Category five hitch installed on bracket.

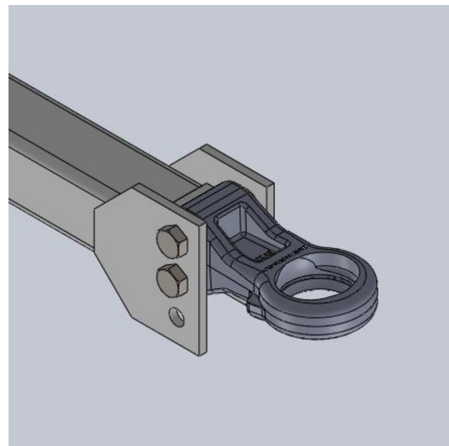


Figure 51. Category three hitch installed on bracket.

A render of the final design can be observed in Figure 52 with the Category 5 hitch attached.

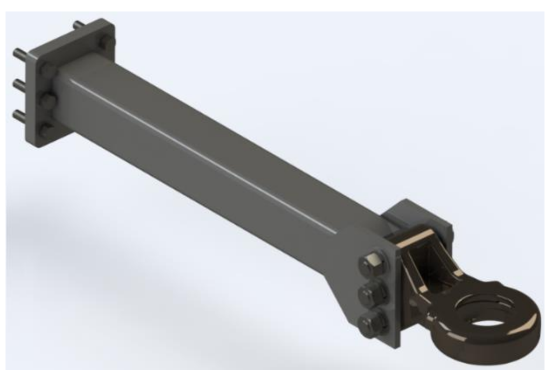


Figure 52. Front hitch attachment with category five hitch.

Details of the front hitch are provided in the CAD models that will be submitted to the client at the end of this course.

5.8.3 Design Assessment

The new front hitch connection design addresses the following needs:

- **Need#2:** The front hitch bracket is compatible with standardized category 3 to 5 agriculture hitches.
- **Need #8:** All load bearing aspects of the front hitch connection satisfy a minimum safety factor of 2 in FEA simulations and is outlined in Section 8.
- **Need #11:** The front hitch connection is compatible with drawbar categories of 3-5 as per ASABE standards.

5.9 Additional Design Features

There are several additional design features that are included in the final design to meet the client needs. These features include a safety chain, a trailer jack with wheel, hydraulic quick couplers, and a bullseye level.

5.9.1 Safety Chain

A requirement per ANSI/ASAE S338.5 is to include a safety chain between a tow vehicle and the towed agriculture field equipment. The method of attaching the safety chain is the same as the method used in the old load cart design. This is shown in Figure 53.

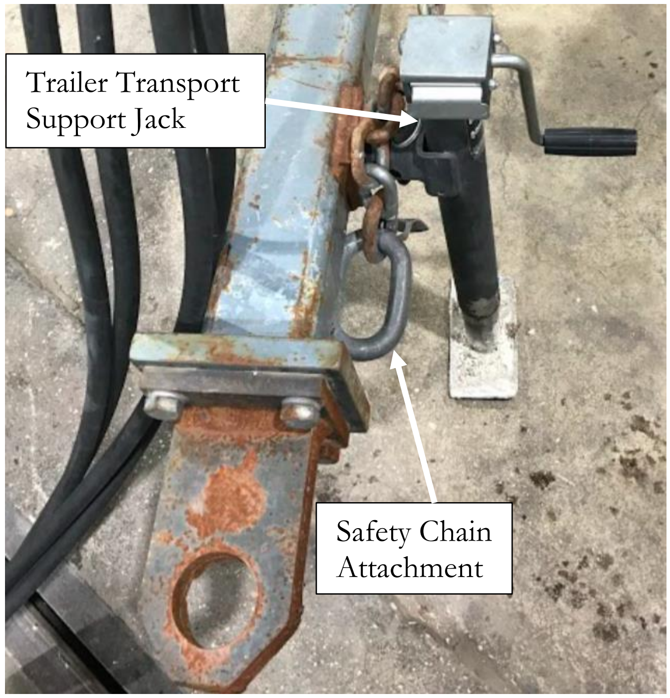


Figure 53. Safety chain attachment current load cart

The re-design incorporates a safety chain and attachment at the front of the extension beam.

This feature addresses the following client need:

- **Need #2**: The inclusion of a safety chain complies with ASABE requirements

5.9.2 Front Hitch Height Adjustment Jack

The new front hitch height adjustment jack will attach at the front of the extension beam. The jack is used to adjust the height of the front hitch when hooking up to the tow machine drawbar and support the load cart when in storage. The new trailer transport support jack is a 1,200lb bolt-on swivel trailer jack. The jack is displayed in Figure 54. The wheels on the jack permit the load cart to be unloaded off of a trailer while being fully self-supporting.

[redacted]

Figure 54. Bolt-on swivel trailer jack. [7]

The specifications of the jack are displayed in TABLE VI.

TABLE VI:

BOLT-ON SWIVEL TRAILER JACK SPECIFICATIONS

Minimum Height	26 inches
Maximum Height	39 inches
Maximum Load Capacity	1,200 lbs
Handle	Side wind

The new trailer transport jack design addresses the following client need:

- **Need #1**: The load cart can easily be loaded and unloaded on a trailer with the installation of the new bolt-on swivel trailer jack.

5.9.3 Hydraulic Quick Couplers

The new load cart design implements 4 sets of lever coupler break-away kits. A hydraulic coupler kit is shown in Figure 55.

Figure 55. Lever coupler break-away kit.

These hydraulic couplers can be implemented at the rear of the load cart with a bracket so that the two sets of hydraulic couplers can be stacked on top of each other. The four sets of hydraulic couplers can be mounted on the horizontal support as shown in Figure 56.

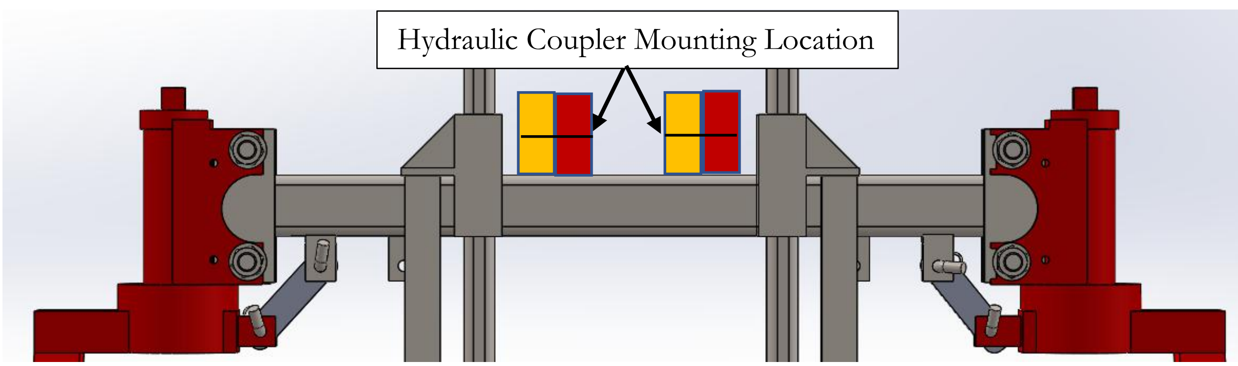


Figure 56. Hydraulic coupler mounting location on horizontal support beam.

The installation of the hydraulic quick couplers on the new load cart addresses the following need:

- **Need #11**: The load cart has four sets of standard pioneer quick couplers for hydraulics.

5.9.4 Bullseye Level

The new load cart design implements a bullseye level at the rear of the load cart on the horizontal support beam. This method of implementing the bullseye level is the same as the old load cart design. The bullseye level is displayed in Figure 57.

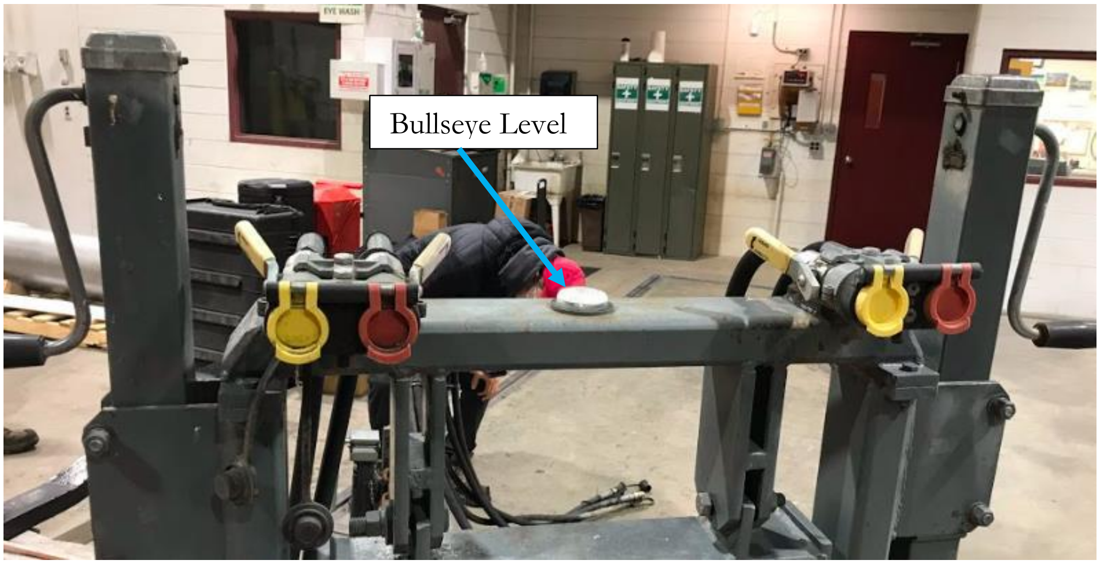


Figure 57. Bullseye level location.

6 Material and Cost Breakdown

This section outlines the specific materials used in the construction of the load cart along with their corresponding suppliers. A detailed estimate of the cost is also be provided.

6.1 Material and Suppliers

The materials used in the design were limited by the clients need for locally sourced materials and standard sized materials. PAMI already has working relationships with several companies in Manitoba, and several more were researched in order to source all materials locally.

The materials used in the redesign of the load cart can be broken up into the following five categories:

1. Frame
2. Fasteners & Pins
3. Hydraulic System
4. PTFE Bearings
5. Jacks and Level

6.1.1 Frame

The frame of the load cart was designed using standard steel tubing sizes and steel plate thicknesses. Brunswick Steel is one supplier that PAMI already has a relationship with for sourcing the steel for the load cart. Five different steel components were sourced from Brunswick Steel, summarized in TABLE VII.

TABLE VII:
FRAME MATERIAL SPECIFICATIONS

Component	Material	Yield Strength [psi]	Tensile Strength [psi]
1” Plate	ASTM – A36 Steel	36,000	58,000
½” Plate	ASTM – A36 Steel	36,000	58,000
2”x2”, ¼” Wall Thickness Tubing	HSS Steel Tubing ASTM A-500 Grade C	46,000	62,000
2”x4”, 3/8” Wall Thickness Tubing	HSS Steel Tubing ASTM A-500 Grade C	46,000	62,000
4”x4”, 5/16” Wall Thickness Tubing	HSS Steel Tubing ASTM A-500 Grade C	46,000	62,000

The two sizes of steel plates are both made up of ASTM A36 Steel which has a yield strength of 36,000 psi, while the steel tubing sections were made up of ASTM A-500 Grade C steel with a yield strength of 46,000 psi from the Brunswick Steel reference catalogue [8].

6.1.2 Fasteners and Pins

The fasteners and pins used for the different features of the load cart were sourced from companies that have distribution centers in Winnipeg. As the load cart must withstand 60,000 lbs loads, high strength bolts were sourced for all load bearing connections. Additionally, high quality pins were sourced for the load cell connections. Most of the fasteners and pins were able to be sourced from Fastenal, a company which PAMI already has an account with [9]. The pins used in the caster locking system were sourced from Acklands Grainger as they had a lower cost [10]. A breakdown of the fastener and pin specifications is provided in TABLE VIII.

TABLE VIII:

FASTENER AND PIN SPECIFICATIONS

Component	Material	Supplier	Quantity	Yield Strength [psi]
¾" Bent Hitch Pin for Caster Locking	AISI 1038 Steel	Acklands Grainger	4	70,000
1 ½" x 8" Hitch Pin for Load Cell Mounting	AISI 1045 Steel	Fastenal	2	45,000
1 ¼" x 6.5" Bolts	Grade 8 Steel	Fastenal	4	150,000
1 ¼" Nylon Insert Lock Nuts	Grade 8 Steel	Fastenal	4	150,000
1 ¼" Flat Washer	Grade 8 Steel	Fastenal	8	N/A
1" x 6 ½" Bolts	Grade 8 Steel	Fastenal	3	150,000
1" Hex Nuts	Grade 8 Steel	Fastenal	3	150,000
1" Flat Washer	Grade 8 Steel	Fastenal	6	150,000
1" Lock Washer	Grade 8 Steel	Fastenal	3	150,000
5/8" x 3" Bolts	Grade 8 Steel	Fastenal	6	150,000
5/8" Hex Nuts	Grade 8 Steel	Fastenal	6	150,000
5/8" Flat Washer	Grade 8 Steel	Fastenal	6	150,000
5/8" Lock Washer	Grade 8 Steel	Fastenal	3	150,000
½" x 4 ¼" Bolts	Grade 5 Steel	Fastenal	4	120,000
½" Hex Nut	Grade 5 Steel	Fastenal	4	120,000
½" Lock Washer	Grade 5 Steel	Fastenal	4	120,000
½" x 1" Flat Socket Cap Screw	Grade 316 Steel	Fastenal	4	55,000
½" x 1 ¾" Bolt	Grade 5	Fastenal	2	120,000
½" x 1" Bolt	Grade 5	Fastenal	2	120,000

Grade 8 bolts were sourced for all load bearing connections due to their high strength in tensile applications. For the steel plate to PTFE sheet connections, lower grade bolts are used.

6.1.3 Hydraulic System

The hydraulic system on the load cart is strictly for extending the hydraulics from the towing machine past the load cart to the implement. This allows the implement to utilize the hydraulic pump from the tractor to properly perform its application. The hydraulic system breakdown is provided in TABLE IX.

TABLE IX:
HYDRAULIC SYSTEM SPECIFICATIONS

Component	Supplier	Quantity	Maximum Pressure [psi]
Lever Coupler Break-Away Kit **[11]**	Napa	4	3600
Male Crimp Fitting **[12]**	Green Line Hose	16	3600
5/8" x 14' Hydraulic Hose **[13]**	Green Line Hose	8	3600
½" NPT Hydraulic Quick Coupler Tip **[14]**	Princess Auto	8	3625
Aluminum Line Clamps **[15]**	Amazon	20	N/A
½" x 36" Plain Steel Round Rod **[16]**	Home Depot	1	N/A

All components in the hydraulic system were sourced for high pressure applications, due to the large hydraulic pumps located on towing machines and large hydraulic cylinders located on the implements. The maximum pressure of all components with fluid flowing through them can withstand is 3600 psi.

6.1.4 PTFE Bearings

The PTFE bearings allow the PowerPin system to effectively transfer the draft load to the load cell, making them one of the most crucial components on the load cart. PTFE was able to be locally sourced from Johnston Industrial Plastics. The PTFE Bearing system is provided in TABLE X.

TABLE X:

PTFE BEARING SPECIFICATIONS

Component	Supplier	Quantity [in2]	Maximum Compressive Strength [psi]
½" PTFE Sheet Virgin	Johnston Plastics	264	3700
1" PTFE Sheet Virgin	Johnston Plastics	150	3700

The PTFE from Johnston Plastics has an operating temperature range of -240°C to 260°C and is not affected by outdoor weathering making it ideal for the load cart application [17].

6.1.5 Jacks and Level

The main component for the height adjustability of the load cart are the two manual jacks incorporated on the vertical supports of the main frame. Once the load cart is adjusted to the right height it also needs to be level to ensure that the load cell is in the correct plane. The hardware for the jacks and levelling system were again sourced locally, except for the level. A level on Amazon provided a low-cost alternative with optional 2-day shipping. The specifications for the jacks and level are provided in TABLE XI.

TABLE XI:

HEIGHT ADJUSTMENT SYSTEM SPECIFICATIONS

Component	Supplier	Quantity	Maximum Rated Load [lbs]
Weld-On Drop Leg Trailer Jack **[5]**	Princess Auto	2	5000
Bolt-On Swivel Trailer Jack **[7]**	Princess Auto	1	1200
Bullseye Spirit Level **[18]**	Amazon	1	N/A

The maximum rated load for adjusting the height of the jacks is 5000 lbs, allowing the two of them to easily lift the frame. The bolt-on swivel jack is incorporated onto the extension beam of the load cart to allow for loading and unloading the load cart from the trailer.

6.2 Total Cost

The total cost of the load cart is the cost of the materials outlined, along with the associated manufacturing to build the cart. The cost of the pre-purchased materials, the front hitches, swivel casters, PowerPin, and load cells will not be included in this estimate at the request of PAMI.

6.2.1 Material Cost

The total cost of the system was able to be calculated from quotes provided by Brunswick Steel, and Johnston Industrial Plastics, in addition to online pricing provided by all other suppliers. A summary of the total cost of the load cart materials is provided in TABLE XII.

TABLE XII:
SUMMARY OF MATERIAL COST

Category	**Cost**
Frame	$1,146.00
Fasteners & Pins	$748.09
Hydraulic System	$2,056.39
PTFE Bearings	$456.84
Jacks and Level	$219.19
Total with tax	$5,227.96

A full summary of the materials, along with part numbers, quantities, and individual material cost is in Appendix B.

6.2.2 Manufacturing Cost Estimate

PAMI does most of its own manufacturing in house and contracts out more complicated designs to third parties. In general, PAMI pays a shop rate of $110/hr for contracted manufacturing and this rate is used to provide the number of manufacturing hours that will keep the project within the budget.

Based on the total budget of $10,000 and taking into account the material cost of approximately $5,228. This leaves $4,772 for what amounts to approximately 42 hours to manufacture the load cart redesign. The majority of the manufacturing hours will be spent machining parts and welding the assemblies. Any laser cut parts will be outsourced, which would also contribute to the total manufacturing cost. The team is confident that manufacturing costs for the load cart redesign will be below budget.

7 FMEA

The team implemented a standard FMEA template for use in this report. The ratings for severity, frequency and likelihood to detect the failure were taken from the MECH 4860 FMEA lecture and are displayed in Appendix C. Based on the severity, frequency and likelihood a risk priority number was calculated to determine how critical each failure mode would be. The FMEA conducted for the project is displayed in TABLE XIII

TABLE XIII:

FAILURE MODE EFFECTS ANALYSIS FINAL DESIGN

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
Item	In what ways can the Process Step or Input fail?	What is the impact on the Key Output Variables once it fails (customer or internal requirements)?	How **Severe** is the effect to the customer?	What causes the Key Input to go wrong?	How **often** does cause or FM **occur?**	How well can you **detect** the Cause or the Failure Mode?	Risk Priority Number	What are the actions for reducing the occurrence of the cause, or improving detection?
Front hitch bracket	Bolt failure at the hitch and bracket connection.	• Failure of other bolt connections at front hitch bracket • Front hitch detachment from load cart • Load cart detachment from tow vehicle.	10	Loading exceeds the yield strength of the bolts.	1	10	100	No action recommended.
	Plate failure of the bracket	• Front hitch detachment from load cart. • Load cart detachment from tow vehicle.	10	Loading exceeds the yield strength of the plate.	1	10	100	No action recommended.
	Weld connection failure at front hitch assembly at extension beam	• Front hitch assembly detachment from load cart • Load cart detachment from tow vehicle.	10	Loading exceeds the yield strength of the weld.	1	10	100	Inspect weld areas to determine if signs of failure are present.

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
Extension Beam	Bolt failure at extension beam connection to load cart	• Extension beam detachment from load cart. • Load cart disconnected from tow vehicle.	10	Loading exceeds the yield strength of the bolts.	1	10	100	No action recommended.
	Bearing failure of load cart front mounting plate.	• Extension beam detachment from load cart. • Load cart disconnected from tow vehicle.	10	Loading exceeds the yield strength of the plate.	1	10	100	No action recommended.
Main Frame	Contact with tow vehicle	• Damage to the main frame or tow vehicle • Yielding of main frame or tow vehicle	10	Tow vehicle takes a tight turn in the field	2	5	100	Perform routine checks on the load cart to determine if contact with tow vehicle is occurring.
	Contact with implement	• Damage to the main frame or implement • Yielding of main frame or implement	10	Tow vehicle takes a tight turn in the field	2	5	100	Perform routine checks on the load cart to determine if contact with the implement is occurring.
	Contact with caster wheel	• Tire wear • Flat tire	5	Tow vehicle takes a tight turn in the field	4	1	20	Perform routine checks on caster wheels for signs of wear
Load Cell Connection	Weld failure at load cell connection front/rear	• Load cell damage	8	Loading exceeds the yield strength of the weld.	3	10	240	Inspect weld areas to determine if signs of failure are present.
	Pin failure at load cell connection front/rear	• Load cell damage	8	Loading exceeds the pin yield strength	1	10	80	No action recommended.

