Honey Tote Hydraulic Tilter

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MEC E 460			Capstone Design Project			Fall 2019

Project Background

Apiculturists in Alberta are converting from conventional steel drums to IBC tote containers for honey storage and transport. Totes offer several advantages to beekeepers: larger in capacity, more durable, and less expensive over their lifespan. Although the totes have a higher purchase cost, their estimated lifespan is over six times that of barrels.

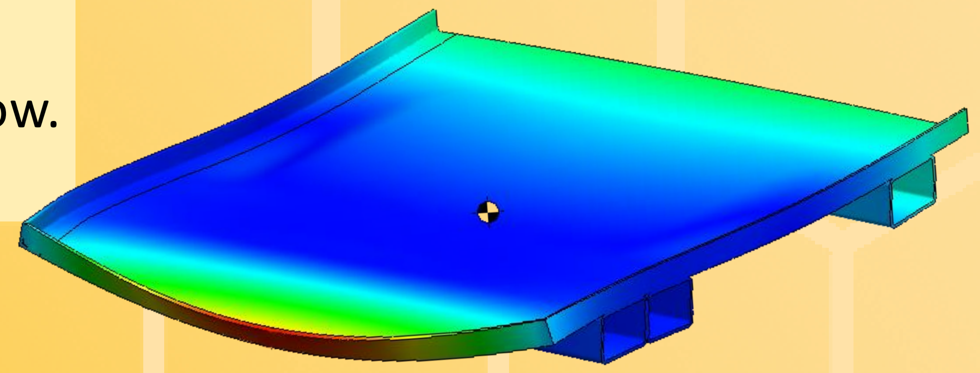
Design Objectives

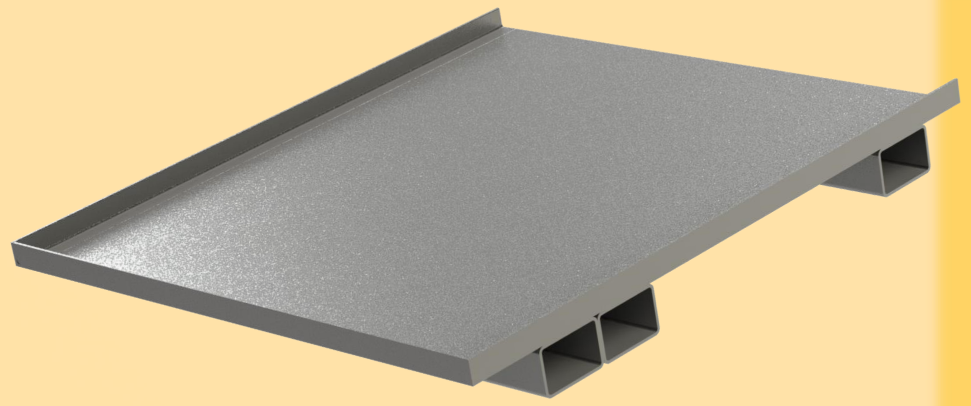
1. Recommend a suitable tote
2. Design a machine to aid in the draining and cleaning of a tote loaded with honey

Finite Element Analysis

Finite element analysis was conducted on the shaft, shaft mount, base plate, and top plate. For each component, stress, strain, displacement, and factor of safety were analyzed.

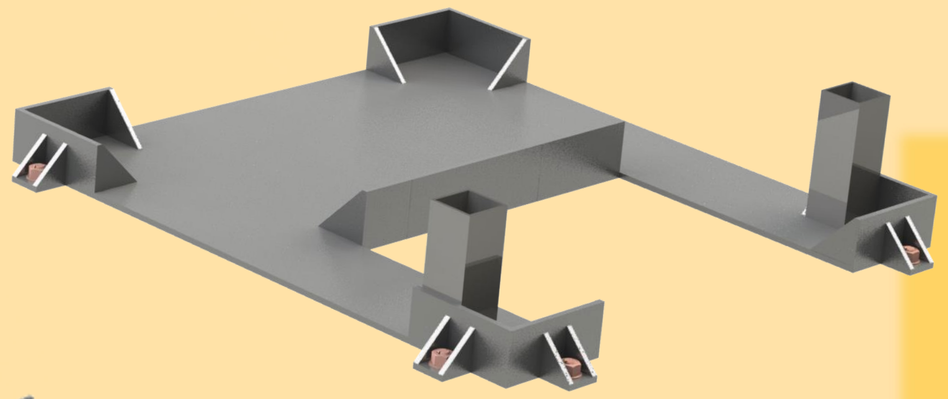
A displacement plot of the top plate is shown below. The factor of safety for the top plate is 2.6.





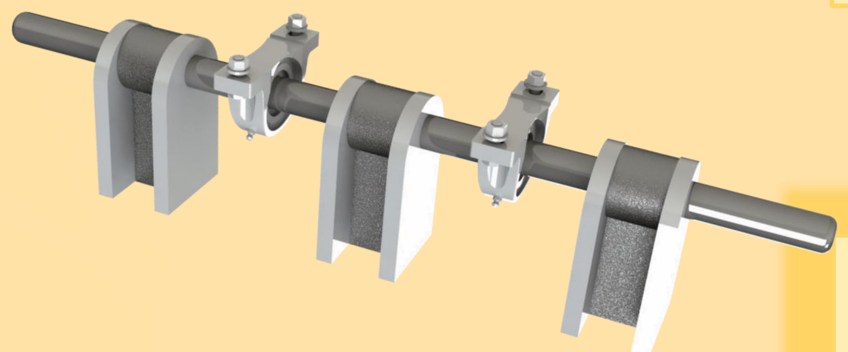
Top Plate Subassembly

- Rectangular tubes provide mounting points for the hinge assembly and hydraulic actuator, and provide additional structural rigidity
- Spout can be attached to front lip



Bottom Plate Subassembly

- Provides mounting points for hydraulic cylinders and hinge assembly
- Secured to floor with anchor bolts at all corners



Hinge Subassembly

- Connects the top plate and bottom plate and allows for rotation
- Top plate is bolted to ball bearings, which rotate freely on shaft
- Shaft mounts are welded to bottom plate and fixed to shaft by set screws

Technical Specifications

Cost	$7520
Maximum tilt	25 deg
Load Capacity	3500 lbm (1631 kg)
Dry Mass	4687 lbm (2126 kg)
Full tilt dimensions	70 x 56 x 38 in

UNIVERSITY OF ALBERTA

Department of Mechanical Engineering

Honey Tote Machine

Phase II Design Report

MEC E 460 Fall 2019

Group 2

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Executive Summary

Apis Consulting has been contracted to select a tote container for storing honey and design a machine capable of aiding in the cleaning or draining process.
The primary design constraints are the project budget of $5000 and the loading cases of 3500 lb for a full tote to be drained, and 135 lb for an empty tote to be washed.