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
	Bearing failure of front and rear support brackets	• Load cell damage	8	Loading exceeds the yield strength of the brackets.	1	10	80	No action recommended.
	Cross thread adjuster bolt	• Deformed threads in adjuster bolt hole • Inoperable adjuster bolt	1	Adjuster bolt cross threaded when fitting load cell connection	1	2	2	Ensure operator checks that adjustment is possible.
	Load cell struck by foreign material in the field	• Load cell damage • Main frame damage	9	Foreign material of significant size strikes load cell or main frame.	3	10	270	Ensure test operator does not run over foreign objects while using load cart.
Caster Wheel Assembly	Tire contact with foreign material in field	• Tire wear • Flat tire	5	Abrasive materials being run over in the field	4	1	20	Perform routine checks on caster wheels for signs of wear
	Bolt failure at caster connection to load cart	• Caster assembly detachment from load cart. • Bending or yielding of caster attachment beam	10	Loading exceeds the yield strength of the bolts.	1	10	100	No action recommended.
	Bearing failure of plate connecting caster to beam.	• Caster assembly detachment from load cart.	10	Loading exceeds the yield strength of the plate.	1	10	100	No action recommended.
	Pin failure of caster locking mechanism	• Caster tire and frame contact • Tire wear • Flat Tire	5	Loading exceeds the yield strength of the pins.	2	10	100	No action recommended.

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
Caster Wheel Assembly	Bearing failure of tabs holding caster locking pins in place.	• Caster tire and frame contact • Tire wear • Flat Tire	5	Loading exceeds the yield strength of the caster locking tabs.	2	10	100	No action recommended.
	Failure due to insufficient lubrication.	• Restricted rotational motion	1	Operator does not lubricate the caster assembly at proper maintenance intervals.	4	2	8	Perform proper lubrication maintenance on the caster.
Vertical Supports	Bending of vertical support beams.	• Caster assembly misalignment • Bending/yielding of jack	10		1	10	100	No action recommended.
	Yielding of vertical supports.	• Horizontal caster attachment beam and caster assembly detachment from load cart	10	Loading exceeds the yield strength of the vertical supports.	1	10	100	No action recommended.
Jack Supports	Yielding of jack	• Load cart reverts to maximum height resting against the angled supports • Height adjustment not operational	10	Loading exceeds the yield strength of the jack.	1	10	100	No action recommended.

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
Jack Supports	Weld connection of jack to vertical beam fails	• Load cart reverts to maximum height resting against the angled supports • Height adjustment not operational • Jack contact with ground	9	Loading exceeds the yield strength of the weld.	1	1	9	Inspect weld areas to determine if signs of failure are present.
	Weld connection to horizontal beam failure	• Load cart reverts to minimum height • Height adjustment not operational	2	Loading exceeds the yield strength of the weld.	1	2	4	Inspect weld areas to determine if signs of failure are present.
	Jack handle contacts wheel	• Removal of jack handle • Tire wear • Flat tire	5	Jack handle left in travel path of caster wheel after adjusting height of load cart.	4	1	20	Ensure that jack handle is put into upright position before operating load cart.
Power Pin Assembly	PTFE wears out	• PowerPin adjustability no longer operable	2	Adjusting the PowerPin to install load cell.	10	2	40	Inspect the PTFE surfaces to determine if the bearing surfaces are still useable.
	Downward forces on PowerPin cause vertical motion	• Yielding of horizontal PowerPin supports	8	Vertical loading on the PowerPin connection to the implement.	3	3	72	Inspect the PowerPin support plates for signs of contact with PowerPin
	PowerPin pin failure	• Rear hitch detachment from load cart • Load cart detachment from implement	10	Loading exceeds the pin yield strength	1	10	100	No action recommended.

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	DET	RPN	Actions Recommended
Power Pin Assembly	Hairpin failure on bottom of pin.	• PowerPin pin reverts to upright position	10	Loading exceeds the hairpin yield strength	1	1	10	Ensure operator places the hairpin on the bottom of the PowerPin before operating load cart.
Hydraulics	Failed seal on the quick coupler	• Hydraulic oil leak • Pressure loss in hydraulic lines	7	High pressure in hydraulic lines	4	2	56	Monitor pressure in hydraulic lines to determine if pressure drop occurs indicating a leak.
	Worn out hydraulic lines	• Hydraulic oil leak • Pressure loss in hydraulic lines	7	Contact between hydraulic lines and components of load cart or tow machine.	4	2	56	Monitor pressure in hydraulic lines to determine if pressure drop occurs indicating a leak.

8 Stress Analysis

Finite Element Analysis (FEA) has been performed on the major load bearing components of the load cart. Hand calculations are also performed on simplified models to compare with FEA results. The applied loads, fixtures, and model details for each analysis are outlined in this section. The results are then analyzed with a focus on high deformation and high stress areas.

The major loads experienced by the load cart during operation are draft loads, tongue weight, sideways loads on the wheels, and drag on the wheels. The maximum allowable loads for each case are shown in TABLE XIV. Applied loads for the analysis will be multiplied by the factor of safety for the respective simulation (equal to 2 or 3) to show the stress distributions with the factor of safety.

TABLE XIV:

FEA ANALYSIS APPLIED LOAD TYPES AND FORCES

	Draft Load (Category 5) [19]	Draft Load (Category 3) [20]	Tongue Weight [21]	Sideways Caster Load	Wheel Drag
Load [lbf]	60,000	40,000	6,700	200	333

Two types of standard steel are used in the design. These include one grade of steel for the plates, and another grade for the tubing. The properties of both steels are listed in TABLE XV.

TABLE XV:

DETAILED MATERIAL SPECIFICATIONS FOR FEA ANALYSIS

	ASTM Steel Standard	Ultimate Tensile Strength [psi]	Yield Strength [psi]	Shear Modulus [ksi]	Poisson's Ratio
Plating [22]	A36	58,000	36,000	11,500	0.28
Tubing [23]	A-500 Grade C	62,000	46,000	11,600	0.23

All simulations were completed using a solid tetrahedral mesh. The solid elements are more suitable for bulkier objects that are common throughout the load cart design. The tetrahedral mesh is very common and works well with different geometries, but the inherent inaccuracies in the mesh type must be considered in the analyses conducted. Shear locking in cases with only one or two elements across the thickness of a section of the part was considered acceptable. Also, the percentage of distorted elements with given aspect ratios are displayed and kept to a minimum for each analysis.

8.1 Frame

The frame is analyzed for stresses and deflections. The analysis includes hand calculations to compare to the results of the FEA analysis.

8.1.1 Hand Calculations

Hand calculations were performed for the frame. The loading scenario for the frame is complex but was simplified for hand calculation purposes. The maximum stress calculated at the vertical support was 2,450 psi. This stress was compared to the maximum stress found after performing FEA on the part. The detailed hand calculations can be observed in Appendix D.

8.1.2 FEA Model Simplification

The focus of the frame analysis is on the section of the load cart shown in Figure 58. The model is modified for FEA by removing the non-load bearing parts at the base, adding a beam connection between the vertical supports to simulate the horizontal support connection, and approximating the jack connection as solid blocks. The FEA model with the changes is shown in Figure 59.

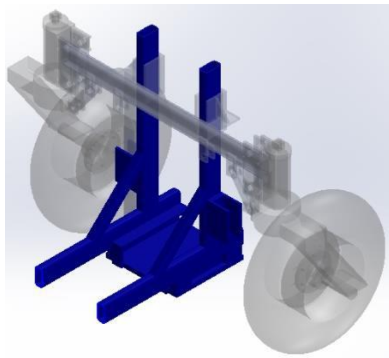


Figure 58. Vertical support FEA analysis section.

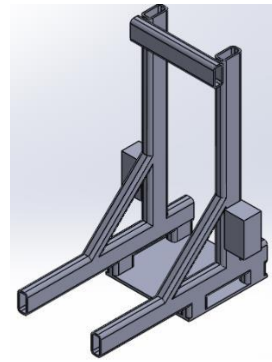


Figure 59. Actual frame FEA model.

8.1.3 Loads and Fixtures

The forces applied to the caster wheels are transferred through the frame to the power pin and the front hitch connection during operation. The PowerPin is designed on a linear bearing that only allows the hitch to slide either forwards or backwards. Due to the linear bearing, any load along the direction of the frame must transfer through the front hitch connection with all other loads being transferred through the PowerPin. To simulate this load transfer, roller bearing fixtures are placed on the front of the base frame, the inside of the PowerPin support walls, and the top of the PowerPin support plate. The bearing fixtures are shown in Figure 60 as green arrows which represent the axes in which loads can be transferred through a fixture.

The loads applied to the frame simulate the wheel drag, sideways wheel forces, and maximum tongue weight loads listed in TABLE XIV. The applied loads are shown in Figure 60 in the form of purple arrows. The tongue weight is simulated at the mounting point of the jacks while the caster forces are applied at the very top of the frame to simulate a worst-case scenario.

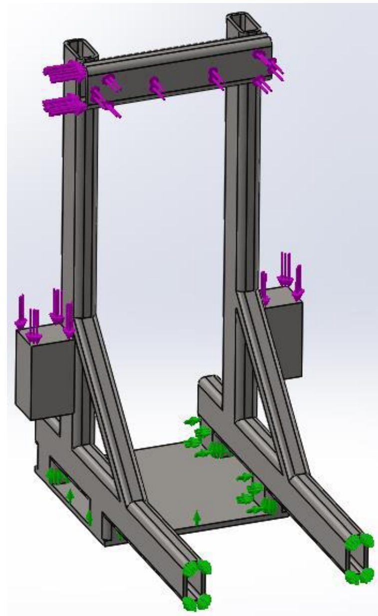


Figure 60. Frame FEA model fixtures (green) and applied loads (purple).

8.1.4 Model Details

The maximum Von Mises stress in the frame converges around 52,000 nodes at a maximum stress of 15,140 psi as shown in Figure 61.

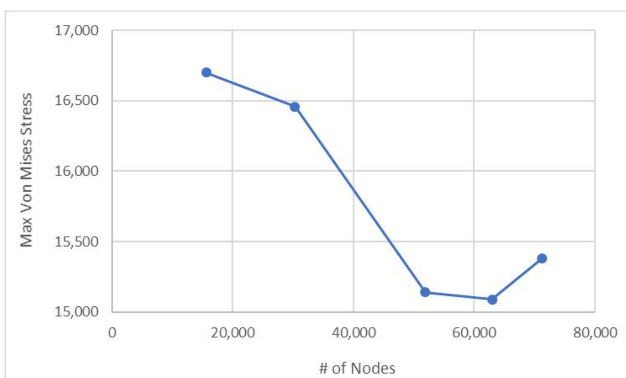


Figure 61. Frame FEA convergence plot.

The details of the final mesh used in this analysis are shown in TABLE XVI.

TABLE XVI:

VERTICAL SUPPORT FEA FINAL MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	71,183
Total Elements	38,139
Element Aspect Ratios < 3	74.3%
Element Aspect Ratios > 10	0.0367%
% Distorted Elements	0.00%

8.1.5 Results

The final Von Mises stress distribution in the frame is shown in Figure 62. The largest stress results in an equivalent factor of safety for the frame equal to 6.2. The largest stress is seen at the connection between the angled supports and the vertical supports. Another high stress area is also shown where the horizontal beam connects to the vertical supports due to the loads from the casters. Lastly, the base of the frame connecting to the PowerPin support plate shows some large stresses.

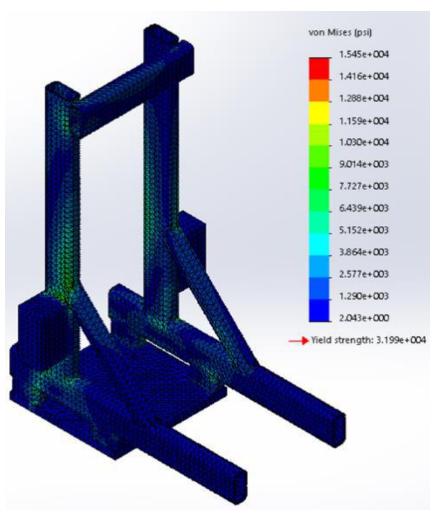


Figure 62. Frame FEA Von Mises stress distribution results.

The maximum deflection at the top of the frame is 0.036 inches as seen in Figure 63. This is a very minor deflection and is considered negligible for the analysis and any possibly interference with other parts.

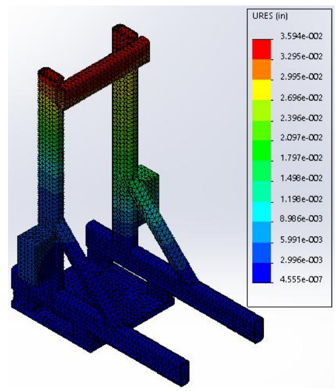


Figure 63. Frame FEA deflection results.

8.2 Front Hitch Extension Connection

The front hitch extension connection includes a plate for the extension to bolt on to and experiences the tensile loads from the implement. The model consists of all parts between the angled frame tubing and the plate that bolts the extension to the frame. As shown in Figure 65, the bolted-on plate is actually merged with front plate. The half inch plate from the welded-on load cell H-bracket is also approximated by being merged with the other bodies.

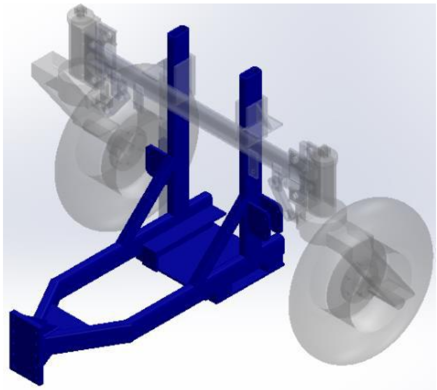


Figure 64. Front hitch extension connection section for FEA analysis.

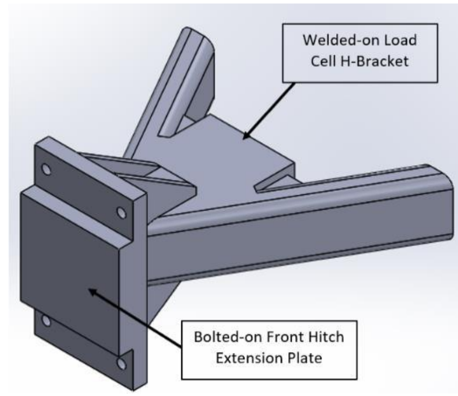


Figure 65. Actual Front hitch extension connection model for FEA.

Another model is used to simulate the height adjustable feature of the hitch extension connection. Figure 66 shows the model with the raised front hitch connection. Since the model is symmetrical from top to bottom, the results are equivalent to a lowered connection.

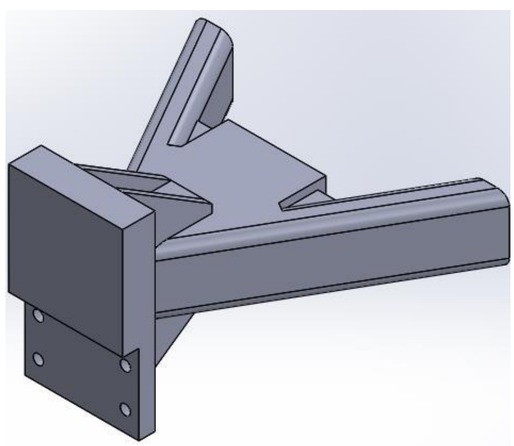


Figure 66. Raised Front hitch extension connection model for FEA.

8.2.1 Fixtures and Loads

The angled tubes from the frame as well as the load cell H-bracket are fixed for the simulation. The category 5 draft load with a safety factor of 3 is then applied to the front hitch extension plate for the centered hitch connection. For the raised hitch connection, the applied load is reduced to a

30,000 lbf draft load with a safety factor of 2. The applied forces and fixtures are shown in Figure 67 as purple and green arrows, respectively.

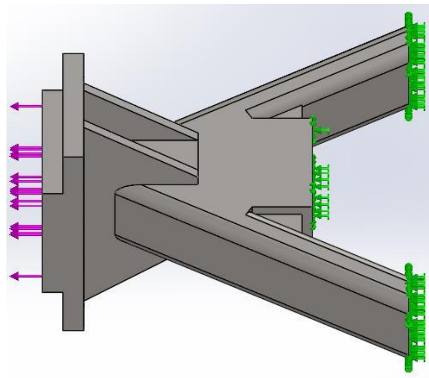


Figure 67. Front hitch extension connection FEA model fixtures (green) and applied loads (purple).

8.2.2 Model Details

Both models converge without any stress singularities. The centered front hitch connection converges with a factor of safety of 3 using the category 5 draft load. The load approaches 24,850 psi at 89,271 nodes. For the raised front hitch connection, the stress reaches approximately 35,000 psi at 54,258 nodes.

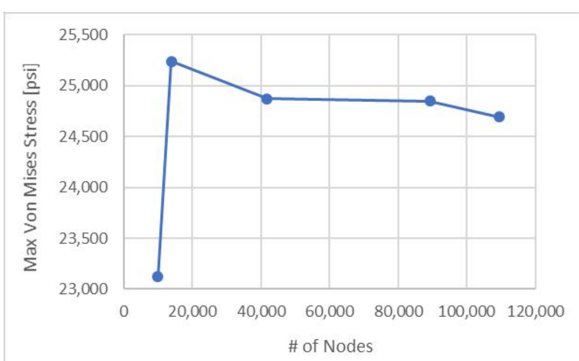


Figure 68. Stress convergence at the centered front hitch connection.

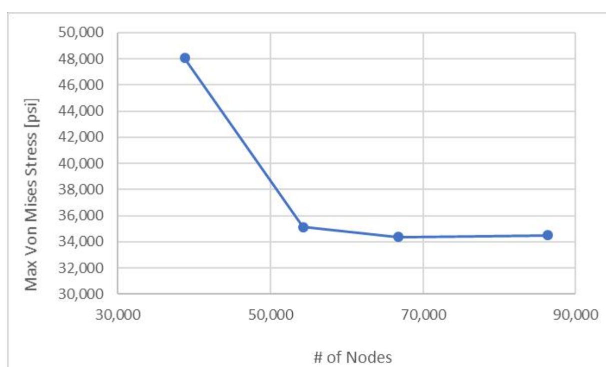


Figure 69. Stress convergence at the raised front hitch connection.

The final mesh for both analyses results in no distorted elements or elements with aspect ratios above 10. The mesh information for the final analyses on the frame front hitch extension connection is shown in TABLE XVII.

TABLE XVII:
FRONT HITCH EXTENSION CONNECTION FINAL MESH DETAILS

	Centered Connection	**Raised Connection**
Mesh Type	Tetrahedral Solid Mesh	Tetrahedral Solid Mesh
Total Nodes	86,367	87,423
Total Elements	55,273	56,132
Element Aspect Ratios < 3	99.4%	99.4%
Element Aspect Ratios > 10	0.0%	0.0%
% Distorted Elements	0.0%	0.0%

8.2.3 Results

The final Von Mises stress distribution in the centered front hitch extension connection is shown in Figure 70. Centered front hitch extension connection FEA stress distribution results.. The largest stress results in an equivalent factor of safety equal to 4.4. The largest stresses are seen at the fillet between the H-bracket and the front connection block, and the fillet between the front connection block and the front plate.

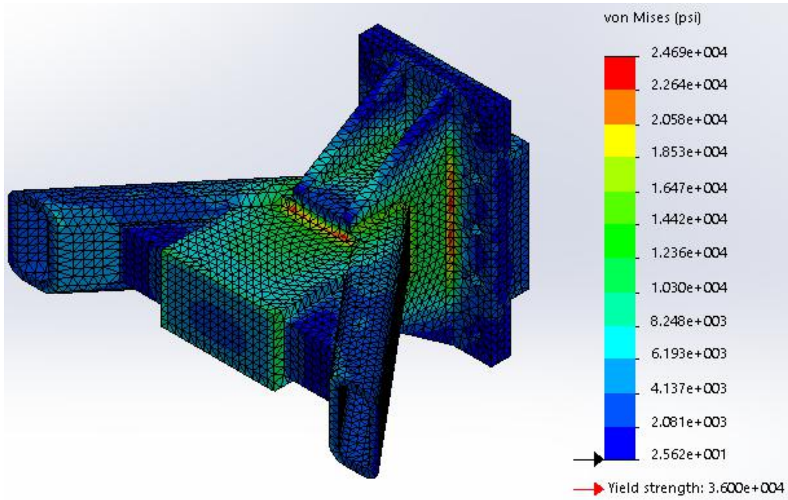


Figure 70. Centered front hitch extension connection FEA stress distribution results.

The final Von Mises stress distribution in the raised front hitch extension connection with an applied load of 30,000 lbf is shown in Figure 71. The largest stress results in an equivalent factor of safety equal to 2.1. Due to the offset applied force the largest stress is seen on the top at the fillet between the H-bracket and the front connection block.

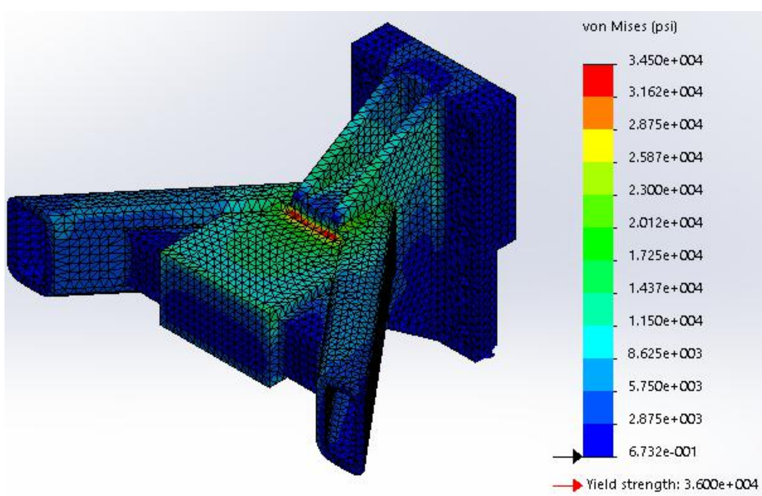


Figure 71. Raised front hitch extension connection FEA stress distribution results.

8.3 Horizontal Beam

The horizontal beam section being analyzed is shown in Figure 72 consisting of the rectangular beam and caster mounting plates. The actual model for analysis in Figure 73 is simplified to include one side of the beam with the sliders approximated as a solid block. The beam experiences all loads from the casters at the caster mounting plates.

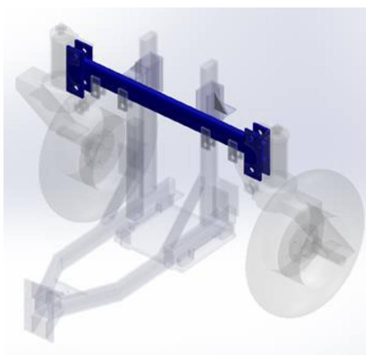


Figure 72. Horizontal beam FEA analysis section.

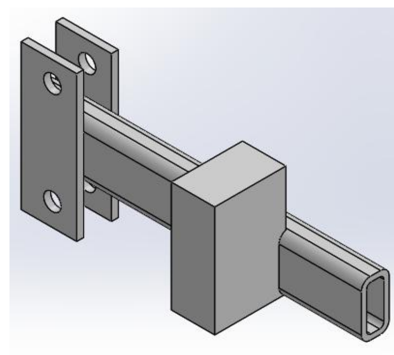


Figure 73. Actual half beam model for FEA.

8.3.1 Fixtures and Loads

A remote vertical load is applied to the caster mounting plates from the location of the center of the caster wheels in the horizontal plane. This simulates the implement tongue weight with a factor of safety of 3. Additionally, the wheel drag force is added to the caster mounting plates in the opposite direction of the load cart. The forces transfer through the beam to the vertical support slider that is represented as fixed object on the top and bottom. The center of the beam is also fixed. The location of the applied forces and fixtures are shown in Figure 74 as purple and green arrows, respectively.

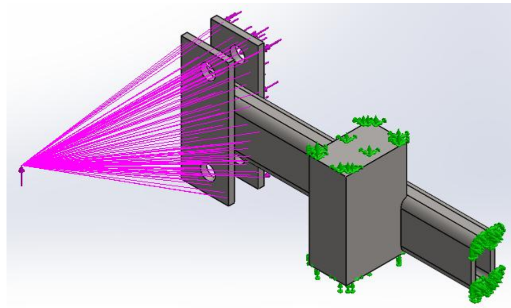


Figure 74. Horizontal beam FEA model fixtures (green) and applied loads (purple).

8.3.2 Model Details

The connection between the vertical support block and the beam experiences a stress singularity at the bottom corner. To analyze the convergence of the Von Mises stress, two of the highest stress points were selected on the model. The points being analyzed are shown in Figure 75, with point A inside the beam at the caster mounting point and point B at the base of the beam near the vertical supports.

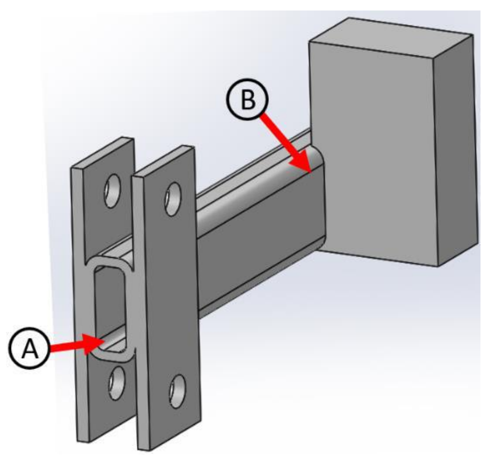


Figure 75. High stress points on the horizontal beam analyzed for convergence.

Point A converges at 65,700 nodes with a Von Mises stress of 32,200 psi as shown in Figure 76. Figure 77 shows point B converging around 62,000 nodes with a Von Mises stress of approximately 25,000 psi.

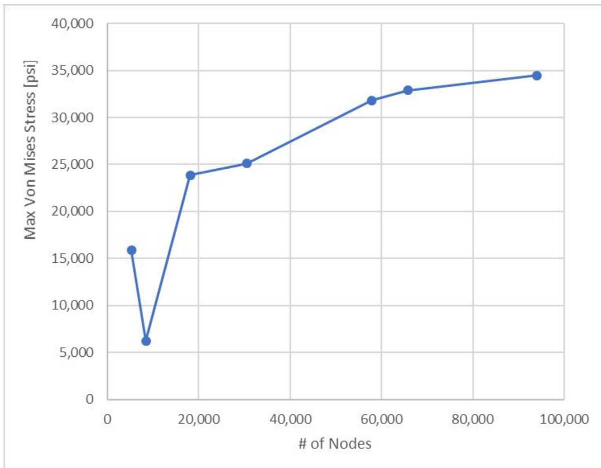


Figure 76. Stress convergence inside the horizontal beam at the caster mounts.

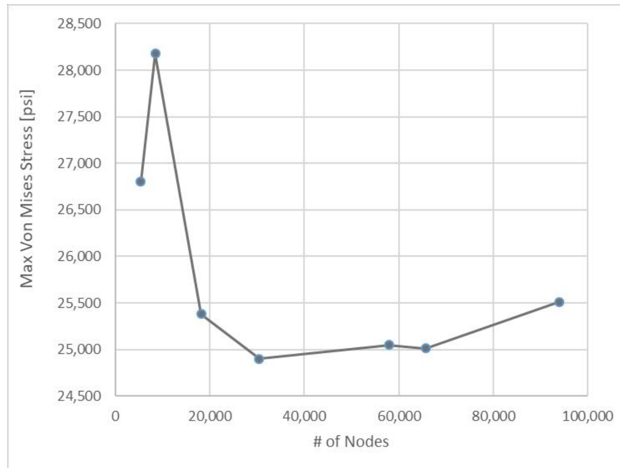


Figure 77. Stress convergence on the horizontal beam near the vertical support.

The details of the final mesh used in the most refined analysis are shown in TABLE XVIII.

TABLE XVIII:
HORIZONTAL BEAM FEA FINAL MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	94,059
Total Elements	58,641
Element Aspect Ratios < 3	99.9%
Element Aspect Ratios > 10	0.0%
% Distorted Elements	0.0%

8.3.3 Results

The final Von Mises stress distribution in the beam and caster mounts is shown in Figure 78 and Figure 79. The largest stress is at point A which results in an equivalent Factor of Safety of 3.35.

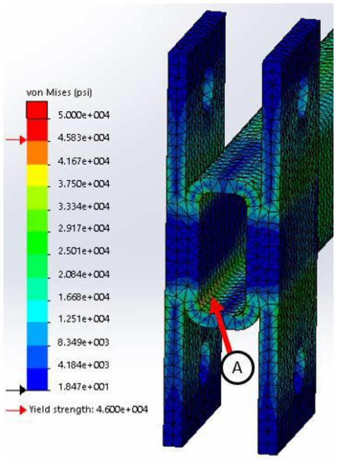


Figure 78. Stress convergence inside the horizontal beam at the caster mounts.

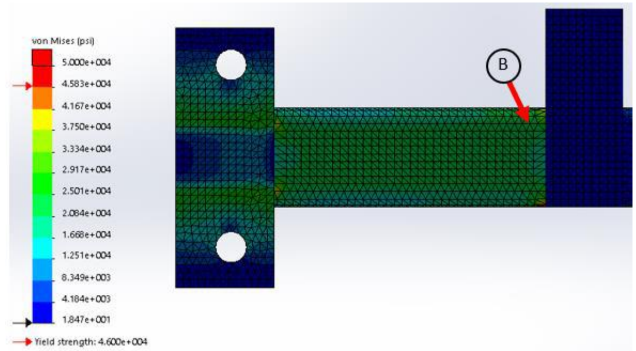


Figure 79. Stress convergence on the horizontal beam near the vertical support.

The deformation of the beam is the largest at the bottom of the caster mounts with a horizontal displacement of almost one-hundredth of an inch. The deformation distribution is shown in Figure 80 with a slight twist in the beam caused by the remote load due to the offset of the caster wheels.

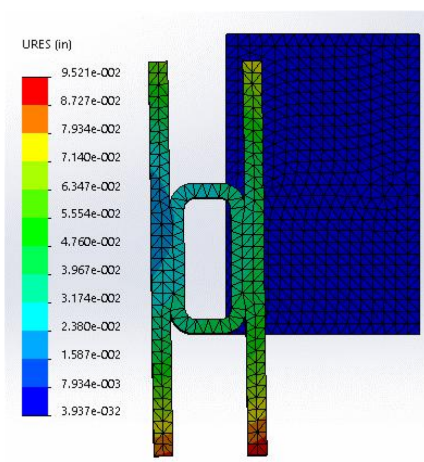


Figure 80. Caster mount deformation distribution.

8.4 Caster Locking Mechanism

The caster locking mechanism highlighted in Figure 81 restricts the rotational force of the caster wheel while the load cart is moving in reverse. The model in Figure 82 is used to simulate the rotating section of the caster and the caster locking mechanism. The locking bar is a separate body from the other two objects and is connected using the FEA pin feature for the simulation.

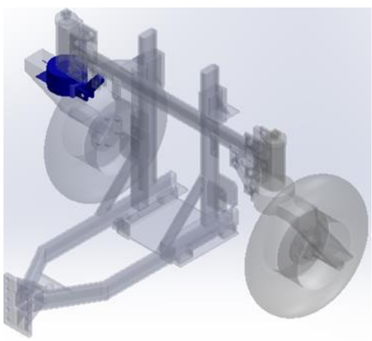


Figure 81. Caster locking FEA analysis section.

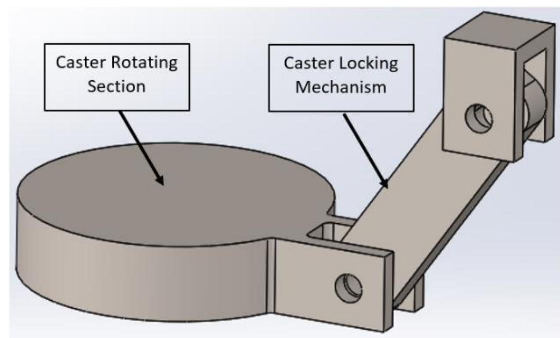


Figure 82. Actual caster locking model for FEA.

8.4.1 Loads and Fixtures

The top of the tabs that connect to the horizontal beam are fixed for the simulation. The two blue circles shown in Figure 83 are pins that connect the tabs to the locking bar which are separate bodies. The rotating section of the caster is a hinged fixture which allows one axis of rotation. The axis of rotation is in the same as the direction of the loads shown by the purple arrows in Figure 83. The torque applied to the caster section is 10,000 lbf*in.

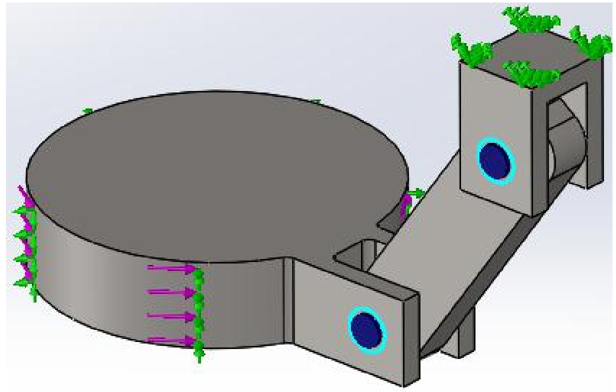


Figure 83. Caster locking mechanism FEA fixtures (green) and applied loads (purple).

8.4.2 Model Details

The model has stress singularities at the surface of the pin. To avoid the singularities from preventing convergence, two points are analyzed to show convergence of stress both at the bottom corner of the lower tabs and near the pin connection on the locking bar. The points can be seen in Figure 84 where point B is actually beneath the tab on the surface of the caster locking bar.

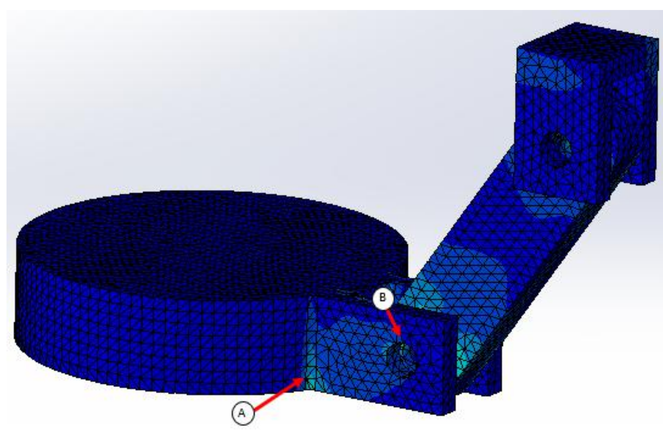


Figure 84. Convergence points on the caster locking mechanism.

Both point A and B converge around 36,700 nodes. Point A has a Von Mises stress of 15,300 psi as shown in Figure 85 while Figure 86 shows point B converging around 19,100 psi.

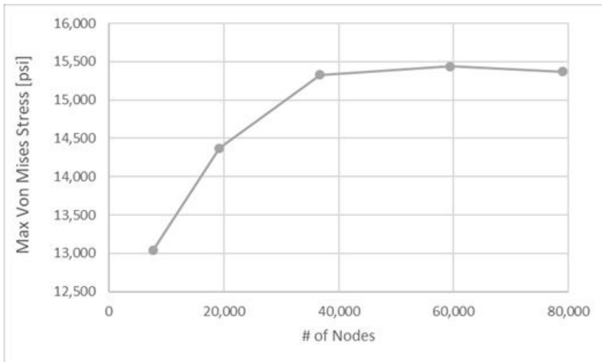


Figure 85. Stress convergence at point A on the caster locking mechanism.

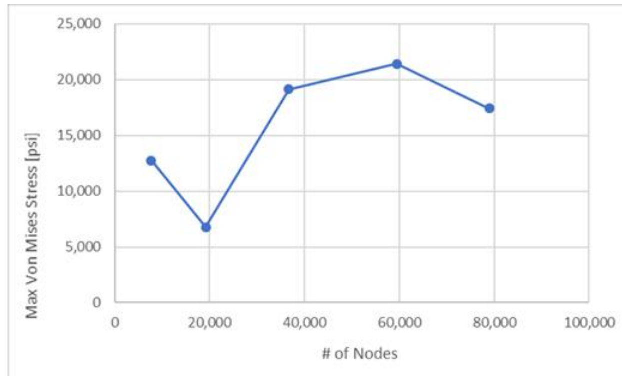


Figure 86. Stress convergence at point B on the caster locking mechanism.

Detail for the final mesh used for the caster locking mechanism analysis is shown in TABLE XIX.

TABLE XIX:
CASTER LOCKING MECHANISM FEA FINAL MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	78989
Total Elements	52515
Element Aspect Ratios < 3	99.8%
Element Aspect Ratios > 10	0.0%
% Distorted Elements	0.0%

8.4.3 Results

The final stress distribution for the caster locking mechanism shows the largest stresses at the pin connections on the locking bar. The stress distribution for the locking bar is displayed in Figure

87. Apart from the singularity at the surface of the pin connection, the model converges well below the yield stress of the material.

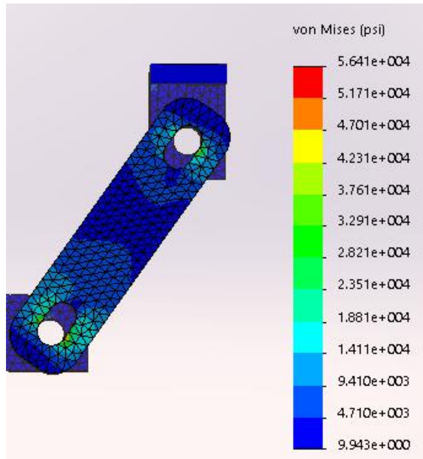


Figure 87. Caster locking bar Von Mises stress distribution.

8.5 Vertical Support Sliders

The vertical support sliders experience all applied loads from the casters, similar to the horizontal beam. The slider shown in blue in Figure 88 is analyzed in this section with a small section of the horizontal tube included as a fixed geometry as shown in Figure 89. The analysis focuses on the stresses around the flat plate where the jack is welded due to the large vertical loads applied by the rear hitch.

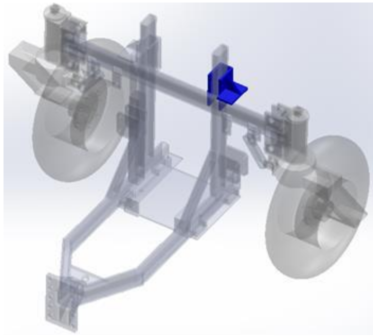


Figure 88. Vertical support slider FEA analysis section.

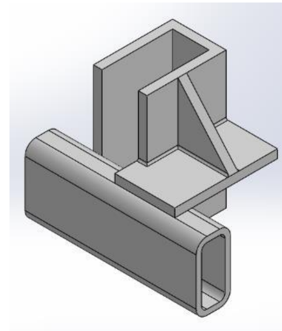


Figure 89. Actual vertical support slider FEA model.

8.5.1 Loads and Fixtures

The largest force applied to the vertical support slider is the tongue weight applied to a 2.5-inch square section on the horizontal plate simulating tongue weight transferred through the jack. Two other loads are applied to the walls where the vertical support is located to simulate the drag load and sideways load from the caster wheels. The model is fixed at the ends of the rectangular tube section. The loads and fixtures are shown in Figure 89 a and b as purple and green arrows, respectively.

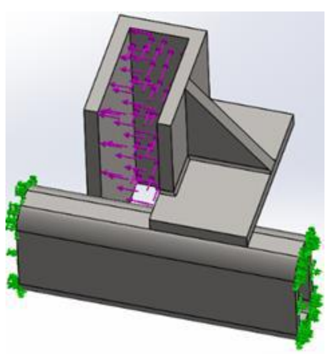


Figure 89a. Vertical support slider FEA loads and fixtures.

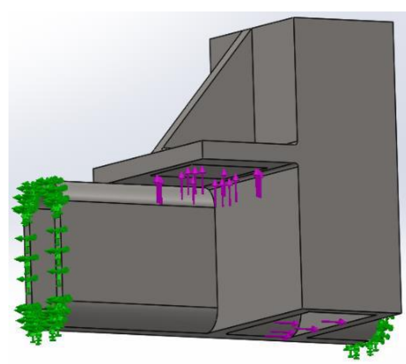


Figure 89b. Vertical support slider FEA loads and fixtures (alternate view).

8.5.2 Model Details

The maximum Von Mises stress in the frame converges around 50,000 nodes at a maximum stress of 21,700 psi as shown in Figure 90.