The food grade IBC Tote selected by Apis is a 275 gallon, 135 lb container with an associated purchasing cost of $354.85 CAD [12].

Apis has developed three design concepts: the Hydraulic Tilter, the Rotating Frame, and the Forklift Attachment. The Hydraulic Tilter concept has the capacity to support both load cases, but does not invert the tote for cleaning. The Rotating Frame is estimated to be the most expensive to manufacture, but is capable of inverting the tote in the cleaning case. The Forklift Attachment incorporates existing equipment in its application but also has increased inherent safety concerns. The three concepts were evaluated using a design evaluation matrix, and the Hydraulic Tilter was determined to be the best option to address both use cases. The decision was approved by the client.

In Phase III, Apis Consulting aims to further optimize the Hydraulic Tilter by increasing manufacturability of the hinge assembly, designing a guard/restraint system, and reducing cost via material selection. In order to perform these tasks, the original engineering cost estimate of 550 hours has been revised to 664 hours (an equivalent cost of $60,360).

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Introduction

The standard method for the storage and transportation of honey in the Alberta apiculture industry is via the use of 55 gallon steel barrels. The price associated with these barrels has been consistently increasing, incurring additional costs to Alberta Beekeepers without any new benefit of continued use. To alleviate this issue, the Alberta Beekeepers Commission has decided to investigate alternative storage solutions.

Apis Consulting has been contracted to provide an innovative and cost effective solution to the problem via the use of a Food Grade IBC tote, and a mechanical device to aid in the associated draining, and cleaning of the container. To facilitate this, the client requested that the device be capable of supporting the weight of a full tote of 3500 lb (16 kN) for draining considerations, and be able to manipulate an empty tote of 135 lb (601 N) to aid in the cleaning process. Emphasis has been placed on simplicity of use and affordability of the device; enabling its widespread use in the Albertan Beekeeping industry.

Design Specification Revisions

Nearing the end of Phase 2, correspondence with the client revealed a change in the use case of the device, which correspondingly lead to changes in the overall scope of the project. These changes are reflected in the Design Specification Matrix collected in Appendix D.

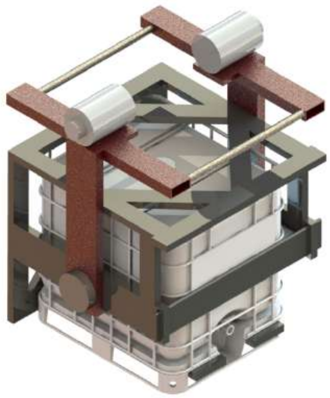
1. The client identified two ideal use cases, draining and cleaning, with two different loading conditions of 3500 lb and 135 lb, respectively.
2. A method to adjust height in the event of power loss is no longer applicable with the concepts developed.
3. Electrical efficiency is no longer a consideration as none of the concepts require significantly complex electrical components.

4. Client concerns that the machine be used to aid draining or cleaning the tote container. Additional design criteria added to reflect the different use cases.
5. With the integration of other equipment such as forklifts, the degree of training required for machine use is now considered.
6. To ensure budgetary goals are met, manufacturing cost with respect to total component and number of custom components is now considered.

Concept Generation

Following the brainstorming process conducted by the team, three unique concepts were developed. Each concept addresses the problem of manipulating the tote in a different manner; with focus being placed on the ease of access, affordability, and simplicity associated with the design. A brief summarization of the three concepts is collected in Table 1. For further elaboration on the generation of concepts refer to Appendix A.

Table 1: Design Concept Summary

		
Concept 1: Hydraulic Tilter	**Concept 2: Rotating Frame**	**Concept 3: Forklift Attachment**
● Simple hydraulic system ● Suitable for cleaning and draining ● Lowest manufacturing cost estimate	● Self-contained system ● 360° access to tote ● Simple operation	● Integration with existing equipment ● Mobility ● Full rotational capabilities

Key Analyses Performed

The general design consideration of all three concepts revolves around their ability to aid in the process of draining or cleaning, as per the client's request. For cleaning considerations, the client prefers the ability to fully invert the tote for easy access through the top hole. As such, concepts focus on facilitating the clients use case for cleaning. This involves the rotation of the empty 135 lb (601 N) tote to allow access via the top hole. For the client’s draining use case, the device needs to be capable of supporting the weight of a full tote of 3500 lb (16 kN) and direct the fluid to the exit spout located at the bottom of the tote.

Each design concept is subject to the general process of determining the forces throughout, and evaluating some of the associated stress on critical components. This general process is shown in Figure 1. However, with the substantial variation in how each of the concepts address the method of tote movement, all the devices are subject to unique design considerations. These calculations are collected in Appendix B. Key identifying features of each concept are collected in Table 2.