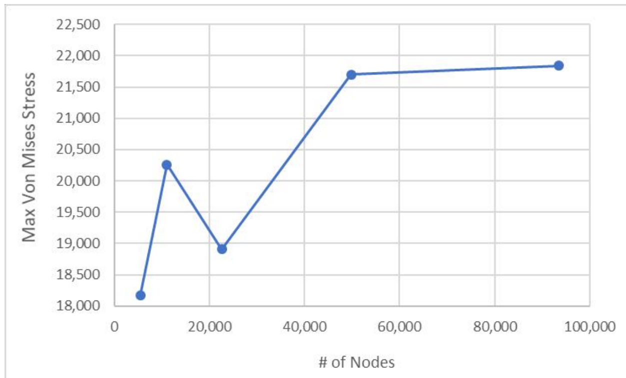


Figure 90. Frame FEA convergence plot.

The details of the final mesh used in this analysis are shown in TABLE XX.

TABLE XX:
VERTICAL SUPPORT SLIDERS FEA MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	93,413
Total Elements	57,425
Element Aspect Ratios < 3	99.6%
Element Aspect Ratios > 10	0.0%
% Distorted Elements	0.0%

8.5.3 Results

The final Von Mises stress distribution in the slider is shown in Figure 91. The largest stress results in an equivalent Factor of Safety for the beam equal to 4.4. The location of the largest stresses is where the slider connects to the beam. Other large stresses are found on at the fillets on the gusset and around the edges of the horizontal plate.

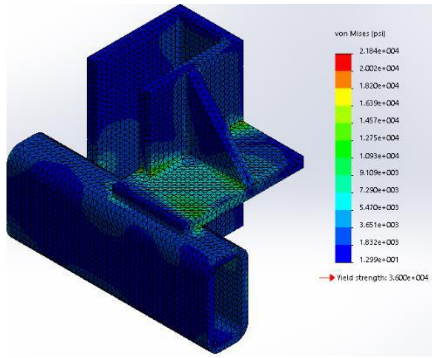


Figure 91. Vertical support slider FEA Von Mises stress distribution results.

8.6 Front Hitch Attachment Category Five Connection

The front hitch attachment is the section of the load cart that the connects load cart to the tow vehicle. The front hitch bracket is a load bearing component on the cart and will see the total draft loading force being applied by the implement.

8.6.1 Hand Calculations

Hand calculations were performed for the front hitch attachment. The loading scenario for the front hitch connection is complex but was simplified for hand calculation purposes. The maximum bearing stress calculated in the plate was 40,000 psi. This stress was compared to the maximum stress found after performing FEA on the part. The detailed hand calculations can be observed in Appendix D.

8.6.2 FEA Model Simplification

The FEA model changes can be observed between Figure 92 and Figure 93. The beam was cut down in the model to achieve computational efficiency. The edges of the FEA setup were filleted to minimize stress concentrations and simulated welds.

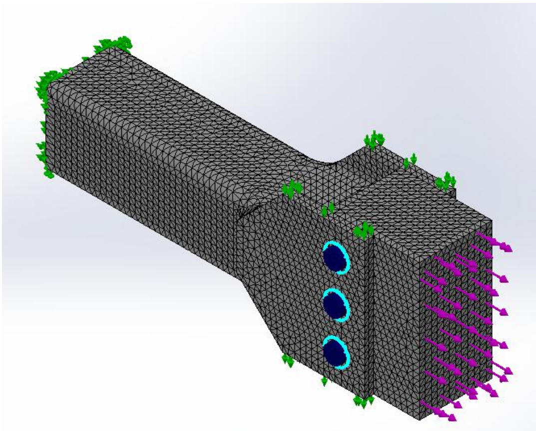


Figure 92. Front hitch class 5 attachment FEA setup.

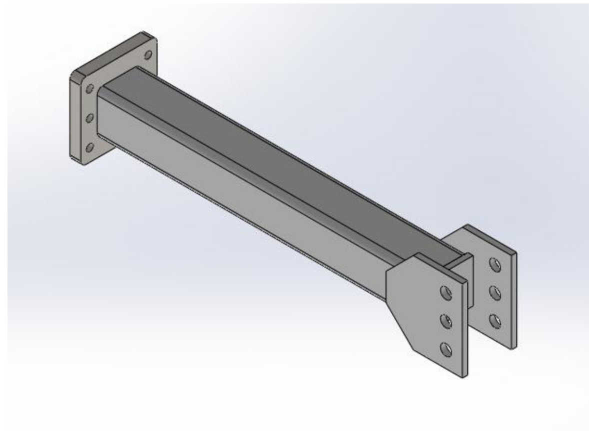


Figure 93. Front hitch extension attachment for FEA analysis.

8.6.3 Loads and Fixtures

During general operation, the load cart is connected from the front hitch attachment to the drawbar of the tow vehicle. The front hitch attachment will see the full draft load that is applied by the implement. The draft loading conditions used in the simulation can be observed in TABLE XIV. The loading conditions used in this simulation were based on the maximum weight of a class 5 tractor. The maximum forces a tractor can pull is equal to its weight or the tractor begins to slip. The average maximum weight of a tractor with a category 5 hitch is 60,000 lbs. Therefore, to maintain a minimum safety factor of 2 a load of 120,000 lbs was applied to the front hitch attachment. The loading scenario is illustrated in Figure 94.

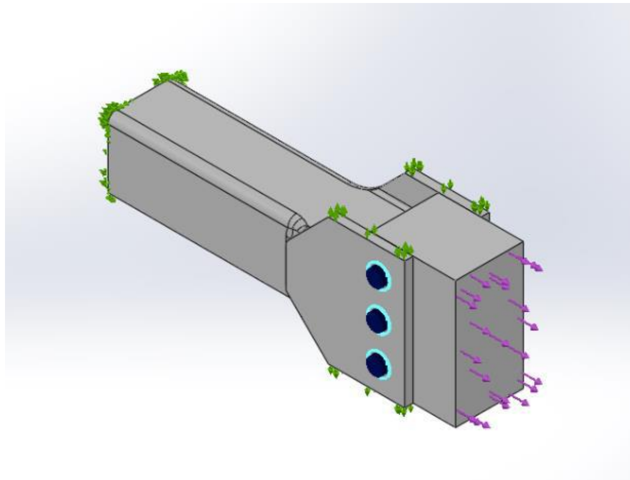


Figure 94. Front hitch attachment load conditions.

The front hitch setup was fixed at the end of the square beam that has the bracket plates welded to the side of it. The bracket connections have roller bearings to inhibit vertical motion. This is done to simulate the hitch being connected to the clevis restricting most vertical motion. The front hitch will see only minor vertical loading and displacement. Therefore, the use roller bearings accurately simulate the loading scenario that the front attachment will see in general operation. The green fixtures can be observed in Figure 95.

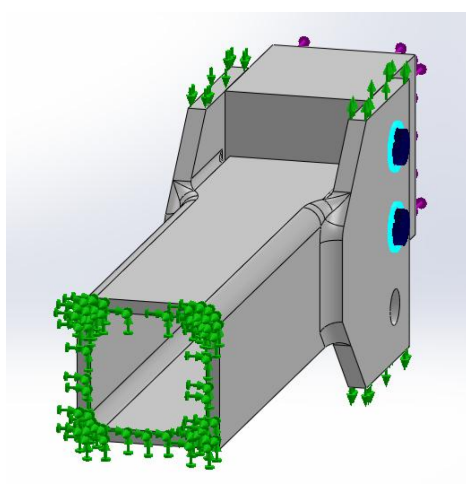


Figure 95. Front hitch FEA fixtures.

8.6.4 Model Details

The maximum Von Mises stress for the front hitch connection approximately converges around 57,000 nodes at a maximum stress of 31,440 psi as shown in Figure 96.

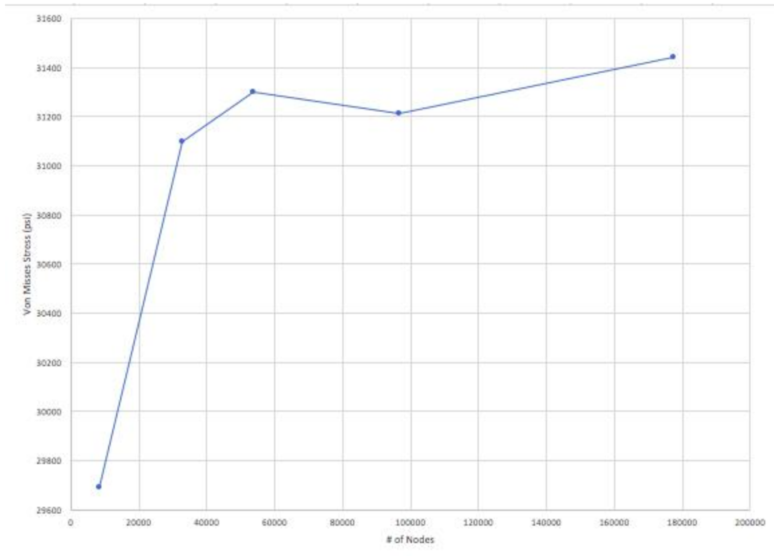


Figure 96. Front hitch FEA convergence plot.

The details of the final mesh used in this analysis are shown in TABLE XXI.

TABLE XXI:
FRONT HITCH CATEGORY FIVE FEA MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	177,512
Total Elements	113,759
Element Aspect Ratio < 3	99.9%
Element Aspect Ratio > 10	0.00%
% Distorted Elements	0.00%

8.6.5 Results

The final von misses stress distribution for the front hitch connection are shown in Figure 97. The largest stress was observed at Point A which corresponds to a stress of 44,940 psi. This is above the yield strength of the material but is considered to be a singularity or stress concentration. The stresses at this point are considered to be infinite in the simulation will increase when the mesh is made finer. Therefore, since the stress is theoretically diverging to infinity the singularity or stress concentration will be omitted from the analysis. When omitting the singularity, the largest stress the front hitch connection exhibited was 31,450 psi.

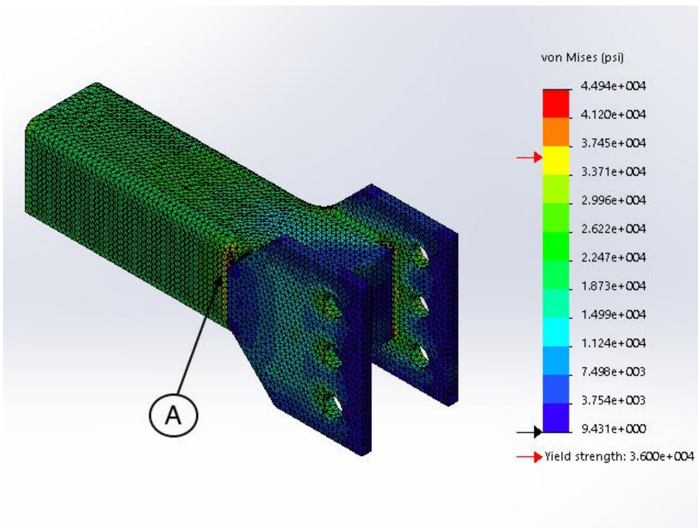


Figure 97. Front hitch FEA von mises distribution

The deformation of the front hitch connection is largest at front portion of the bracket which corresponds to a deformation of .0115”. This deformation is considered negligible and the full distribution can be observed in Figure 98.

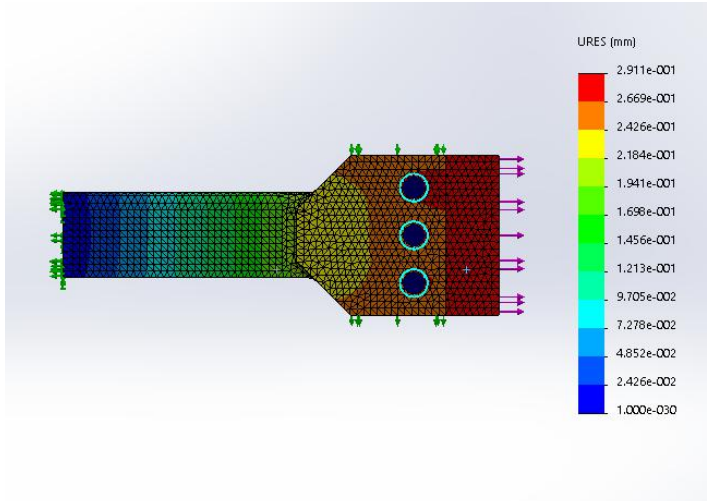


Figure 98. Front hitch FEA deformation distribution.

8.7 Front Hitch Attachment Category Three Connection

The front hitch attachment is the section of the load cart that the connects load cart to the tow vehicle. The front hitch bracket is a load bearing component on the cart and will see the total draft loading force being applied by the implement. The FEA model changes can be observed between Figure 99 and Figure 100. For this analysis, the bending of the beam was not considered.

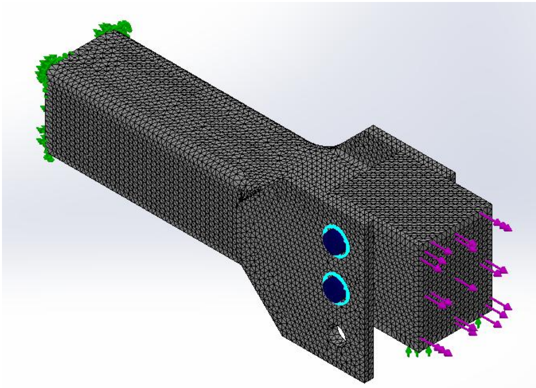


Figure 99. Front hitch class 3 attachment FEA setup.

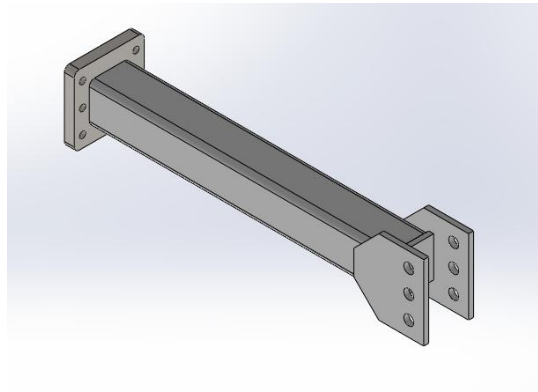


Figure 100. Front hitch extension attachment for FEA analysis.

8.7.1 Loads and Fixtures

The loading conditions used in this simulation were based on the maximum weight of a class 3 tractor which can be observed in TABLE XIV. The maximum forces a tractor can pull is equal to its weight or the tractor begins to slip. The average maximum weight of a tractor with a category 3 hitch is 40,000 lbs. Therefore, to maintain a minimum safety factor of 2 a load of 80,000 lbs was applied to the front hitch attachment. The load was applied to the block that is pinned in two places to the bracket as illustrated in Figure 101.

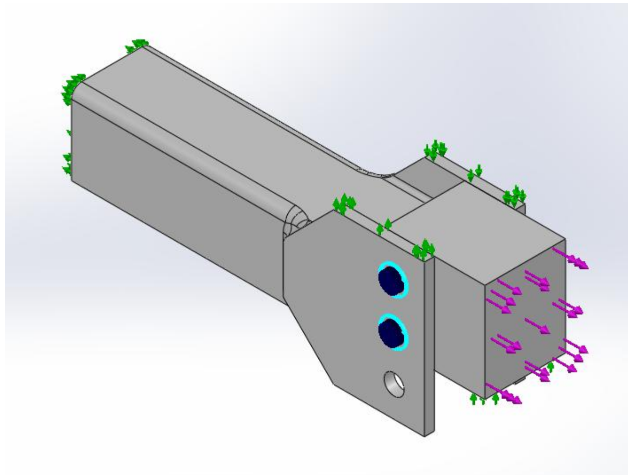


Figure 101. Category 3 front hitch loading scenario.

The fixtures used for the category three simulation were the exact same ones used for a category five hitch.

8.7.2 Model Details

The maximum Von Mises stress for the front hitch connection approximately converges around 90,000 nodes. The two nodes probe yield a maximum stress of approximately 34,000 psi as shown in Figure 102.

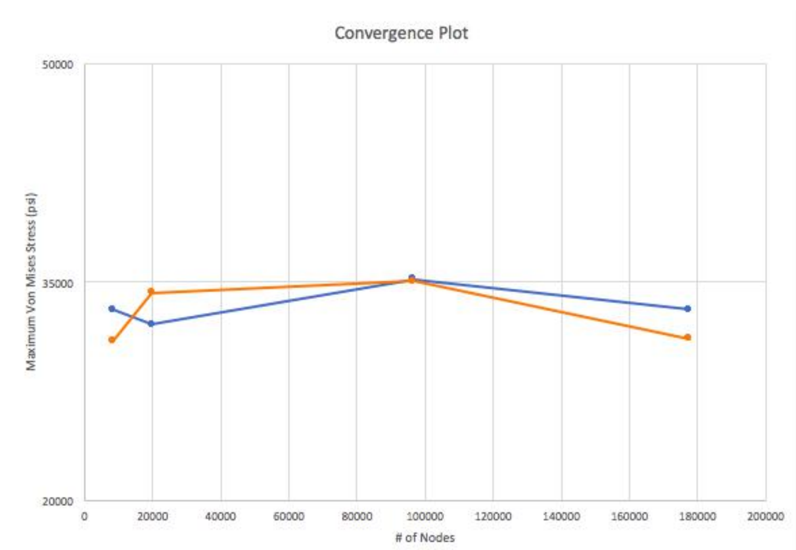


Figure 102. Category three front hitch FEA convergence plot.

The details of the final mesh used in this analysis are shown in TABLE XXII.

TABLE XXII:
FRONT HITCH CATEGORY THREE FEA MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	177,512
Total Elements	113,759
Element Aspect Ratio < 3	99.9%
Element Aspect Ratio > 10	0.00%
% Distorted Elements	0.00%

8.7.3 Results

The final von misses stress distribution for the front hitch connection are shown in Figure 103. The largest stress was observed at the pin connection in Figure 104, which corresponds to a stress of 44,990 psi. This is above the yield strength of the material but is considered to be a singularity or stress concentration. The stresses at this point are considered to be infinite in the simulation will increase when the mesh is made finer. Therefore, since the stress is theoretically diverging to infinity the singularity or stress concentration will be omitted from the analysis. When omitting the singularity, the largest stress the front hitch connection exhibited was 34,100 psi.

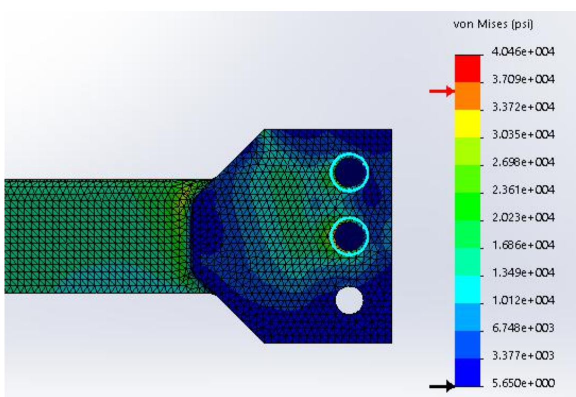


Figure 103. Category three front hitch Von mises stress distribution

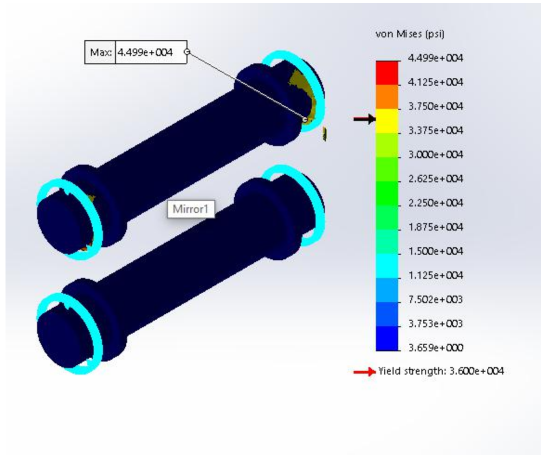


Figure 104. Singularities at pin connection.

The deformation of the front hitch connection is largest at the top right portion of the bracket which corresponds to a deformation of .0087”. This is due to the uneven loading condition of the two-hole category 3 setup. This deformation is considered negligible and the full distribution can be observed in Figure 105.

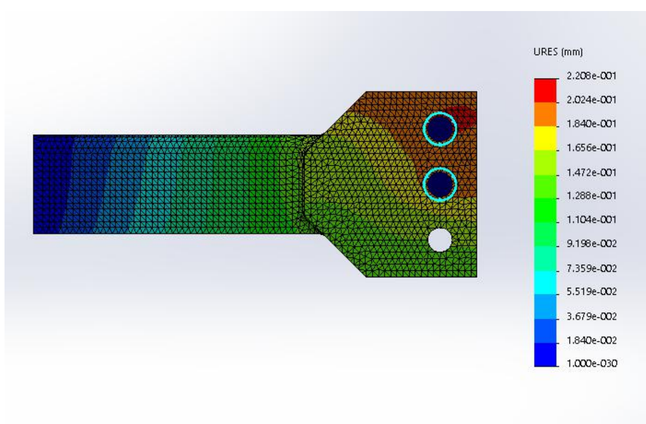


Figure 105. Category three front hitch FEA deformation distribution.

8.8 Vertical Load Cell Bracket

The vertical load cell bracket is the section of the load cart that the connects load cell to the PowerPin and frame. The vertical load cell bracket is a load bearing component on the cart and will see the total draft loading force being applied by the implement.

8.8.1 Hand Calculations

Hand calculations were performed for the vertical load cell bracket. The loading scenario for the vertical load cell bracket is complex but was simplified for hand calculation purposes. The maximum bearing stress calculated in the plate was 40,000 psi. This stress was compared to the maximum stress found after performing FEA on the part. The detailed hand calculations can be observed in Appendix D.

8.8.2 FEA Model Simplification

The major change for the FEA model was adding fillets to all edges to simulate welds and reduce stress concentrations. The FEA model changes can be observed between Figure 106 and Figure 107.

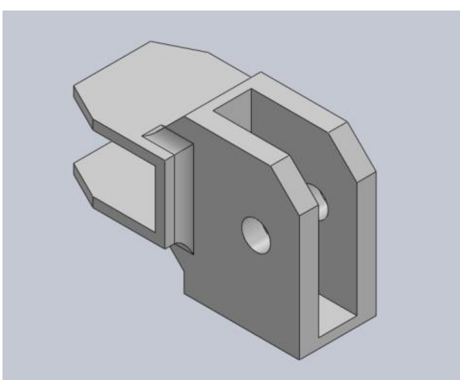


Figure 106. Vertical load cell bracket FEA setup with fillets.

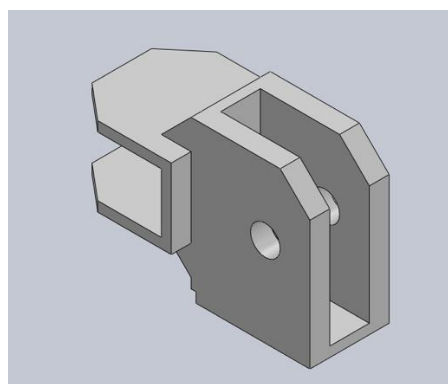


Figure 107. Vertical load cell bracket component

8.8.3 Loads and Fixtures

The loading conditions used in this simulation were based on the maximum weight of a class 5 tractor which can be observed in TABLE XIV. The maximum forces a tractor can pull is equal to its weight or the tractor begins to slip. The average maximum weight of a tractor with a category 5 hitch is 60,000 lbs. Therefore, to maintain a minimum safety factor of 2 a load of 120,000 lbs was applied to the front hitch attachment. The load was applied to the block that is pinned through the hole in the bracket as illustrated in Figure 108.

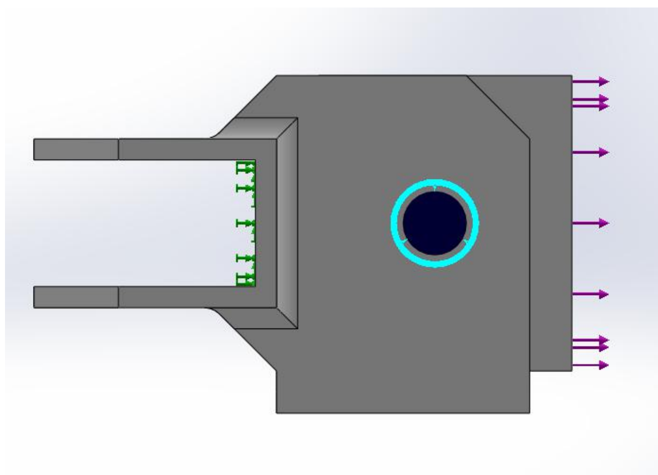


Figure 108. Vertical load cell bracket FEA loading scenario.

The vertical load cell setup was fixed on the rear of the plate. This fixture simulates the bracket being welded all around the edges of the back plate onto the frame. The fixture allows for an accurate simulation of the loading scenario that the vertical load cell bracket will see in general operation. The green fixtures can be observed in Figure 108.

8.8.4 Model Details

The maximum Von Mises stress for the vertical load cell bracket approximately converges around 90,000 nodes. The two nodes probe yield a maximum stress of approximately 29,000 psi as shown in Figure 109.

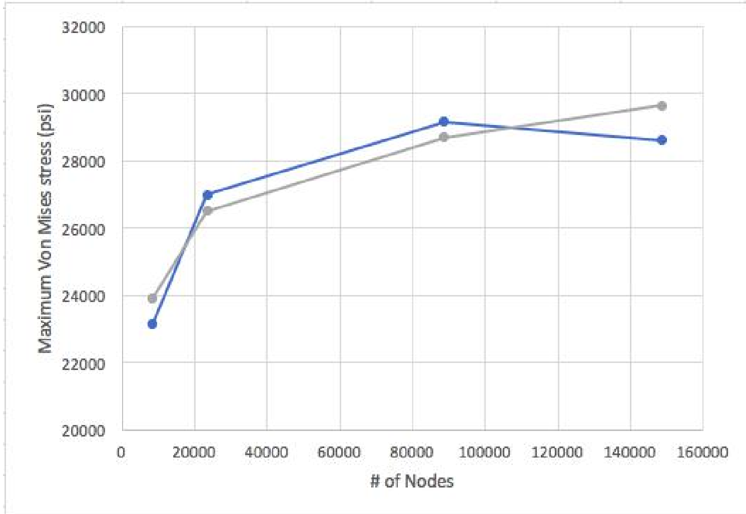


Figure 109. Category three front hitch FEA convergence plot.

The details of the final mesh used in this analysis are shown in TABLE XXIII.

TABLE XXIII:

VERTICAL LOAD CELL BRACKET FEA MESH DETAILS

Mesh Type	Tetrahedral Solid Mesh
Total Nodes	177,512
Total Elements	113,759
Element Aspect Ratio < 3	99.9%
Element Aspect Ratio > 10	0.00%
% Distorted Elements	0.00%

8.8.5 Results

The final von misses stress distribution for the vertical load cell bracket are shown in Figure 110. The largest stress was observed at the pin connection which corresponds to a stress of 30,310 psi. With an applied load of 120,000 lbs this indicates that the bracket will not fail with a high degree of certainty. The factor of safety of the bracket is 2.16.

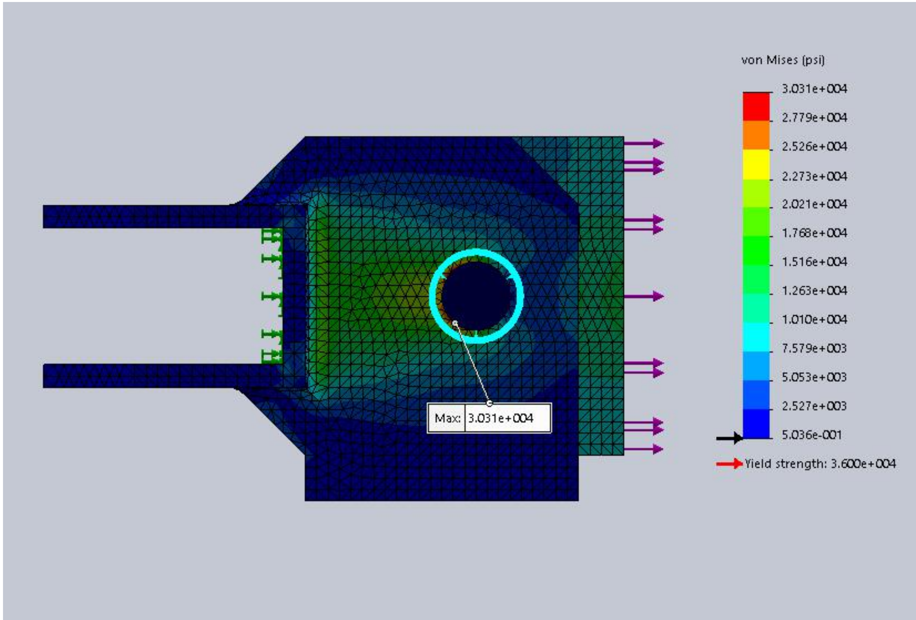


Figure 110. Vertical load cell bracket Von mises stress distribution.

The deformation of the vertical load cell bracket is largest at the far-left portion of the bracket which corresponds to a deformation of .0038". This deformation is considered negligible and the full distribution can be observed in Figure 111.

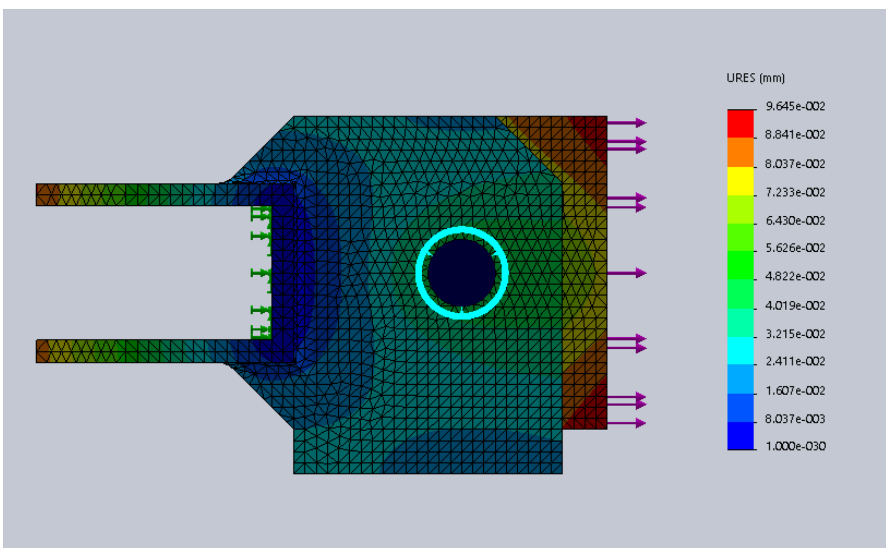


Figure 111. Vertical load cell bracket FEA deformation distribution.

8.9 Summary of FEA Results

A summary table of the all load bearing components maximum stress, deformations, and factor of safety can be observed in TABLE XXIV.

TABLE XXIV:

SUMMARY OF FEA RESULTS

Component	Applied Load (lbs)	Maximum Stress (psi)	Maximum Deformation (in)	Factor of Safety
Frame	7,233	15,450	0.036	6.2
Front Hitch Extension	120,000	24,690	-	4.4
Front Hitch Extension(Offset)	60,000	34,500	-	2
Horizontal Beam	7,033	13,731	0.09521	3.35
Caster Locking Mechanism	10,000	19,100	-	2.4
Vertical Support Sliders	6,700	21,840	-	4.4
Front Hitch Connection(3-hole)	120,000	31,450	0.0115	2.15
Front Hitch Connection(2-hole)	120,000	34,100	0.0087	2.05
Vertical Load Cell Bracket	120,000	30,310	0.0038	2.18

9 Conclusion

The main goal of this project was to redesign a testing load cart for the client, PAMI. The design report started with the project background outlining PAMI and the various features of the old load cart design. The background illustrated the issues with the old load cart including the cart wheels skidding through turns, lack of compatibility with various towing machines and implements, and poor ergonomics when installing the load cell. The following section presented the project definition including the project objectives, deliverables, client needs, metrics and the constraints and limitations. Next, a summary of the concept selection which outlined the six different design aspects selected for the load cart design. The concept selection process included detailed concept generation with weighted criterion for the final selection including manufacturability, cost, physical attributes and compatibility.

The final design was detailed in eight different sections. This included swivel casters, horizontal support, main frame, manual jacks, PowerPin (Rear Hitch), rear hitch support (PTFE bearings), load cell integration, and front hitch connection. All project objectives are achieved within these eight design sections. The design is compatible with all pre-purchased parts and test equipment. The pre-purchased parts implemented into the re-design are Bourgault caster wheels and a PowerPin rear hitch. The test equipment that the load cart is compatible with includes load cells with capacities of 20,000, 50,000, and 100,000 pounds. The design is also compatible with a wide range of towing machines and implements including compatibility with category 3 to 5 hitches at the front and rear.

By incorporating the swivel casters in the new load cart, the wheels will follow the direction of travel of the load cart. This allows the new design to obtain more accurate load cell measurements in the field compared to the old solid axle design.

The design complies with all relevant PAMI and ASABE standards. A finite element analysis performed on the load bearing sections of the design verify with a high degree of certainty that a safety factor of two has been met. This proves compliance with the safety factor requirement outlined by PAMI. The new load cart also includes a safety chain at the connection between the tow vehicle and load cart to comply with requirements stated in ASAE S338.5.

The design has improved test setup time and ergonomics compared to the old load cart. First, a vertical load cell installation process is integrated for easy installation. Second, clearance between the load cell connectors and bracket is increased to allow for load cell connector alignment during installation. Lastly, screw adjusters are added to allow the load cell connectors and brackets to align for pin installation.

All other requirements specified by the customer are also satisfied by the new load cart. The load cart can be transported on a 7ft x 18ft trailer when being moved to the testing location. The load cart is height adjustable with two manual top-wind crank jacks to accommodate drawbar heights ranging from 15 to 24 inches. An extension beam was installed at the front of the load cart to ensure adequate distance between the load cart and tow machine to avoid any interference. The load cart includes four sets of hydraulic quick couplers, a bullseye level and front hitch adjustment jack. Finally, the load cart is capable of performing compressive load testing with the addition of main frame attachments.

The materials used for the design have been split up into the following five categories: frame, fasteners and pins, hydraulic system, PTFE bearings, and jacks and bullseye level. The total material cost is $5,228. This leaves $4,772 for manufacturing cost which includes all machining, welding, and laser cutting required for the parts and final assembly.

FMEA was conducted on possible adverse outcomes with the new load cart design. The major concerns with the design include weld or material failure at the front and rear load cell connections. Another concern from the analysis is foreign material striking the load cell during operation. Both failures would cause major damage to the load cell and are difficult to detect before occurrence.

The team's redesign of the testing load cart will expand PAMI's testing capabilities. This will provide more accurate data for agriculture producers and manufacturers in the prairies resulting in more informed decision making.

10 Recommendations

A list of possible improvement for the overall design of the load cart are as follows.

1. Include routing and mounting points for the hydraulic system so can be easily integrated into the main frame of the load cart.
2. Repurpose the original PowerPin base plate as a safety chain slot on the extension beam. This will allow a safety chain to be connected from the tractor to the load cart using strengthened steel at no added cost.
3. Mount the rolling swivel jack near the end of the extension beam to allow for front hitch height adjustment to easily load and unload the load cart off of the trailer.
4. Integrate a mounting system at the front of the A-frame to allow two more extension beam connections. These extension beams would attach to the towing machine to keep the load cart straight during braking allowing PAMI to conduct compressive braking load tests.
5. Reduce the number of quick disconnect hydraulic couplers on the rear of the load cart to keep the total cost of materials under $5,000.
6. Add height indicators on each jack. This would allow the two jacks to be leveled without requiring the load cart to be on even ground (unlike the current bullseye level method).

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Appendix A: Concept Generation

Contents

List of Figures

List of Tables

The detailed concept generation is covered in this section.

Overarching Weighting Criteria

Manufacturability and cost criterion were used in all sections as weighted criteria for concept selection.

Cost: Cost is a crucial factor to consider for selection criteria. The project budget is broken down into two sections: materials and parts, and manufacturing costs. The budget is $5,000 dollars for materials and parts, and $5,000 dollars for manufacturing cost. Cost will be considered in all comparison matrices throughout this concept development and selection sections.

Manufacturability: The manufacturing of the load cart will be completed by PAMI in house. Therefore, it is important to be conscious of the manufacturing capabilities of PAMI's shop. The load cart components should use basic manufacturing processes where possible. Manufacturability will be considered in all comparison matrices throughout the concept development and selection sections.

Caster Mounting

PAMI has purchased two Bourgault casters that are approximately 40 inches tall with 30-inch diameter wheels installed. The casters are mounted at the top as shown in **Figure 1** and has wheels that can rotate a full 360 degrees. Using the swivel casters will allow the wheels to follow the motion of the load cart eliminating the previous issue of the load cart wheels skidding during turns.

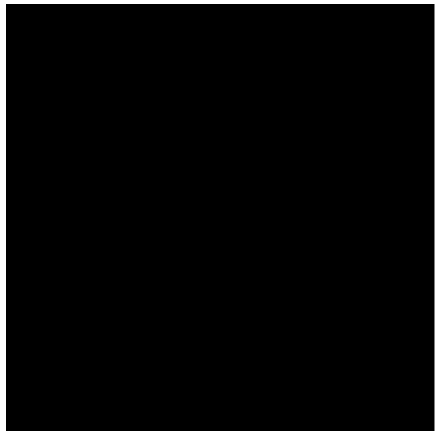


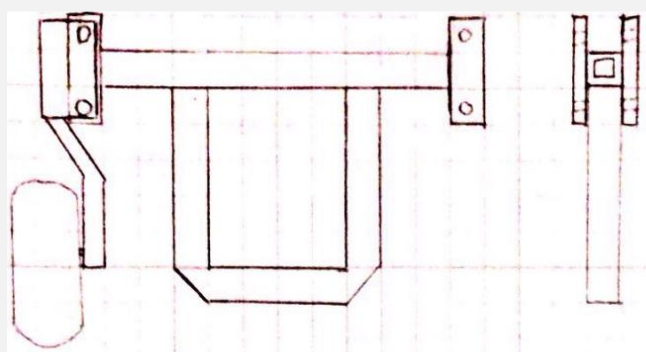
Figure 1. Bourgault caster assembly implementation example [1]**.**

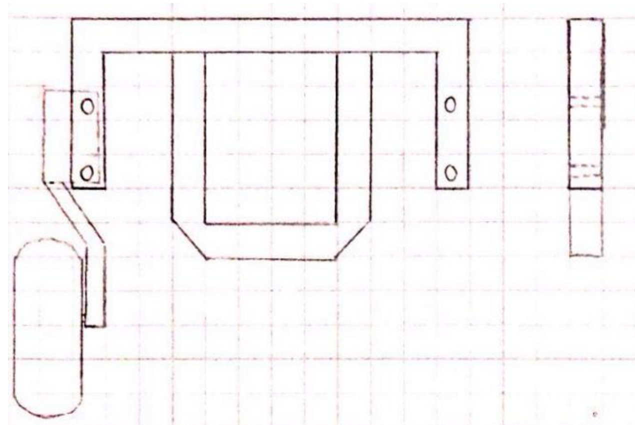
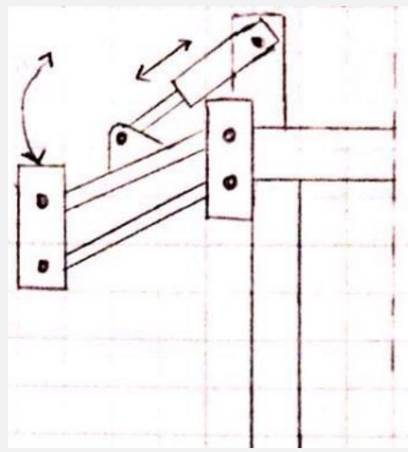
Concept Generation

A summary of the names and figures for each caster mounting concept is displayed in

TABLE I.

TABLE I:
CASTER MOUNTING CONCEPT SUMMARY

Concept	**Image of Concept**
Horizontal Beam	

U Beam	
Dual Link Connection	

Weighted Selection Criteria

Caster mounting concepts were weighed against five criteria including cost and manufacturability. The definitions for all other criteria are explained as follows:

Bending Strength: The ability of the system to resist upwards force at both caster mounting points resulting in a bending moment.

Weight: A lighter design will result in improved cart maneuverability and correlates to reduced materials.

Compatible with Casters: The casters require mounting points that offer clearance from the frame to avoid interference during operation.

The hit matrix in **Figure 2** shows how the weight of each criteria was determined for caster mounting concepts.

Criteria	ID	A (Cost)	B (Manufacturability)	C (Bending Strength)	D (Weight)	E (Compatible with Casters)
Cost	A		A	C	A	E
Manufacturability	B			B	B	E
Bending Strength	C				D	C
Weight	D					E
Compatible with Casters	E					
Total Hits		2	2	2	1	3
Weightings [%]		20.00	20.00	20.00	10.00	30.00

Figure 2. Caster mounting concept hit matrix.

Concept Selection

The concept selection matrix in **Figure 3** shows the weights determined in the previous section to rank each concept. The result of the matrix is a tiered ranking of each concept with the top rank as the selected design.