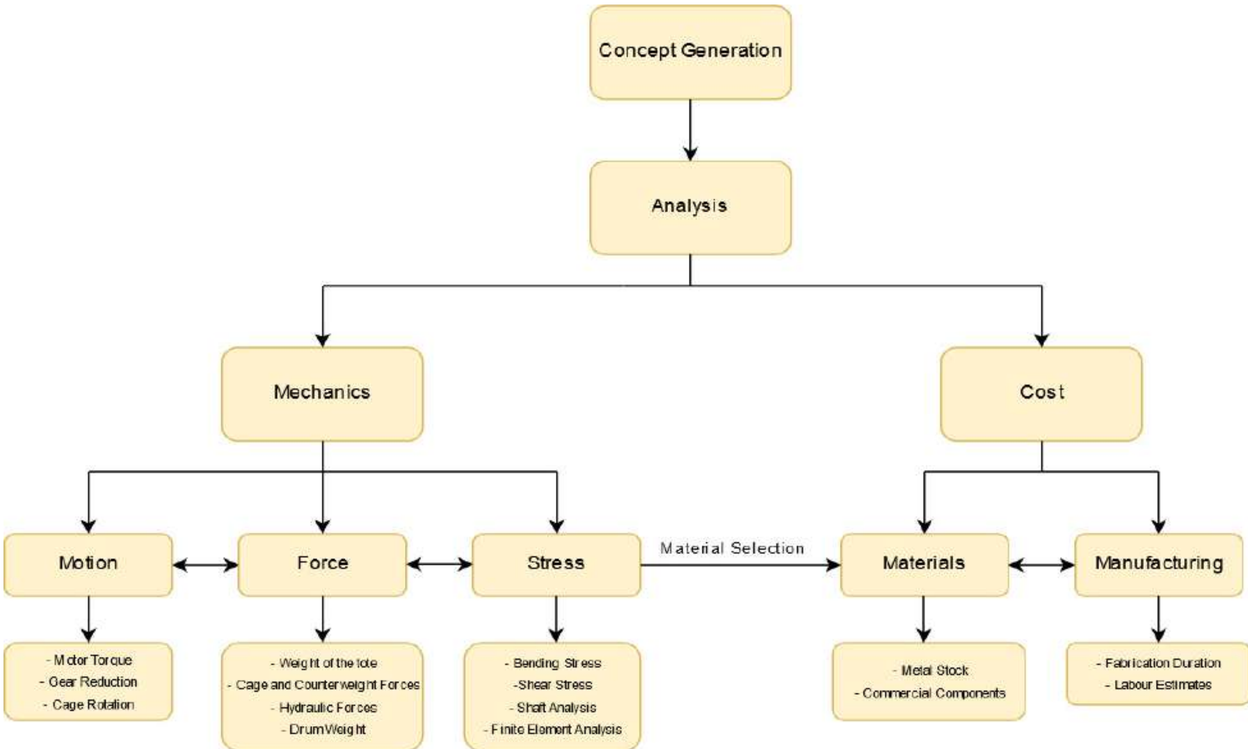


Figure 1: Concept Analysis Process Flow Chart

Table 2: Summary of key parameters

Parameter	Hydraulic Tilter	Rotating Frame	Forklift Attachment	Reference
Rated Capacity (lb)	5250	405	405	Appendix B
Angular Displacement (deg)	35	360	360	Appendix B
Labour Cost (CAD)	900	2700	3500	Appendix C
Maximum Dimensions L x W x H (in)	91 x 50 x 83	58 x 66 x 86	65 x 56 x 70	Concept Descriptions

Concept Descriptions

Concept 1: Hydraulic Tilter

The Hydraulic Tilter design makes use of a simple hydraulic based tilt system akin to a "teeter - totter". Figure 2 collects the key components of the design with associated component descriptions collected in Table 3.

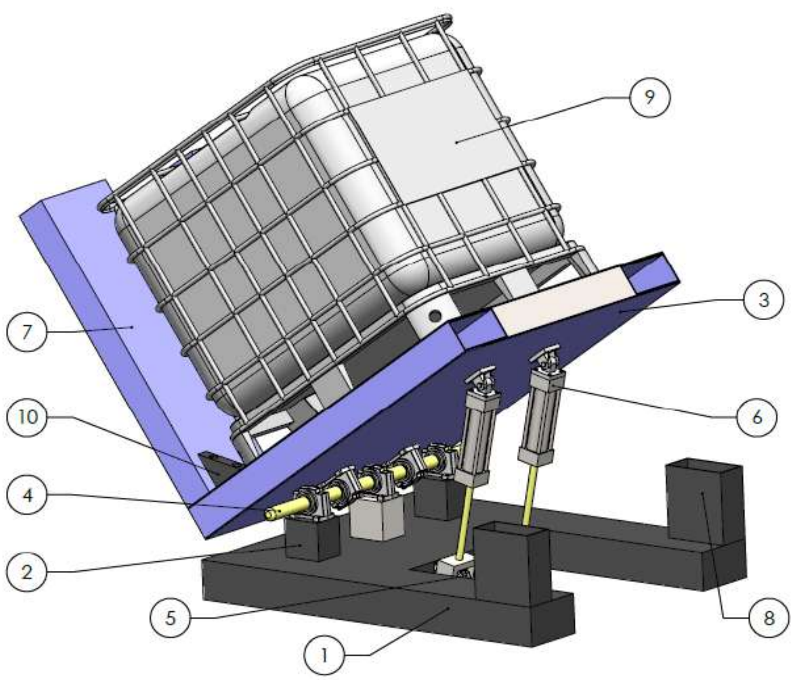


Figure 2: Hydraulic Tilter features

Table 3: Hydraulic Tilter Assembly Parts

Label	Component	Feature
1	Bottom Plate	Base of the machine, mounted to the ground
2	Bottom Bearing Shaft Mount	Connects bottom plate to shaft plate
3	Top Plate	Solid surface used for tote positioning
4	Hinge Shaft	Allows top plate to be tilted
5	Hydraulic Base Mount	Mounts hydraulic to the bottom plate
6	Hydraulic Cylinder	Used to tilt the top plate
7	Front Support	Maintain tote position once device is in operation
8	Rear Supports	Supports top plate and tote at stationary position
9	Tote	Container to be tilted
10	Front Support Bracket	Provide additional structural support for for front supports

The key advantage of this concept is its affordability and its ability to facilitate drainage of the tote. At the Hydraulic Tilter's stationary position, the tote is placed onto the top plate via the use of a forklift, as shown in Figure 3.

Figure 3: Hydraulic Tilter in Stationary Position

The operator will then make use of the hydraulic cylinder system to extend the cylinders and allow the top plate to tilt about the Hinge shaft. The machine will be capable of inclining to a maximum tilt angle of $\theta = 35°$ to aid in the draining of honey from a full tote, or water from an empty tote during the process of cleaning; this is shown in Figure 4.