		Concepts					
		A		B		C	
		Horizontal Beam		*U-Beam*		*Dual Link Connection*	
Selection Criteria	*Weight*	*Rating*	*Weighted score*	*Rating*	*Weighted score*	*Rating*	*Weighted score*
Cost	*20*	5	100.0	5	100.0	3	60.0
Manufacturability	*20*	5	100.0	4	80.0	2	40.0
Bending Strength	*20*	5	100.0	3	60.0	5	100.0
Weight	*10*	5	50.0	4	40.0	5	50.0
Compatible with Casters	*30*	4	120.0	5	150.0	5	150.0
Total Score		470.0		430.0		400.0	
Rank		1		2		3	
Continue?		Yes		No		No	

Figure 3. Caster mounting concept selection matrix.

Based on the selection matrix in **Figure 3** the horizontal beam concept was selected for the final design. The ratings selected for each concept is explained further.

Concept A: Horizontal Beam *(Selected Concept)*

The horizontal beam excels in each criterion due to the simplicity and overall strength of the design. A possible minor issue with the design is interfering with other features since the beam travels across the frame horizontally. The final ranking results in the selection of the horizontal beam design for mounting the caster wheels.

Concept B: U-Beam

The U-beam mitigates the interference issue with the horizontal beam by raising the beam relative to the frame. The alternate design has the downside of reducing manufacturability and increasing weight. Also, the downward extensions on the beams makes the design more susceptible to bending moments.

Concept C: Dual Link Connection

The dual link connection offers formidable bending strength at a lower weight. Although the design has the possibility of superior performance, the complex design is difficult to manufacture and costly. The fallbacks of the design results in the lowest overall rating compared to the other concepts. Due to the very limited budget of this project the dual link design has a much lower rank than the other concepts.

Height Adjustability

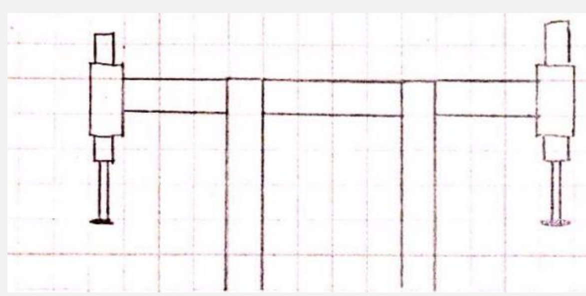
The load cart needs to vary in height to be compatible with different towing machines and implement types. The current design has two trailer jacks with a hand crank at the top as shown clearly on the left side of **Figure 4**. These cranks allow for precise height adjustment and the full range of motion required to accommodate different hitch heights. There is also a bubble level at the center of the top beam between the jacks.

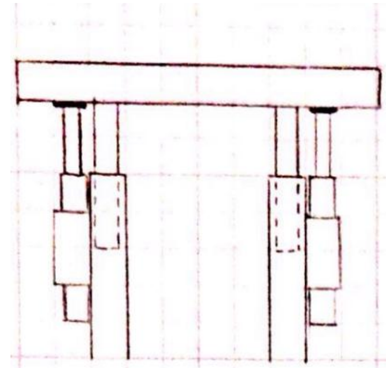
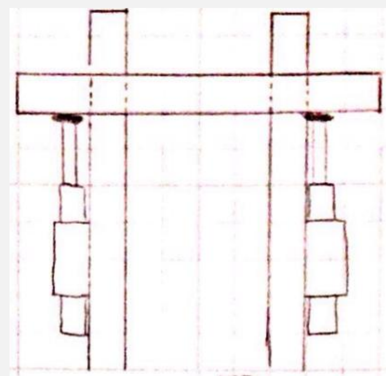
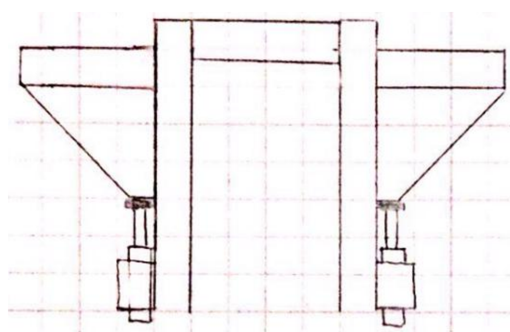
Figure 4. Current design with weld-on trailer jacks.

Concept Generation

A summary of the names and figures for each height adjustability concept is displayed in **TABLE II**.

TABLE II:
HEIGHT ADJUSTABILITY CONCEPT SUMMARY

Concept	**Image of Concept**
Direct Caster Jacks	

Inner Guide Beam	
Outer Guide Beam	
Webbed Independent Beam	

Weighted Concept Criteria

Height adjustability concepts were weighed against five criteria including cost and manufacturability. The definitions for all other criteria are explained as follows:

Height Variability: The minimum height variability of the frame is 12 inches, but more variability is advantageous for added compatibility.

Strength: The height adjustable feature will be exposed to forces from the jacks and casters. The concept must be able to resist the applied forces.

Ease of Use: The height adjustment will be done in the field and should be accessible from ground level. Smooth motion of the design is also desired.

The hit matrix in **Figure 5** shows how the weight of each criteria was determined for height adjustment concepts.

Criteria	ID	A (Cost)	B (Manufacturability)	C (Height Variability)	D (Stength)	E (Ease of Use)
Cost	A		A	C	A	A
Manufacturability	B			C	D	E
Height Variability	C				D	B
Stength	D					D
Ease of Use	E					
Total Hits		3	1	2	3	1
Weightings [%]		30.00	10.00	20.00	30.00	10.00

Figure 5. Height adjustment concept hit matrix.

Concept Selection

The concept selection matrix in **Figure 6** used the weights determined in the previous section to rank each concept. The result of the matrix is a tiered ranking of each concept with the top rank as the selected design.

		Concepts							
		A		B		C		D	
		Direct Caster Jacks		Inner Guide Beam		Outer Guide Beam		Webbed Independent Beam	
		Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	30	5	150.0	4	120.0	5	150.0	4	120.0
Height Variability	10	5	50.0	5	50.0	5	50.0	5	50.0
Manufacturability	20	5	100.0	3	60.0	4	80.0	4	80.0
Stength	30	1	30.0	4	120.0	4	120.0	2	60.0
Ease of Use	10	1	10.0	3	30.0	5	50.0	4	40.0
			0.0		0.0		0.0		0.0
			0.0		0.0		0.0		0.0
Total Score		340.0		380.0		450.0		350.0	
Rank		4		2		1		3	
Continue?		No		No		Yes		No	

Figure 6. Height adjustability concept selection matrix.

Based on the selection matrix in **Figure 6** the outer guide beam concept was selected for the final design. The ratings selected for each concept is explained further below.

Concept A: Direct Caster Jacks

Having the jack system directly attached to the casters results in the lowest cost and best manufacturability. The issue with the design is the large non-vertical forces applied to the jacks. This would cause deformation resulting in bad strength and ease of use ratings.

Concept B: Inner Guide Beam

The inner guide system is simple and low cost with great height variability. The issue with this concept is the tolerance required to have interconnected tubes. The tight tolerances result in difficulty for manufacturing and height adjustment.

Concept C: Outer Guide Beam *(Selected Concept)*

The beam with guides on the outside of the vertical frame supports has all the advantages of Concept B without the negative aspects. Manufacturing will be similar to the current load cart and the design provides enough strength while remaining low in cost. This concept is substantially better than the rest and will be considered for the final design.

Concept D: Webbed Independent Beams

The main drawback of having the independent beams is the large moment applied to the vertical frame supports due to the forces from the caster wheels. The result is inadequate strength for the design to resist the applied moments.

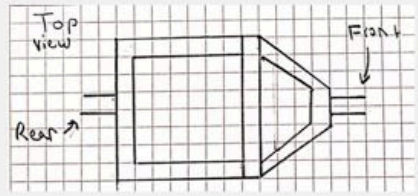
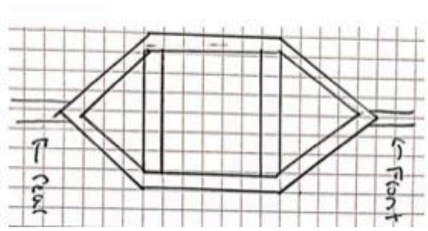
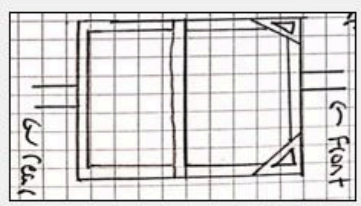
Frame Design

The frame is an aspect of the load cart that must be able to integrate with an extension beam at the front of the frame and a PowerPin attachment at the rear. The frame must also contain ample space for load cell integration.

Concept Generation

The three main frame concepts are displayed in **TABLE III**.

TABLE III:
FRAME CONCEPT SUMMARY

Concept	Image of Concept
A-Frame	
Diamond Frame	
H-Frame	

Weighted Concept Criteria

Frame concepts were weighed against seven criteria including cost and manufacturability. The definitions for all other criteria are explained as follows:

Space required for load cell mounting: The frame must provide an adequate amount of space to mount a load cell. The load cell will be mounted in the interior of the frame.

Ability to integrate extension beam: The frame must have the ability to integrate an extension beam that will allow proper clearances between the load cart and tow machine. The extension beam will be bolted onto the front of the frame.

Ability to integrate rear hitch connection: The frame must have the ability to integrate a PowerPin rear hitch connection. The PowerPin will connect to the implement to allow for a variety of test setups.

Ability to withstand high tensile and compressive loading scenarios: The load cart will experience high tensile and compressive loads while in use. The frame must be able to withstand all loading conditions used while testing. The frame will experience heavy tensile load forces up to 100,000 lbs as the implement is pulled by the load cart, and in turn high compressive forces during breaking. The potential for shock and bending loads is also apparent due to the wide variety of conditions and soil types that the load cart will be used in.

Geometry will allow ease of transportation: The geometry of the frame will affect its ability to be transported to various location on a trailer. The shape of the frame should accommodate the user in easy loading and unloading.

A hit matrix was created to determine weights for each selection criteria, displayed in **Figure 7**.

Criteria	ID	A	B	C	D	E	F	G
		Space required for load cell mounting	Ability to integrate extension beam	Ability to integrate rear hitch connection	Cost	Manufacturability	Ability to withstand high tensile and compressive loading	Geometry allows easy transport
Space required for load cell mounting	A		A	A	A	A	F	A
Ability to integrate extension beam	B			C	B	B	F	B
Ability to integrate rear hitch connection	C				C	C	F	C
Cost	D					E	F	D
Manufacturability	E						F	G
Ability to withstand high tensile and compressive loading	F							F
Geometry allows easy transport	G							
Total Hits		5	3	4	1	1	6	1
Weightings [%]		23.81	14.29	19.05	4.76	4.76	28.57	4.76

Figure 7. Hit matrix for the frame selection criteria.

Concept Selection

The result of the concept ranking is displayed in **Figure 8** which shows a matrix with a tiered ranking of each concept with the top rank as the selected design.

		Concepts					
		A		B		C	
		A-Frame		Diamond Frame		H-Frame	
Selection Criteria	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Space required for load cell mounting	23.8	5	119.1	5	119.1	3	71.4
Ability to integrate extension beam	14.3	5	71.5	3	42.9	5	71.5
Ability to integrate rear hitch connection	19.1	4	76.2	3	57.2	4	76.2
Cost	4.8	5	23.8	4	19.0	3	14.3
Manufacturability	4.8	5	23.8	5	23.8	5	23.8
Ability to withstand high tensile and compressive loading	28.6	5	142.9	5	142.9	5	142.9
Geometry allows easy transport	4.8	5	23.8	4	19.0	5	23.8
Total Score		457.2		404.8		400.0	
Rank		1		2		3	
Continue?		Yes		No		No	

Figure 8: Frame concept selection matrix

Based on the selection matrix in **Figure 8** the A-frame concept was selected for the final design. The ratings selected for each concept is explained further.

Concept A: A-Frame *(Selected Concept)*

The A-frame scored well in all selection criteria due to its simplistic and strong design. It had an ample amount of space for load cell mounting and could easily implement the extension beam with a bolt on connection. The frame geometry is much like a trailer which will allow it to be easily transportable and manufacturable. Overall this was the concept selected to move forward with into the final design.

Concept B: Diamond Frame

The diamond frame scored well in most selection criteria categories due to its robust shape and ability to withstand high compressive and tensile loading. Due to the extra members compared to the A-frame concept there would have been an increase in cost. The connection at the front of the diamond frame would have made it difficult to integrate the extension beam. For these reasons the diamond frame was not selected to move forward into the final design.

Concept C: H-Frame

The H-frame scored well in most selection criteria categories due to its overall geometry and ability to integrate front and rear hitch connections. Due to the central member that supports the H-frame there would have been issues with load cell mounting. The overall geometry of the frame allowed it to withstand high tensile and compressive loading but with the additional members it would have costed more than both the A-frame and diamond frame concepts. For these reasons the H-frame was not selected for the final design.

PowerPin Integration

On the current load cart design the rear hitch of the load cart is built into the frame of the load cart. To accommodate class 4 and 5 hitches PAMI purchased a PowerPin drawbar system for the rear hitch. This system will connect the implement to the load cart. The current system for connecting the implement to the load cart can be seen in **Figure 9**.

Figure 9. Current rear hitch support on load cart.

The PowerPin has to be incorporated in the redesign of the testing load cart to improve safety when attaching the implement to the load cart. The PowerPin must be implemented in a way that prevents any moment from being transferred from it to the load cell. The PowerPin drawbar is shown in **Figure 10** and **Figure 11**.

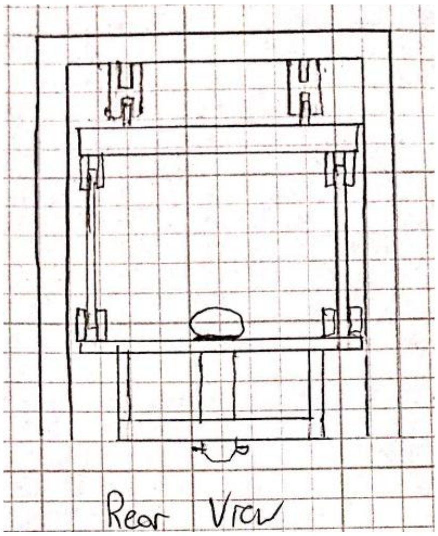
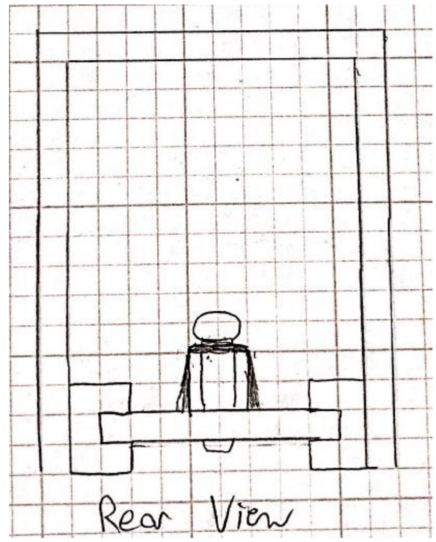
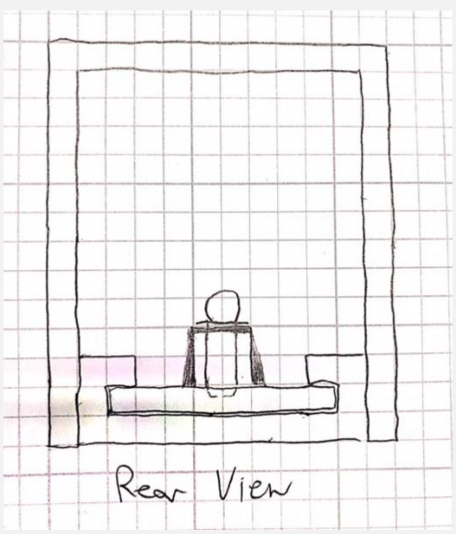
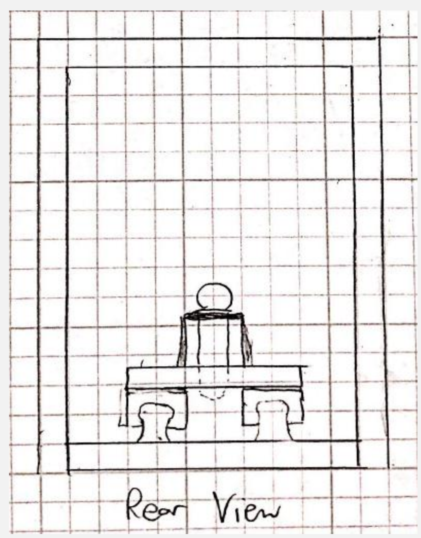
Figure 10. PowerPin drawbar rear view.

Figure 11. PowerPin drawbar top view.

Concept Generation

A summary of the names and figures for each PowerPin concept is displayed in **TABLE IV**.

TABLE IV:
POWERPIN CONCEPT SUMMARY TABLE

Hinged Design 	**Sliding Channel** 
Bearing Plate 	**Rail Bearings** 

Weighted Concept Criteria

The PowerPin integration concepts were weighed against five criteria including cost and manufacturability. The definitions for all other criteria are explained as follows:

Strength: The ability of the system to withstand the vertical loads applied on it by the implement, as well as shock loads from uneven terrain.

Compatibility with PowerPin: The PowerPin must be able to be fixed to the design, and not interfere with the frame of the load cart.

Elimination of Bending Moments in the Load Cell: The PowerPin integration system must ensure that no bending load is being transferred from it into the load cell. If the load cell experiences a bending moment it will result in inaccurate readings with possible damage to the load cart occurring.

Compatibility with Load Cell Integration System: The PowerPin integration system must be adjustable to allow for the three different load cells

A hit matrix was used to determine the weight of each criterion for the PowerPin integration concepts, as seen in **Figure 12**.

Criteria	ID	A	B	C	D	E
		Cost	Manufacturability	Compatibility with PowerPin	Stength	Elimination of Bending Moments in Load Cell
Cost	A		A	A	D	A
Manufacturability	B			C	D	B
Compatibility with PowerPin	C				D	E
Stength	D					D
Elimination of Bending Moments in Load Cell	E					
Total Hits		3	1	1	4	1
Weightings [%]		30.00	10.00	10.00	40.00	10.00

Figure 12. PowerPin integration concept hit matrix.

Concept Selection

Using the weights determined by the hit matrix the four concepts were ranked using the selection matrix in Figure 13.

		Concepts							
		A		B		C		D	
		Hinged Design		Sliding Channel		Bearing Plate		Rail Bearings	
Selection Criteria	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	*30*	4	120.0	4	120.0	4	120.0	2	60.0
Manufacturability	*10*	5	50.0	5	50.0	5	50.0	5	50.0
Compatibility with PowerPin	*10*	4	40.0	5	50.0	5	50.0	5	50.0
Strength	*40*	5	200.0	5	200.0	5	200.0	3	120.0
Elimination of Bending Moments in Load Cell	*10*	3	30.0	5	50.0	5	50.0	5	50.0
Total Score		440.0		470.0		470.0		330.0	
Rank		3		1		1		4	
Continue?		No		No		Yes		No	

Figure 13. Selection matrix for PowerPin integration.

Based on the selection matrix the most viable option was the bearing plate design. The ratings selected for each concept are further explained below.

Concept A: Hinged Design

The hinged design is already proven as a working concept on the current load cart. It was able to be manufactured by PAMI in house and has withstood 15 years of testing. However, the cost of the design is high due to the addition of material needed for the support structure of the

PowerPin. Additionally, this concept would allow the PowerPin to pivot at the hinges which could result in a bending moment introduced into the load cell if the PowerPin is not properly mounted.

Concept B: Sliding C-Channel Design

The sliding C-channel design concept excels in all criterion as it is a simple concept that fully supports the PowerPin. The C-channel could be used with PTFE sliders for a low-cost solution. A disadvantage to this concept is limited area of contact between moving parts. PTFE sliders require a large surface between moving sections to reduce friction.

Concept C: Bearing Plate *(Selected Concept)*

The bearing plate concept has all the advantages of the C-channel design, with the added advantage of a large surface area for the PTFE sliders. The large surface area reduces the pressure and consequentially friction between the two moving parts to improve performance. The low-cost bearing and advantages over the C-channel concept results in the bearing plate being selected as the final concept despite have the same score in the selection matrix.

Concept D: Rail Bearings

The main two disadvantages of the rail bearing concept were cost and strength, which go hand in hand. Most rail bearings cannot support very high loads, and the ones that can are very expensive.

Load Cell Integration

The load cell integration refers to how the load cell will be mounted on the load cart. Mounting should ensure that only tensile and compressive loads are measured, and that the installation process is quick and ergonomic. PAMI has already manufactured threaded rods that connect to either side of the load cell. The rods have 1" thick steel plates with centred holes to allow for pin connections similar to the sketch in **Figure 14**.