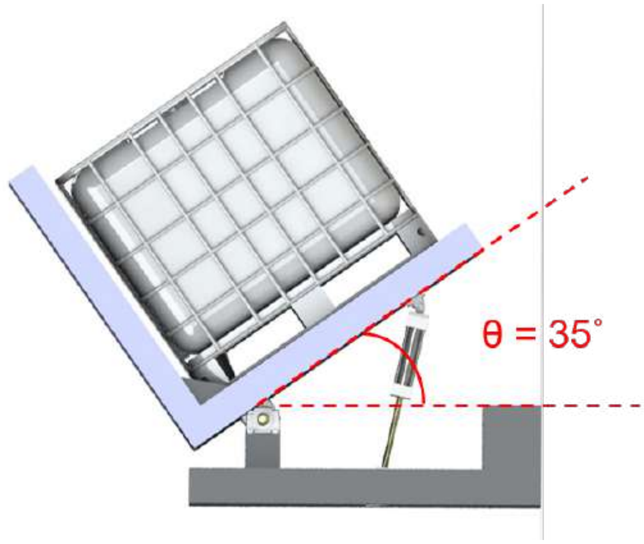


Figure 4: Hydraulic Tilter at maximum operating position

The maximum dimensions in the stationary and maximum angular position of the Hydraulic Tilter are shown in Figures 5 and 6, respectively.

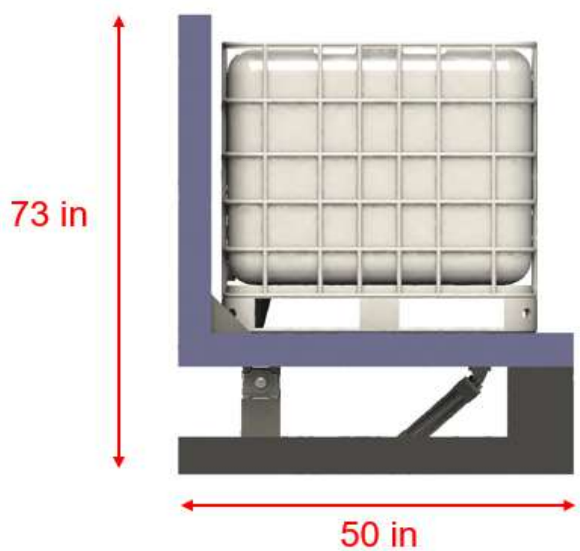


Figure 5: Dimensions of the Hydraulic Tilter at the stationary position

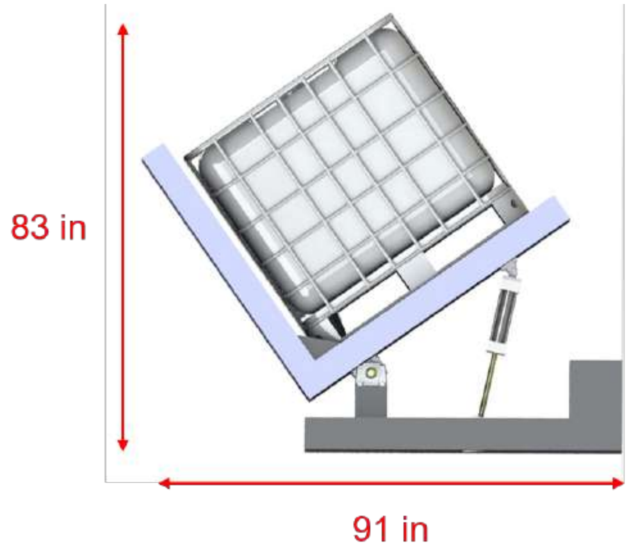


Figure 6: Dimensions of the Hydraulic Tilter at maximum operating angle

The overall geometry for this concept is simple, allowing for further changes to be made without major difficulty, such as the inclusion of an additional container restraint system.

Unlike the Rotating Frame and Forklift Attachment concepts, this machine is incapable of fully inverting the tote for cleaning, resulting in reduced accessibility via the top hole of the tote container. This can be remedied via the use of a third party specialized cleaning device, such as the Alfa Laval Gamajet Cleaner [1] or a pressure washer wand attachment similar to that of Hydro-Chem Systems' [2]. This concept, akin to the Rotating Frame, is a stationary device that would be mounted to the concrete floor of a given beekeepers processing facility.

As previously stated, this concept is the most affordable of the three designs. This is due to the simplicity of the mechanism to facilitate the draining of the tote, and the associated reduced difficulty in manufacturing. Stock materials such as 5 in x 10 in rectangular tubing and off the shelf components such as Hydraulic Cylinders [5], limit the cost associated with custom designed components. A conservative cost estimate for the manufacturing of this device was provided by MEC E shop technicians as $900; with an estimated raw materials cost of $1509. The predicted total cost associated with the Tilter of $2409 falls well within the client specified cost of $5000. Further details on cost estimation is found in Appendix C.

Concept 2: Rotating Frame

The Rotating Frame concept is designed for inverting a near-empty tote for cleaning. Like the Hydraulic Tilter concept, the Rotating Frame system is a stationary, self-contained machine which must be loaded using a forklift. The tote is securely held by a steel subframe which incorporates a large hoop. The subframe is supported in the main frame by the driveshaft at the rear, and by two guide rollers which support the hoop. The driveshaft at the rear also carries a chain sprocket, which transmits power from the motor mounted on the baseplate. Figures 7-8 display the full design dimensions for this concept. Figure 9 shows the concept during rotation with the tote inside. Figure 10 and Table 4 show the features of the Rotating Frame design.

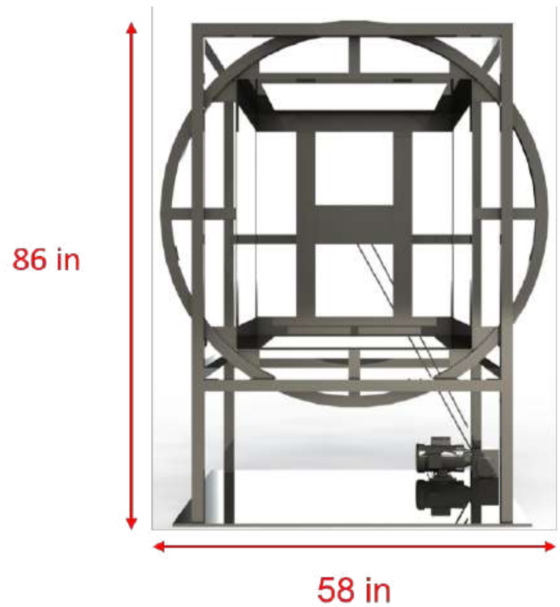


Figure 7: Overall width and height of Rotating Frame concept

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Figure 8: Maximum length of Rotating Frame concept

Figure 9: Rotating Frame concept with tote

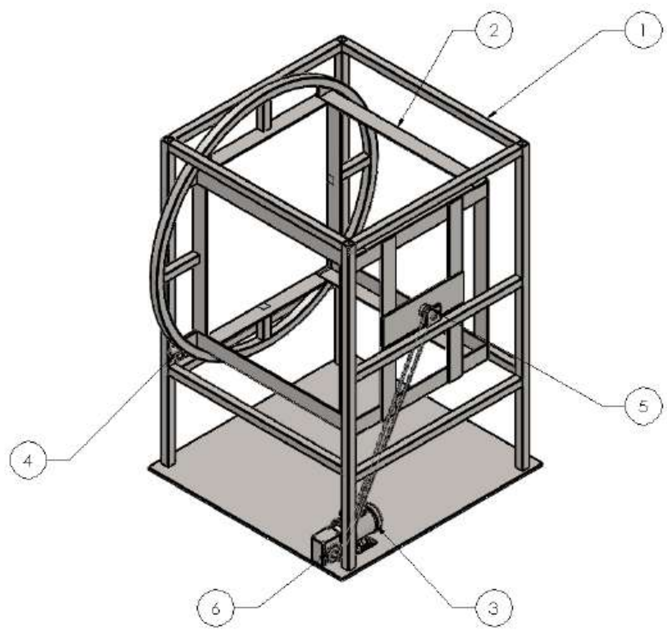


Figure 10: Rotating Frame parts on isometric view

Table 4: Rotating Frame Parts

Label	Component	Feature
1	Main Frame	Primary structural support
2	Subframe	Secures tote during rotation
3	Motor & Reduction Gear	Rotates the subframe via chain drive
4	Guide roller	Supports subframe ring
5	Ball Bearing	Houses the Tote
6	Chain Sprocket	Transmits power via chain
7	Tote	Contains the honey residue

A particular advantage of this concept is the 360° access to tote, allowing for a multitude of cleaning positions. In the event of catastrophic failure, the design of the structure ensures that the drum and load will not drop suddenly, reducing risk associated with operation.

Concept 3: Forklift Attachment

The Forklift Attachment is designed for maximum mobility of the tote in terms of rotational freedom. As the name suggests, the Forklift Attachment is meant to be used with a forklift. The Forklift attachment is designed for cleaning purposes only, as it allows for a full inversion of an empty tote. The limitation for this concept is that it is not designed for draining or inversion of a tote full of honey. This is due to the large torque requirements of 12.7 kNm.

Altogether, the cost for materials and manufacturing is approximately $3205. See Appendix C for a detailed analysis of the cost estimation.

An isometric view of the concept can be seen in Figure 11. An exploded view of the Forklift attachment is in Figure 12, and the descriptions of the bubbled components are in in Table 5.

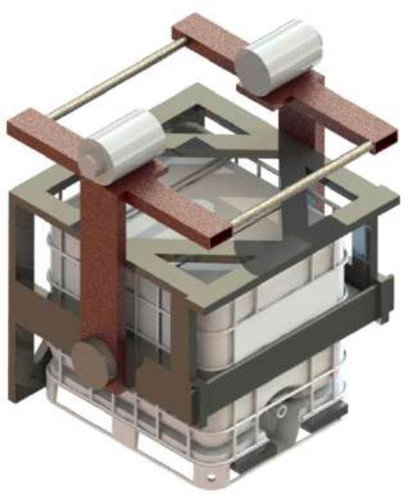


Figure 11: Forklift Attachment concept isometric view

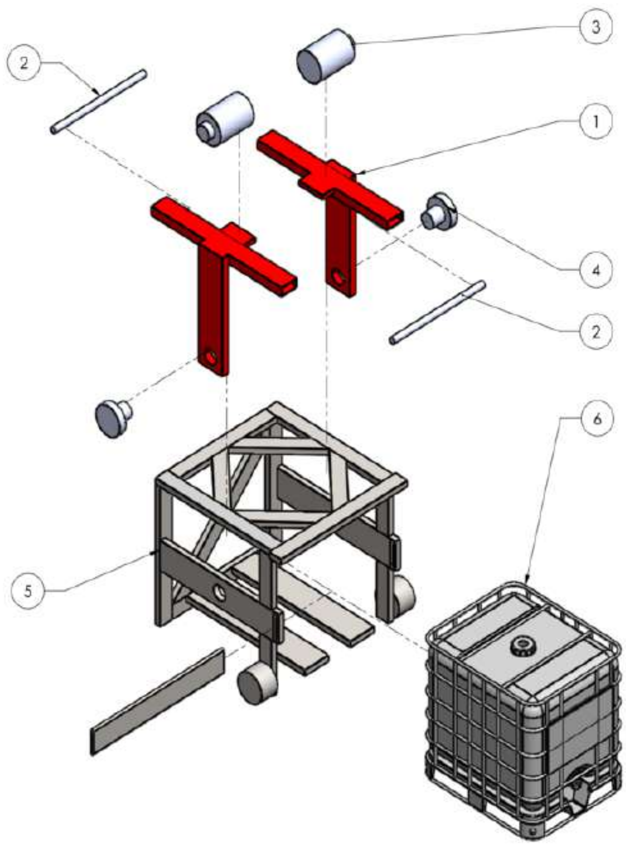


Figure 12: Forklift Attachment features

Table 5: Forklift Attachment Features

Label	Component	Features
1	Vertical Arm and Sleeves	Main mount for the motor, top shafts, and rotating shaft. Forklift's fork insert into the sleeves
2	Top Shaft	Connects the vertical arms together
3	Motor	Rotates the rotating shaft via belt
4	Rotating Shaft	Rotates the Cage
5	Cage	Houses the Tote
6	Tote	Contains the honey residue

Vertical Arm and Sleeves

The motors, chain drive, top shafts, and rotating shafts are mounted to the vertical arms of the assembly. The vertical arms are the most crucial component of the assembly as they carry the majority of weight. The top portion of the vertical arms are the sleeves. The sleeves allow the forklift to connect to the Forklift Attachment. Additionally, there are top shafts to hold the left and right vertical arms together.

Description of Motion

The Forklift attachment has sleeves which allow a forklift to connect to the concept. Once connected, the forklift itself governs the vertical motion. The forklift raises the Forklift Attachment and the tote. While raised, the Forklift Attachment rotates the tote using the chain drive. A detailed chain drive design was planned for Phase III, and it is not shown in the figures.