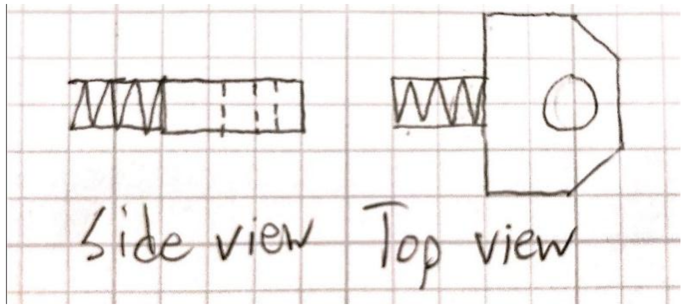


Figure 14. Load cell mounting plates.

There are two of these threaded rods on either side of the load cell to allow for a pin connection to the front and rear of the load cart as shown in **Figure 15** and **Figure 16**. The current load cart used by PAMI uses a solid horizontal H-bracket for the front and rear of the load cell.

Figure 15. Top view of current load cell integration system with the front of the load cart at the bottom of the figure.

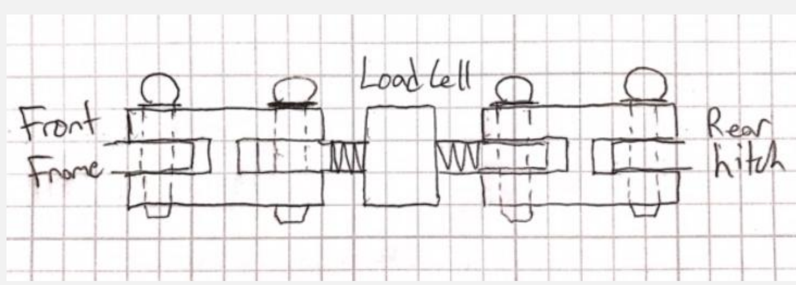
Figure 16. Side view of the current load cell integration system.

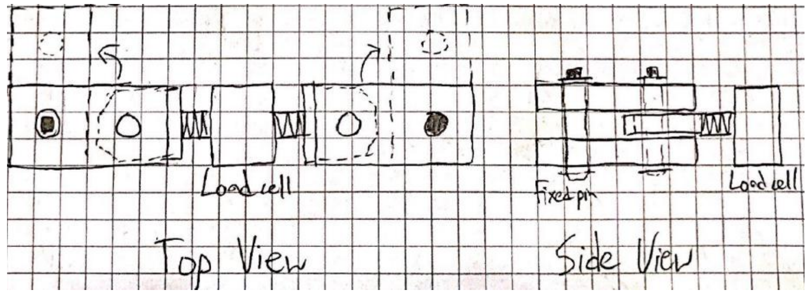
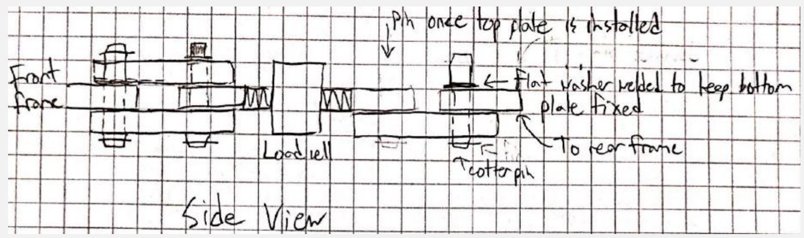
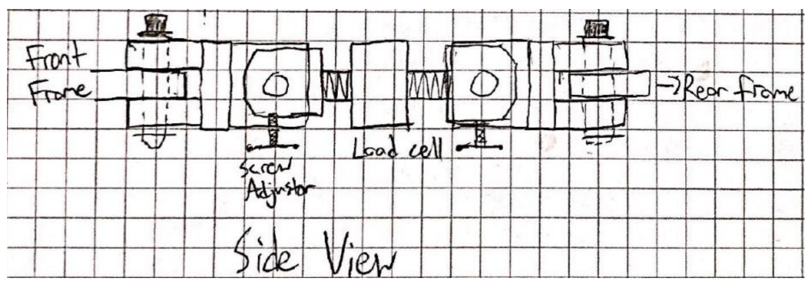
The main constraint for this section was incorporating the three load cells. The current process for load cell integration results in safety issues for the employee as they have to hammer the connecting pins into place and support the 40 lb load cell while trying to fit it in the tight-fitting H-brackets.

Concept Generation

A summary of the names and figures for each load cell integration concept is displayed in TABLE V.

TABLE V:
LOAD CELL INTEGRATION CONCEPTS

Concept	Image of Concept
Solid H Design	

Rotating Top Plate	
Removable Top Plate	
Vertical Install	

Weighted Concept Criteria

Load cell integration concepts were weighed against five criteria including cost and manufacturability. The definitions for all other criteria are explained as follows:

Ability to integrate load cell mounting plates: The mounting plates for the load cell can be installed and removed from the system with minimal effort. The load cell mounting plates currently have to be hammered into place and lined up with pry bars. This adds potential safety concerns for the employee as well as possible damage to the load cell.

Strength: The load cart experiences high tensile and compressive loads while in use. The load cell integration system must be able to withstand all loading conditions used while testing. The maximum load cell rating that PAMI has is 100,000 lbs, considering the minimum factor of safety of 2 all parts of the load cell installation system will have to withstand 200,000 lbs.

Vertical Movement Restriction: The load cell must avoid bending loads to prevent binding. The vertical movement of the system must be limited to ensure accurate readings from the load cell.

Using the criteria above a hit matrix was developed to determine weights for each selection criteria, illustrated in **Figure 17**.

		Cost	Manufacturability	Ability to integrate load cell mounting plates	Stength	Vertical Movement Restriction
Criteria	**ID**	**A**	**B**	**C**	**D**	**E**
Cost	**A**		B	C	D	A
Manufacturability	**B**			C	D	E
Ability to integrate load cell mounting plates	**C**				C	C
Stength	**D**					D
Vertical Movement Restriction	**E**					
Total Hits		3	1	1	4	1
Weightings [%]		30.00	10.00	10.00	40.00	10.00

Figure 17. Load cell integration concept hit matrix.

Based on the hit matrix in **Figure 17** the most important aspect of the concept is its ability to integrate the load cell mounting plates followed closely by strength.

Concept Selection

The result of the concept ranking is displayed in **Figure 18** which shows a matrix with a tiered ranking of each concept with the top rank as the selected design.

Selection Criteria	Weight	Concepts							
		A		B		C		D	
		Solid H Design		Rotating Top Plate		Removable Top Plate		Vertical Install	
		Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	*10*	5	50.0	5	50.0	5	50.0	5	50.0
Manufacturability	*10*	5	50.0	5	50.0	5	50.0	4	40.0
Ability to integrate load cell mounting plates	*40*	3	120.0	4	160.0	4	160.0	5	200.0
Strength	*30*	5	150.0	4	120.0	4	120.0	5	150.0
Tolerance	*10*	4	40.0	3	30.0	2	20.0	5	50.0
Total Score		410.0		410.0		400.0		490.0	
Rank		2		2		4		1	
Continue?		No		No		No		Yes	

Figure 18. Load cell integration concept selection matrix.

Based on the selection matrix the Vertical Install concept was the most viable concept for development. The ratings selected for each concept are explained further below.

Concept A: Solid H Design

The solid H design is the simplest design and would have been very easy to manufacture. The thickness of the steel plates used to manufacture the design can be modified to provide the appropriate amount of strength required to support the tensile and compressive loads. The fallbacks of the solid H design are its lack of ability to integrate with the load cell mounting plates. This would not be user friendly, and the load cell mounting plates could not easily be installed or removed.

Concept B: Rotating Top Plate

The rotating top plate design would allow for easier integration of the load cell mounting plates, as the employee can set the load cell mounting bracket right onto the bottom plates of the design. However, when the top plate is allowed to rotate it acts independently of the bottom plate. This would result in alignment issues for the load cell as the load cell mounting plates might have been able to slightly move up and down in between the top and bottom plates of the design. This movement could introduce loads that are not intended to be measured.

Concept C: Removable Top Plate

Similar to the rotating top plate design, the employee could set the load cell mounting bracket right onto the bottom plates of the design. The removable top plates would have needed to be fastened securely with bolts to ensure that the load cell could not move vertically. This would add time to the load cell installation process and also has the potential to loosen during test. This movement could introduce unintended loads to the load cell.

Concept D: Vertical Install *(Selected Concept)*

The vertical install design with screw adjustment exceled in all categories due to its simplicity and the ability for the load cell to be lowered into position with the help of gravity. The screw adjusters can align the mounting plates which takes strain off the employee. The vertical install design is slightly harder to manufacture than the other concepts, however it is not beyond the manufacturing capabilities of PAMI and the companies they work with.

Front Hitch Integration

The front hitch attachment includes two components. An extension beam that attaches to the load cart frame and second, a hitch attachment to the end of the extension beam, shown in **Figure 19**. This connection point must be designed for the full 100,000 lb load that may be applied to the load cart in accordance with need #3. A typical application of a PowerPin front hitch connection to a tow vehicle drawbar is depicted in **Figure 20**. The hitch attachments currently used by PAMI are depicted in **Figure 21** and **Figure 22**.

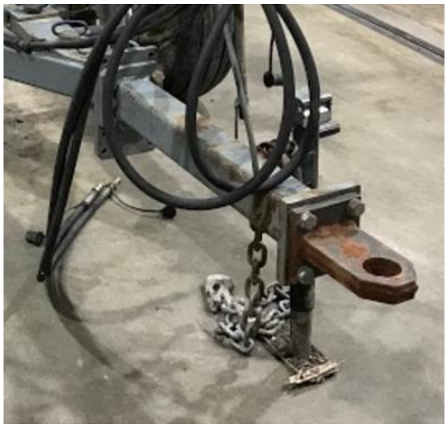


Figure 19. Front hitch attachment to extension.

Figure 20. Application of PowerPin hitch connection (in red).

Figure 21. Category 4 and 5 front PowerPin hitch.

Figure 22. Category 3 front PowerPin hitch.

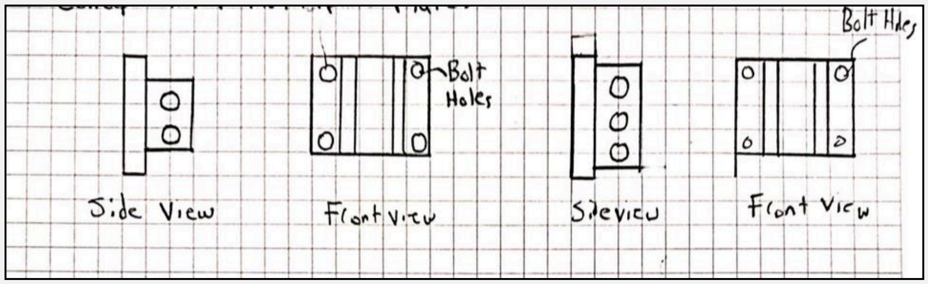
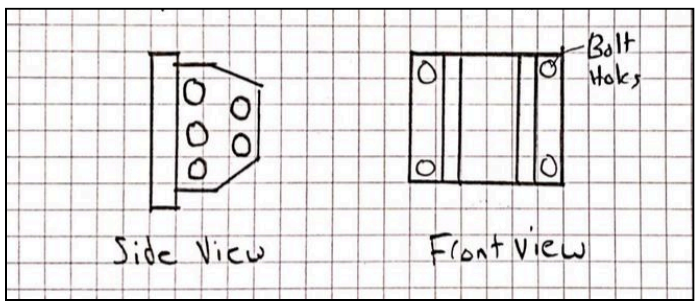
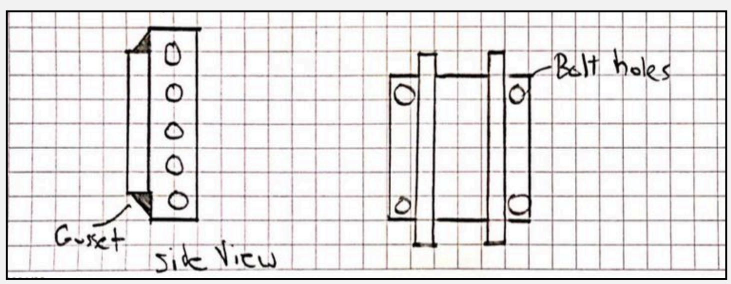
The major constraints for the front hitch attachment are:

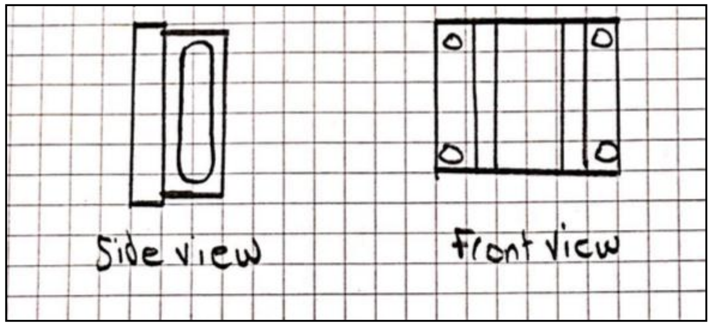
- Compatibility with category 3-5 hitches.
- Ability to integrate with the load cart
- Minimum factor of safety of 2.

Design Concepts

Four main concepts were generated and are shown in **TABLE VI**.

TABLE VI:
FRONT HITCH ATTACHMENT CONCEPT SUMMARY

Concept	Image of Concept
Multiple Brackets	
Multi-Adapter	
Multi-Holes	

The Slot	

Weighted Concept Criteria

The seven main criteria the project team decided on include cost and manufacturability. The other five criteria are explained further below.

Hitch height adjustability: The front hitch had to be height adjustable from 13 inches to 24 inches to connect to the drawbar. Additionally, the drawbar height varies with tires size and inflation pressure.

Ability to integrate category 3-5 hitches: The front hitch attachment had to accommodate category 3-5 hitches. This is important due to the various types of tow vehicles being tested.

Strength: The front hitch attachment must have adequate strength to withstand high vertical loads and tensile loads as high as 100,000 lbs as per need #3.

Ease of connection with drawbar: The drawbar connection to the front hitch must be a straight forward process. The process must reduce the time required by the PAMI technician for connection.

Ease of hitch category conversion: The front hitch must be able to convert between different category hitches in a timely manner. This is required by PAMI test personnel to reduced downtime and increase field test time.

The hit matrix in **Figure 23** shows how the weight of each criteria was determined for front hitch concepts.

Criteria	ID	A (Cost)	B (Manufacturability)	C (Hitch height adjustability)	D (Ability to integrate category 1-5 hitches)	E (Strength)	F (Ease of connection with drawbar)	G (Ease of hitch category conversion)
Cost	A		A	C	D	E	F	G
Manufacturability	B			C	D	B	B	G
Hitch height adjustability	C				D	C	F	G
Ability to integrate category 1-5 hitches	D					D	D	D
Strength	E						F	G
Ease of connection with drawbar	F							G
Ease of hitch category conversion	G							
Total Hits		1	2	3	6	1	3	5
Weightings [%]		4.76	9.52	14.29	28.57	4.76	14.29	23.81

Figure 23. Front hitch hit matrix.

Concept Selection

The matrix in **Figure 24** uses the weights determined in the previous section to rank each concept. The result of the matrix is a ranking of each concept with the highest rank as the selected design.

		Concepts							
		A		B		C		D	
		Multiple Brackets		Multi-Adapter		Multi-Holes		Slotted	
Selection Criteria	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	4.8	2	9.5	5	23.8	5	23.8	3	14.3
Manufacturability	9.5	3	28.6	3	28.6	4	38.1	2	19.0
Hitch height adjustability	14.3	1	14.3	1	14.3	4	57.2	4	57.2
Ability to integrate category 1-5 hitches	28.6	5	142.9	4	114.3	4	114.3	2	57.1
Strength	4.8	4	19.0	2	9.5	4	19.0	2	9.5
Ease of connection with drawbar	14.3	5	71.5	4	57.2	5	71.5	5	71.5
Ease of hitch category conversion	23.8	3	71.4	5	119.1	3	71.4	4	95.2
	Total Score	357.1		366.7		395.2		323.8	
	Rank	3		2		1		4	
	Continue?	No		No		Yes		No	

Figure 24. Front Hitch Concept Selection Matrix

Concept A: Multiple Brackets

Having multiple brackets allows for a high strength and the ability to integrate category 3-5 hitches with ease. The drawback of this design was the lack of height adjustability and the increased cost of manufacturing two separate brackets.

Concept B: Multi-Adapter

Having the adaptability for all category hitches allows for a cost effective and easy to use design. The drawback of this design was the lack of height adjustability and the layout of the holes which would lead to decreased strength properties.

Concept C: Multi-Holes *(Selected Concept)*

Having multiple vertical holes integrated into the design allowed for ample height adjustability, cost effectiveness, and simple manufacturing while maintaining high strength properties. The main drawback of this design is the time-consuming process of fully removing the hitch to change the height.

Concept D: Slotted

Having a large slot running down the side of the bracket allows for quick, continuous height adjustability and ease of connection with the drawbar. The drawbacks were that the strength properties were reduced compared to the other designs. There was also a potential for slippage caused by shock loading to the hitch. This could cause the hitch to slide resulting in a misalignment of the load cart and alter the load cell angle with respect to the towing machine and implement.

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Appendix B: Total Cost Breakdown

List of Tables

The total cost of all materials, along with part numbers, quantities, and individual material cost is provided in TABLE I.

TABLE I:
TOTAL COST OF ALL MATERIALS [1], [2], [3], [4], [5], [6], [7], [8], [9], [10], [11]

Component	Part Number	Supplier	Quantity	Unit Cost	Total Cost
Frame					
1" Steel Plate (2'x3')	N/A	Brunswick Steel	1	$569.00	$569.00
0.5" Steel Plate (2'x5')	N/A	Brunswick Steel	1	$258.00	$258.00
2x2, 1/4" Wall Thickness Steel tubing	N/A	Brunswick Steel	6	$5.10	$30.60
2x4, 3/8" Wall Thickness Steel Tubing	N/A	Brunswick Steel	22	$11.20	$246.40
4x4, 5/16" Wall Thickness Steel Tubing	N/A	Brunswick Steel	3	$14.00	$42.00
Subtotal					$1,146.00
Fasteners & Pins (USD)					
Hitch Pin,Bent,Stl,Zinc,3/4X3 1/2L	EBP3HLT5	Acklands Grainger	4	$6.75	$27.00
1-1/2" x 8" Effective Length Yellow Zinc Plated Hitch Pin	120709	Fastenal	2	$59.92	$119.84
1-1/4"-7 x 6.5" Grade 8 Plain Finish Hex Cap Screw	14576	Fastenal	4	$26.56	$106.24
1-1/4"-7 Grade 8 Plain Finish NE Steel Nylon Insert Lock Nut	37182	Fastenal	4	$33.91	$135.64
1-1/4" ASTM F436 Type 1 Plain Steel Structural Flat Washer	33124	Fastenal	8	$2.55	$20.40
1"-8 x 6-1/2" Grade 8 Plain Finish Hex Cap Screw	14476	Fastenal	3	$15.49	$46.47
1"-8 Grade 8 Plain Finish Hex Nut	36419	Fastenal	3	$6.21	$18.63
1" Plain Finish Medium Split Lock Washer	33635	Fastenal	3	$0.82	$2.46

1" x 2.000" OD Thru-Hardened Plain Finish Steel SAE General Purpose Flat Washer	33805	Fastenal	6	$1.45	$8.70
5/8"-11 x 3" Grade 8 Plain Finish Hex Cap Screw - TAA Certified	11107891	Fastenal	6	$2.03	$12.18
5/8" Plain Finish Medium Split Lock Washer	33629	Fastenal	6	$0.19	$1.14
5/8"-11 Plain Finish Grade 8 Finished Hex Nut	11137100	Fastenal	6	$0.89	$5.33
5/8" x 1.312" OD Thru-Hardened Plain Finish Steel SAE General Purpose Flat Washer	33802	Fastenal	6	$0.63	$3.80
1/2"-13 x 4-1/4" Grade 5 Plain Finish Hex Cap Screw	12220	Fastenal	4	$2.02	$8.08
1/2"-13 Plain Finish Grade 5 Finished Hex Nut	36309	Fastenal	4	$0.27	$1.09
1/2" Plain Finish Medium Split Lock Washer	33625	Fastenal	4	$0.10	$0.42
1/2"-13 x 1" ASTM F879 Hex Drive Grade 316 Stainless Steel Flat Socket Cap Screw	0 178225	Fastenal	4	$10.51	$42.04
1/2"-13 x 1-3/4" Grade 5 Plain Finish Hex Cap Screw	12210	Fastenal	2	$0.89	$1.77
1/2"-13 x 1" Grade 5 Plain Finish Hex Cap Screw	12205	Fastenal	2	$0.62	$1.24
Subtotal USD					$562.47
Subtotal CAD					$748.09
Hydraulic System					
Lever Coupler Break-Away Kit, Quick Disconnect Adapter	FRM 95024TSC	Napa	4	$255.89	$1,023.56
4216-08-10 NPT Male Crimp Fitting	882974641119	Green Line Hose	16	$16.50	$264.00
5/8" 122-10:Reeled SAE100R2AT Hydraulic Hose	882974172453	Green Line Hose	112	$5.90	$660.80
1/2" NPT Hydraulic Quick Coupler Tip	8350845	Princess Auto	8	$7.99	$63.92

Allstar ALL18304 5/8" Aluminum Line Clamp, (Pack of 10)	N/A	Amazon	2	$19.17	$38.34
1/2 in. x 36 in. Plain Steel Round Rod	N/A	Home Depot	1	$5.77	$5.77
Subtotal					$2,056.39
PTFE Bearings					
1/2" PTFE Sheet Virgin	N/A	Johnston Plastics	264	$0.81	$213.84
1" PTFE Sheet Virgin	N/A	Johnston Plastics	150	$1.62	$243.00
Subtotal					$456.84
Jacks and Level					
5000 lbs Weld-On Drop Leg Trailer Jack	8055055	Princess Auto	2	$69.99	$139.98
1200 lbs Bolt-On Swivel Trailer Jack	8690075	Princess Auto	1	$69.99	$69.99
Bullseye Spirit Level Plastic Circular Level	HF-60	Amazon	1	$9.22	$9.22
Subtotal					$219.19
Total with Tax					$5,227.96

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Appendix C: FMEA Ranking Descriptions

List of Tables

This appendix presents the tables used to identify the ranking numbers for severity, probability and likelihood of failure modes in the FMEA process. Each potential failure mode was subjected to evaluation using the rating systems in the tables below. The rating scale for severity, probability and likelihood of detection is from 1 to 10, with 10 being the highest.