The chain drive connects the motors to the rotating shaft. The shaft is locked with the cage to allow for the transfer of torque. Figure 13 shows the rotational motion.

Figure 13: Partial rotation of the tote in the forklift attachment

Cage and Counterweight

The cage is designed around the dimensions of the tote to allow for minimal tolerance and a secure fit. The overall dimensions of the assembly are shown in Figure 14. The cage has two forks at the bottom that act like the forks of a forklift and attach to the bottom of the tote. A front cross-bar is placed across the cage to secure the tote and prevent it from tipping out of the cage. The cage is designed using similar beams of 2”x5” rectangular pieces of various lengths. The beams use the same cross section. This allows for ease of manufacturing as only cuts for length will be made. Therefore, no CNC specialized parts will be required. By simply welding the pieces together, the construction of the cage can easily be done with low incurred cost.

There are counterweights strategically attached to the bottom of the cage. The counterweights nullify cage imbalances by intentionally positioning the combined center of gravity of the cage and counterweight in the center of rotation.

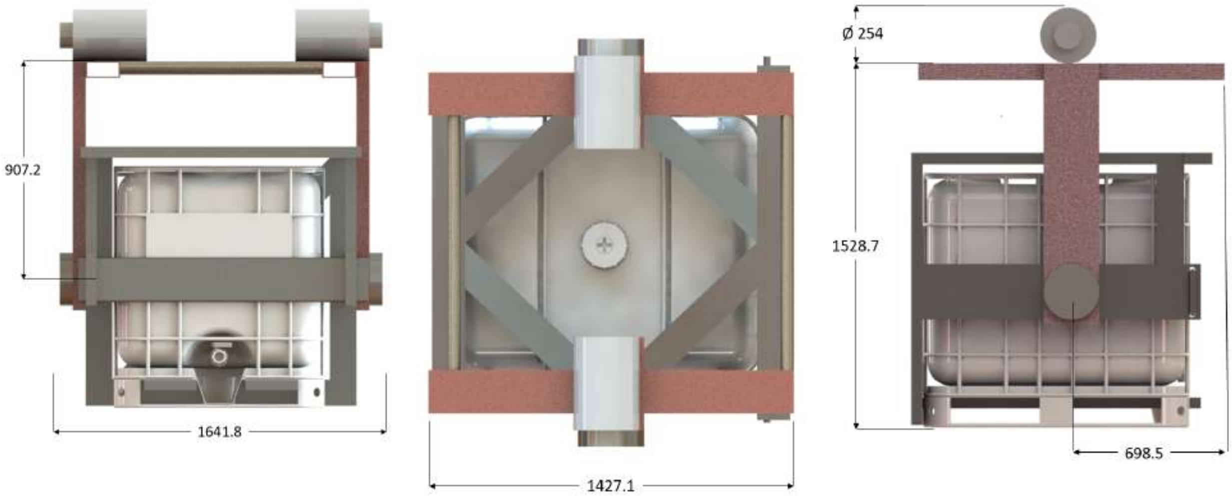


Figure 14: Overall dimensions of the Forklift Attachment assembly. From left to Right: Front view, Top view, Side view.

Design Decision Matrix

Table 6 collects the criteria used to evaluate the three design concepts. The criteria are scored in importance from 1 to 5, with 1 being the least possible score and 5 being the highest. This indicates their ability to address the given criteria.

Table 6: Design Decision Matrix

Item	Description	Specification	Importance (1-5)	Hydraulic Tilter (HT)	Drum (DR)	Forklift Attachment (FA)	Justification
1	**Dimensions**						
1.5	Overall Dimensions	At minimum the footprint of the machine will have the dimensions of the tote (~40"x45")	3	4	3	5	FA is has the smallest footprint of the three concepts
2	**Safety Requirements**						
2.1	Method to Fix Position	Mechanism to fix position of load during lifting to prevent unexpected reduction in height	4	3	5	5	HT Hydraulics would gradually descend back to staitionary postion, not maintain vertical postion DR does not adjust the heigh of the Tote FA relies on the inherent safety mechanism of the Forklift for vertical translation
2.6	Fall Protection	Operator should not be under suspended load while machine is in operation	4	5	5	3	HT and DR do not allow operator to get under the load FA has potential for operator to go under the machine
2.8	Pinch Points	No exposed pinch points	3	4	3	2	HT has least obvious exposed pinchpoints
2.11	Cleaning Load Capacity	Capable of supporting ~ 405 lb load, with a deformation of less than 1% of a given dimension (3 x Safety Factor)	5	5	5	5	HT, DR, and FA all are capable of supporting the empty tote
2.12	Draining Load Capacity	Capable of supporting ~ 5250 lb load, with a deformation of less than 1% of a given	5	5	5	1	HT and DR are capable of facilitating draining with a full tote FA is unable to rotate with a full tote

		dimension (1.5 X Safety Factor)					
4	**Cost**						
4.1	Manufacturing Cost	Cost of manufacturing the machine should be less than $5000. Machine should be designed to be affordable.	5	5	2	4	HT is the cheapest DR is the most expensive
5	**Ergonomics**						
5.4	Accessibility	Clear and intuitive machine operating process	4	5	4	4	HT, DR, and FA all have fairly straighforward operating controls HT's "Up and Down" are the simplest
6	**Maintenance**						
6.1	Part Maintenance	Machine mechanisms should be easily accesible for maintenance	2	4	3	5	Harder to reach components of DR and HT
6.2	Repairs	Standard parts used wherever possible to simplify repairs	3	5	2	4	HT has least custom components
6.4	Total Component Count	Machine should have a simple design with fewest components necessary to reduce potential for part failure	3	5	2	3	HT has lowest component count
8	**Operation**						
8.1	Tote Crush Prevention	Machine should securely hold tote in any position without	5	3	5	5	DR and FA have a caged that encloses the tote HT doe snot

		damaging the tote's structure					
8.2	2-Degrees of Freedom Motion	Machine should be able to lift and tilt the tote	4	5	1	2	DR and FA lack the ability to tilt the tote for draining
8.3	4-Degrees of Freedom Motion	Capable of planar and vertical translational motion, and rotational motion about one axis	4	1	3	5	DR and FA can invert the tote HT is incapable of inversion
8.5	Machine Mobility	Capability of machine to be mobile	1	1	1	5	HT and DR are stationary machines
8.9	Operational Precision Requirement s	Ease of operator to insert the tote into the machine	4	4	2	1	Cages of DR and FA increase precision needed for tote placement
8.10	Additional Training Required	Minimizes extra training required to operate the device, or its assoicated equipment (i.e. forklift)	3	3	3	2	FA requires an individual certified to operate a forklift
8.11	Accessibility for Cleaning Tote	Easier access to tote for cleaning facilitated by machine	3	3	5	5	DR and FA provide easy access to the top hole of the tote for cleaning
8.12	Assist in Draining	Ability to tilt the tote to aid in draining of contents	3	5	1	1	HT is able to direct tote contents to the bottom spout of the container
				278	229	236	Overall Design Rating

Recommended Design Concept

From the results of the Decision Matrix it was found that the Hydraulic Tilter provides the best overall solution for the client's draining, and to a lesser extent, cleaning. The Hydraulic Tilter meets the design objective for draining a container full of honey. As suggested in the concept description section, there is limited access to the top hatch for cleaning. This could be mitigated via the use of an additional third-party tool. The third-party tool attaches to the top hatch and automatically cleans the inside of the tote.

Although the Forklift Attachment does directly address the design objective of inverting the tote for cleaning access via the top hole, it lacks the ability to tilt the tote forward to drain via the front spout. Additionally, it was found that the motor torque necessary to rotate a full tote of honey to an inverted position was unfeasible given the concepts core nature as being a mobile forklift attachment.

The Rotating Frame concept is capable of meeting the clients cleaning use case, but incurs additional cost due to the complexity of its manufacturing. Similarly, if the Rotating Frame were to be used to rotate a tote full of honey, the costs associated with increasing it structural rigidity would not be justifiable and would likely exceed the client's $5000 budget.

The final design of the Hydraulic Tilter is estimated to fall well within the assigned design budget, involves the fewest number of necessary components, and is capable of addressing both the clients use cases to some extent.

Phase III Considerations

Although the Hydraulic Tilter is a relatively simple concept, there are a variety of optimizations that could result in reduced costs and increased structural stability. The following list shows potential features and concepts to be considered moving into the third phase of this project.

- Potentially revising hinge shaft assembly for ease of manufacture and assembly; would require additional analysis for shape and size of mount adjustment
- Using a roller assembly for the top plate from ease of placement and removal of the tote; roller and bearing analysis would need to be conducted
- Combined side guard and front bracket support system. Potentially using a welded-on metal component or a chain.
- Further consideration of floor mounting requirements (bolted connections), or use of casters for mobility
- Evaluation of severe worst case of double catastrophic failure of hydraulic cylinders and impact upon rear supports
- Hydraulic shaft and pin analysis

In Phase III, additional analysis will be undertaken to further evaluate the forces and stresses acting upon the critical components of the design. This will be conducted via the application of FEA to avoid oversimplifying load cases for hand calculations and derivations.

Project Management

Figures 15 and 16 collect individual team member hourly contributions for Phase I and II.

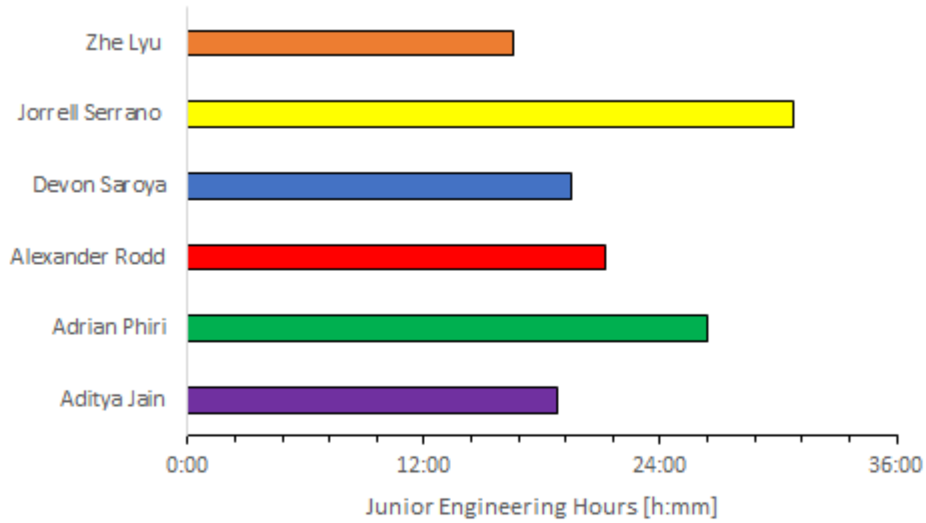


Figure 15: Phase I junior engineering hours per member

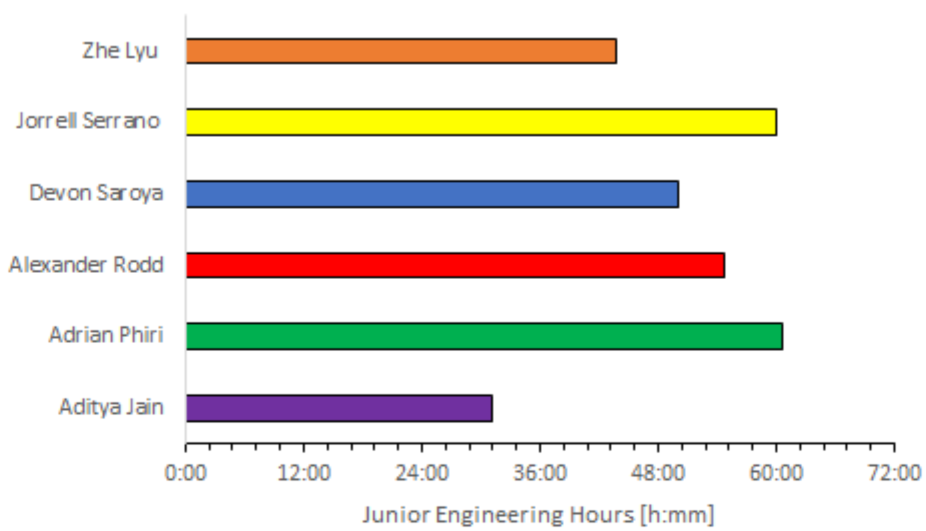


Figure 16: Phase II junior engineering hours per member

The amount of time to be spent on Phase II work was significantly underestimated compared to hours worked. The total discrepancy in worked hours amounted to 127 additional hours worked. The primary reason for this is the inherent complexity associated with two of the designs. The Forklift Attachment specifically had a great deal of analytical requirements than initially expected. Additionally, a change in the initial scope of the project resulted in necessary redesign of concept components that was not initially predicted. Table 7 shows the comparison of project hours as estimated from the onset of Phase I against the actual project hours logged until the end of Phase II. A comparison of the hours worked to those estimated for each phase, and their associated engineering costs is presented in Figures 17 and 18.