TABLE I:
SCALE FOR FAILURE SEVERITY [1]

Severity of Effect		Ranking
Minor	Unreasonable to expect that the minor nature of this failure would cause any substantial effect on system performance or on a subsequent process or service operation. Customer unlikely to either notice or care about the failure	**1**
Low	Low severity ranking due to nature of failure causing only a slight customer annoyance. Customer will probably notice only a minor degradation of the service performance or a slight impact on a a subsequent action; i.e., some quick, minor rework	**2**
Moderate	Failure causes some customer dissatisfaction. Customer is made uncomfortable or is annoyed by the failure. Customer will experience some very noticeable inconvenience or performance degradation. May cause either delay due to rework or irreversible damage.	**4,5,6**
High	High degree of customer dissatisfaction due to the negative impact of the failure such as inaccurate payroll run, loss of vital data or an inoperable convenience system. Does not involve safety or noncompliance to government regulations. May cause serious disruption to subsequent processing; may require major rework or loss to customer and/or create significant financial hardship.	**7,8**
Very High	Failure mode involves serious personal safety hazards, potential for civil litigation or noncompliance with government regulations.	**9,10**

TABLE II:
SCALE FOR PROBABILITY OF FAILURE [1]

Probability of Failure	Possible Failure Rates	Ranking
Remote: Failure is unlikely. No failures ever associated with almost identical processes.	>1 in 2	1
Very Low: Process in Statistical Control. Only isolated failures associated with almost identical processes.	1 in 3	2
Low: Process is in Statistical Control. Isolated failures associated with similar processes.	1 in 8	3
Moderate: Generally associated with processes similar to previous processes which have experienced occasional failures, but not in major proportions. Process is in Statistical Control.	1 in 20	4
	1 in 80	5
	1 in 400	6
High: Generally associated with processes similar to previous processes that have often failed. Process is not in Statistical Control.	1 in 2,000	7
	1 in 15,000	8
	1 in 150,000	9
Very High: Failure is almost inevitable	<1 in 1,500,000	10

TABLE III:
SCALE FOR LIKELIHOOD OF DETECTION [1]

Likelihood of Detection		Ranking
Very High	Current controls will almost certainly prevent the failure (process automatically prevents most failures)	**1,2**
High	Current controls have a good chance of detecting the failure	**3,4**
Moderate	Current controls may detect the failure	**5,6**
Low	Current controls have a poor chance of detecting the failure	**7,8**
Very Low	Current controls probably will not detect the failure	**9**
Absolute Certainty of Non-Detection	Current controls will not or cannot detect the failure	**10**

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Appendix D: Hand Calculations

Contents

List of Figures

List of Tables

Hand Calculation/ Theory

To validate the finite element analysis that will be performed on the load bearing sections of the load cart a hand calculation analysis will be performed. The main areas that require hand calculations are:

- Front Hitch Bolts Bearing/ Shear stresses
- Vertical Beam Normal Stress and deflection
- Load Cell Bolt/Pin Connections Bearing/ Shear stresses
- PTFE (sideways forces)

All of these sections will be analyzed using basic hand calculations to verify the finite element analysis on Solidworks.

Shear Stress Theory

A main area where stress concentrations occur is at bolt or pin connections. The two major stresses that will be analyzed are shear and bearing stress.

Shear stress occurs when two forces are applied transverse to a member (bolt or pin). This is show in Figure 1.

Figure 1. Transverse forces on a member causing shear. [1]

The application of these transverse forces causes an internal force at the plane section C. This force is called shear and the resultant shear force at the section is equal to the force (P). The average shearing stress at the section is found by dividing the shear force by the cross sectional area. This is displayed in Equation 1.

$$\tau_{ave} = \frac{P}{A} \qquad \text{Eq.1}$$

Bolt connections can be subjected to single or double shear configurations. A bolt that is in single shear configuration is displayed in Figure 2 [1].

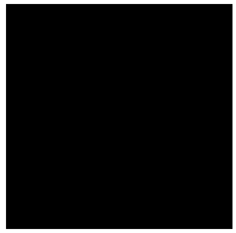


Figure 2. Bolt subject to single shear[1]

Equation 1 is used to determine the average shear stress in a bolt subjected to single shear. A different loading scenario is double shear on a bolt. This is displayed in Figure 3.

Figure 3. Bolts subjected to double shear. [1]

To determine the average shearing stress of a bolt in double shear a free body diagram is used to determine the force distribution. This is shown in Figure 4.

Figure 4. Free body diagram of bolt subjected to double shear. [1]

Based on Figure 4 the average shear stress on the bolt can be determined using Equation 2.

$$\tau_{ave} = \frac{P}{A} = \frac{F/2}{A} = \frac{F}{2A} \qquad \text{Eq.2}$$

Bearing Stress Theory

Bearing stress at bolt or pin connections should be analyzed. This stress is created in the members that are connected by the bolts along the bearing surface. The bolt exerts a force on the plate that is equal and opposite to the force exerted by the plate on the bolt. This is displayed in Figure 5.

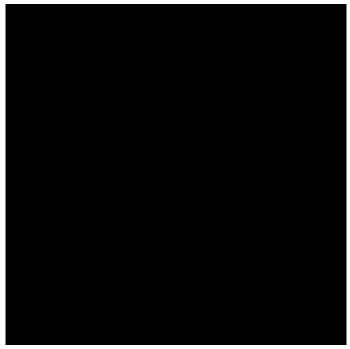


Figure 5. Bearing stress between bolt and plate. [1]

The average nominal value of bearing stress is determined using Equation 3.

$$\sigma_b = \frac{P}{A} = \frac{P}{td} \qquad \text{Eq.3}$$

Where P is the applied force, t is the thickness of the plate and d is the diameter of the bolt. [1]

Front Hitch Bolt Connection

The shearing and bearing stresses that occur at the bolted front hitch connection will be calculated in this section.

The hand calculations will be performed based on the following assumptions and dimensions:

1. The front hitch bolt connection is assumed to be double shear
2. The applied force will be evenly distributed across all three bolt connections.
3. The applied load will be 40,000 lbs in all calculations.
4. The bolt diameter is 1 inch.
5. The bracket plate thickness is 0.5 inches.
6. The hitch will be considered a plate that is 4 inches thick.

The average shear stress at each bolt is calculated using Equation 1.

$$\tau_{ave} = \frac{40{,}000}{2\pi * 0.5^2} = 25{,}464.97\ psi$$

The average shear stress on each bolt is $25{,}464.97\ psi$

The bearing stress must be calculated in two areas. First, between the bolt and the bracket and second, between the bolt and the hitch.

The bearing stress at each bolt between the bolt and the bracket is calculated using Equation 3.

$$\sigma_b = \frac{40{,}000}{(2 * 0.5 * 1)} = 40{,}000\ psi$$

The bearing stress at the connection between the bolt and the bracket is $40{,}000\ psi$.

The bearing stress at each bolt between the bolt and the hitch is calculated using Equation 3.

$$\sigma_b = \frac{40{,}000}{(4 * 1)} = 10{,}000\ psi$$

The bearing stress at the connection between the bolt and the hitch is $10{,}000\ psi$

All stresses that have been calculated in this section are summarized in Table I.

Table I:
FRONT HITCH HAND CALCULATION SUMMARY

Stress Type	Stress Value
Average Shear	$25{,}464.97\ psi$
Bearing Bolt and Bracket	$40{,}000\ psi.$
Bearing Bolt and Hitch	$10{,}000\ psi.$

Load Cell Pin Connections

The shearing and bearing stresses that occur at the load cell connection will be calculated in this section. The hand calculations will be analyzed the stresses at the pins in load cell connection assembly.

The hand calculations for pin connections will be performed based on the following assumptions and dimensions:

1. Pin connection #1 is assumed to be double shear
2. The applied load will be 120,000 lbs in all calculations.
3. All three pin connections in load cell assembly will see the same stresses
4. The pin diameter is 1.5 inches.
5. The load cell support bracket plate thickness is 1 inches.
6. The load cell mount will be considered a plate that is 1.875 inches thick.

The average shear stress at pin connection #1 is calculated using Equation 1.

$$\tau_{ave} = \frac{120{,}000}{2\pi * 0.75^2} = 28294.42\ psi$$

The average shear stress on pin connection #1 is $28294.42\ psi$.

The bearing stress must be calculated in two areas. First, between the pin and the support bracket and second, between the pin and the load cell plate.

The bearing stress between pin connection #1 and the support bracket is calculated using Equation 3.

$$\sigma_b = \frac{120{,}000}{(2 * 1 * 1.5)} = 40{,}0000\ psi$$

The bearing stress at pin connection #1 between the pin and the support bracket is $40{,}0000\ psi$.

The bearing stress at pin connection #1 between the pin and the load cell plate is calculated using Equation 3.

$$\sigma_b = \frac{120{,}000}{(1.5 * 1.875)} = 42{,}666.66\ psi$$

The bearing stress at pin connection #1 between the pin and the load cell plate is $42{,}666.66\ psi$.

A summary of all stresses calculated at pin connection #1 is displayed in Table II.

Table II:
LOAD CELL PIN CONNECTION #1 HAND CALCULATION SUMMARY

Stress Type	Stress Value (At each bolt)
Average Shear	$1.95x10^8 N/m^2$
Bearing Pin and Support Bracket	$4.59x10^8 N/m^2$
Bearing Pin and Load Cell Plate	$2.45x10^8 N/m^2$

Vertical Beam Hand Calculations

The loading scenario that will be used is a cantilever beam with a load applied at the end as shown in Figure 6.

[redacted]

Figure 6. Cantilever beam load scenario. [1]

The equation of elastic curvature of the beam in this load condition is displayed in Equation 4.

$$y(x) = \frac{P}{6EI}(x^3 - 3Lx^2) \qquad \text{Eq.4}$$

Where P is the applied load, E is the modulus of elasticity, I is the moment of inertia and L is the length of the beam. The maximum deflection in the beam will be when x = L. Substituting L for x in Equation 4 yields the maximum deflection shown in Equation 5.

$$y_{max} = \frac{-PL^3}{3EI} \qquad \text{Eq.5}$$

Taking the derivate of Equation 4 yields the equation for the slope of the beam and is shown as Equation 6.

$$\theta(x) = \frac{P}{2EI}(x^2 - 2Lx^2) \qquad \text{Eq.6}$$

To determine the maximum slope of the beam L will be substitute for x. The equation for the slope at the end of the beam is displayed as Equation 7.

$$\theta_{max} = \frac{-PL^2}{2EI} \qquad \text{Eq.7}$$

Taking the derivative of Equation 6 yields the equation for moment in the beam and is shown as Equation 8.

$$M(x) = P(L - x) \qquad \text{Eq.8}$$

The maximum moment is at x=0. Substituting zero for x yields Equation 9.

$$M_{max} = PL \qquad \text{Eq.9}$$

Taking the derivative of Equation 8 yields the equation for the shear stress in the beam and is shown as Equation 10 [1].

$$V(x) = V_{max} = -P \qquad \text{Eq.10}$$

The vertical support beam at the rear of the load cart will experience forces that will be transmitted from the caster wheel connection. The maximum deflection, maximum slope, maximum bending stress and shear stress will be calculated based on the following assumptions and dimensions:

1. The vertical support will be analyzed as a cantilever beam
2. The applied load will be 10,000lbs.
3. The load will be applied at the end of the beam furthest from the fixed support
4. The beam will be a 2"x4" beam that is 3/8' thick
5. The modulus of elasticity of the beam is 29732.7 ksi. [2]

The moment of inertia of the beam must first be calculated based on its rectangular tube geometry using Equation 11.

$$I = \frac{1}{6}h^3t(1 + 3\frac{b}{h}) \qquad \text{Eq.11}$$

Where t is the tube thickness, b is the length of the base and h is the length of the height. Using Equation 11 the moment of inertia is calculated.

$$I = \frac{1}{6} * 4^3 * \frac{3}{8}\left(1 + 3 * \frac{2}{4}\right) = 10\ in^4$$

The maximum deflection in the beam can be calculated using Equation 5.

$$y_{max} = \frac{-10{,}000 * 36^3}{3 * 2.97e8 * 10} = 0.0523\ inches$$

The deflection at the end of the beam is 0.053 inches.

The maximum slope in the beam can be calculated using Equation 7.

$$\theta_{max} = \frac{-10{,}000\ 36^2}{2 * 2.97e8 * 10} = 0.126\ degrees$$

The slope at the end of the beam is 0.126 degrees.

The moment in the beam can be calculated using Equation 9.

$$M_{max} = 10{,}000 * 36 = 3.6x10^4 lb\ ft$$

The moment on the beam is $3.6x10^4 lb\ ft.$

The shear stress in the beam can be calculated using Equation 10.

$$V_{max} = -10{,}000lb$$

The shear stress in the beam is 10,000 lbs.

The stress at the top of the beam is calculated using Equation 11.

$$\sigma_b = \frac{10{,}000}{(3.9375)} = 2539.68\ psi$$

PTFE Bearings

The PTFE bearing support the PowerPin, which is the rear connection of the load cart. The bearings experience vertical, and side normal forces. Normal stress or axial stress occurs when an axial force is subjected to an object. It can be a tensile, or compressive force. The average normal stress, σ, represents the intensity of forces over a given section and can be found by dividing the axial load, P by the area, A.

$$\sigma = \frac{P}{A} \qquad \text{Eq. 11}$$

Normal stress can also be found from the product of the strain in a material and that material's modulus of elasticity, E. The normal strain, ε, in an object is a ratio of deformation, δ, over the total length, L of the object in that direction.

$$\epsilon = \frac{\delta}{L} \qquad \text{Eq. 12}$$

The normal stress can then be found from the following formula, where E is a material constant.

$$\sigma = E\varepsilon \qquad \text{Eq. 13}$$

Finally, combining equations 12 and 13 the deflection in a material in a given loading scenario can also be found. [1]

$$\delta = \frac{PL}{AE} \qquad \text{Eq. 14}$$

The PTFE bearing sheets will experience downwards force from the implement during normal operation, as well as sideways force during turns. The maximum vertical force experienced by the PTFE bearing was estimated from a study previously done by PAMI [3]. This load was found to be 6700 lbs max for category five implements.

The following assumptions will be made for the calculation of the vertical bearing stress in the PTFE sheet.

1. Max load with safety factor of 2 will be 13400 lbs
2. Loads will be distributed across the bearing plate evenly
3. The area of the PTFE bearing plate is 100.9 in^2
4. The PTFE is 0.5" thick
5. The yield strength of the PTFE is 3,700 psi
6. The modulus of elasticity of the PTFE is 80,000 psi [4].

Using equation 11 the maximum stress experience by the PTFE can be found

$$\sigma = \frac{13400\ lbs}{100.9 in^2} = 132.8\ psi$$

This stress is significantly lower than the yield stress of 3,700 psi, therefor this bearing plate will not fail. However, removing assumption 2, the area of the bearing plate required to support the 13,400 lbs load can also be calculated by rearranging equation 11.

$$A = \frac{13400\ lbs}{3700\ psi} = 3.62\ in^2$$

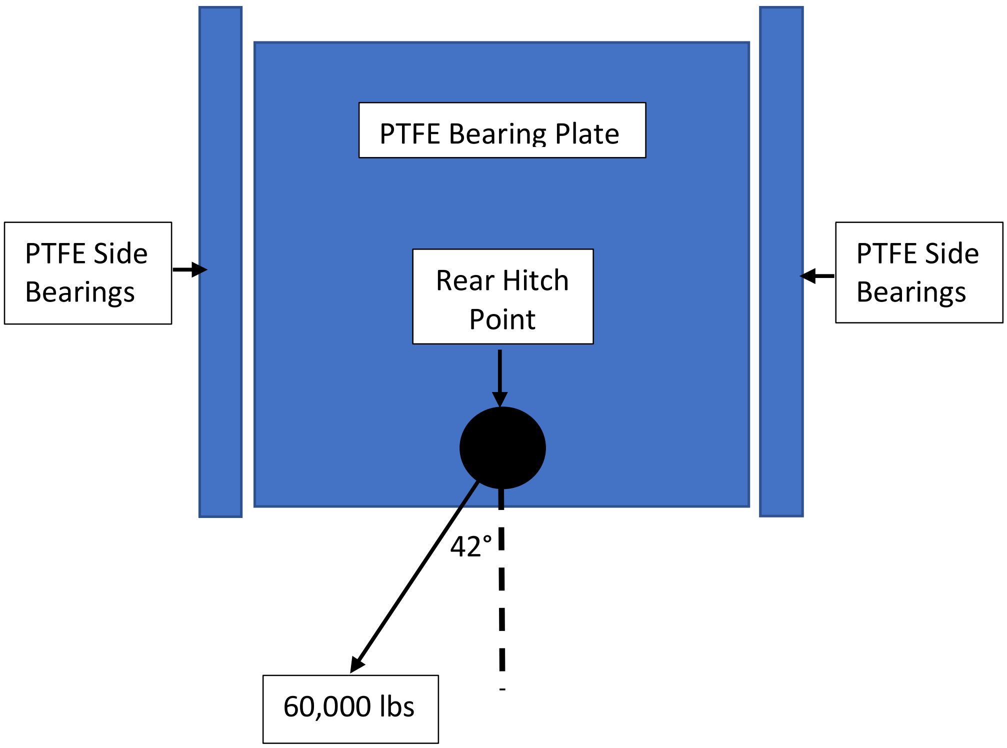
This illustrates that if the load is not distributed over the entire 127.6 in^2 PTFE plate, only 3.62 in^2 of the plate is required to support the vertical load.

The following assumptions will be made for the sideways loading of the side PTFE bearings.

1. The maximum load applied to the rear hitch of the load cart with a safety factor of 2 will be 50,000 lbs.
2. The maximum articulation angle of the towing machine will be 42° [5].
3. The maximum compressive strength of the PTFE is 3,700 psi.
4. The modulus of elasticity of the PTFE is 80,000 psi.
5. Loads will be distributed across the PTFE bearing evenly.
6. The PTFE is 0.5" thick

7. The area of the side PTFE bearing is 10 in^2

Assumption 1 is made due to the fact that the full load experienced by the towing machine will not be present while it is completing the sharpest turn it can make. The 42° angle will also only be experienced when an articulated towing machine is used and will be the worst-case scenario. The general layout of the rear hitch and PTFE bearing is shown in



Using equation 11 again the maximum stress experienced by the PTFE side support is.

$$\sigma = \frac{50000 \sin(42)\ lbs}{10 in^2} = 3346\ psi$$

This stress is lower than the yield stress of 3,700 psi, therefor this bearing plate will not fail. However, removing assumption 2, the area of the bearing plate required to support the 60,000 lbs load can also be calculated by rearranging equation 11.

$$A = \frac{50{,}000 \sin(42)\ lbs}{3700\ psi} = 9.0\ in^2$$

At the absolute worse case loading scenario the area required to support the maximum load is 11.5 in^2.

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