Table 7: Total Project Engineering Hours

Item	Proposed Hours		Actual Hours	
	Junior Engineer	Senior Engineer	Junior Engineer	Senior Engineer
Phase I Hours	129	2	133	2
Phase II Hours	173	5	300	3
Phase III Hours	200	6	N/A	
Presentation Hours	34	1		
Total Hours	536	14	433	5

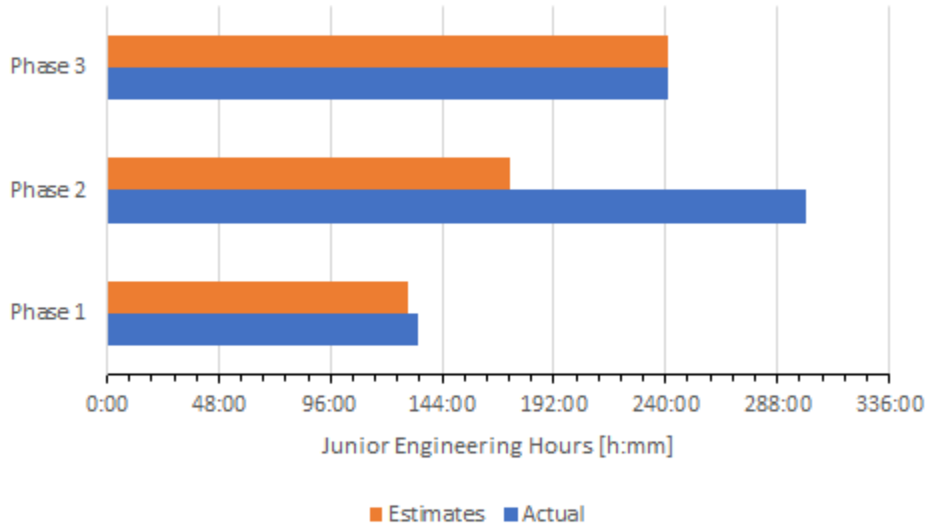


Figure 17: Estimated vs. actual engineering hours per phase

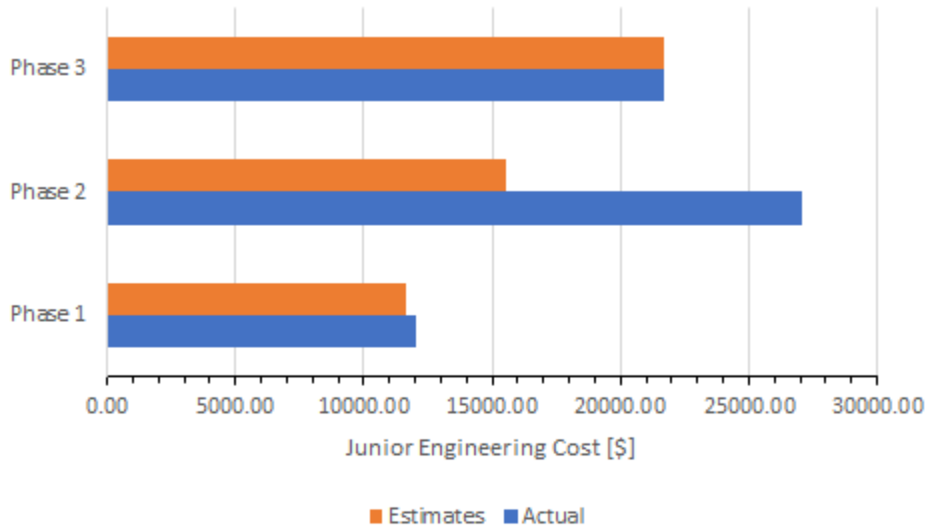


Figure 18: Estimated vs. actual engineering cost per phase

The Gantt Chart, Timesheets, and Meeting Minutes associated with the project are collected in Appendix E.

Conclusion

Three concepts were developed to meet design specifications determined in Phase I of the project. Each concept was evaluated using a decision matrix to determine the one that provided the best overall solution to the client's use case(s). The Hydraulic Tilter was determined to be the most appropriate design to meet the necessary requirements. Apis Consulting is excited to continue developing this design in Phase III.

References

[1] Holland Applied Technologies, [location unknown]. Gamajet Tote Cleaning. (Nov. 30, 2012). Accessed: Sept. 6, 2019. [Online Video]. Available: https://www.youtube.com/watch?v=5hlCnWgV7Vg

[2] Dan Swede, [location unknown]. Tote and Drum Cleaning, Chemical tote cleaning made easy. (Jan. 4, 2014). Accessed: Sept 6, 2019. [Online Video]. Available: https://www.youtube.com/watch?v=eqPTrnDh6AM

[3] R.C. Hibbeler, Mechanics of Materials, 9th ed. New Jersey: Prentice Hall, 2014, pp. Front Matter

[4] McMaster-Carr, *Online Catalog*, McMaster-Carr, Elmhurst IL, 2019, Accessed: Oct. 20, 2019. [Online]. Available: https://www.mcmaster.com/

[5] Princess Auto, *1-1/2 in. Bore x 10 in. Stroke Threaded Head Implement Cylinder*, Princess Auto, Winnipeg: Princess Auto, 2019, Accessed: Oct. 18, 2019. [Online]. Available: https://www.princessauto.com/en/detail/A-p8497638e

[6] McMaster-Carr, *Online Motor Catalog*, McMaster-Carr, Elmhurst IL, 2019, Accessed: Oct. 28, 2019. [Online]. Available: https://www.mcmaster.com/

[7] “Royal Hydraulics - Piston Pumps,” Royal Hydraulics. Accessed: 04-Nov-2019. [Online]. Available: https://www.royalhydraulics.com/index.php/pumps?id=131.

[8] Metal Supermarkets, *Online Catalog*, Metal Supermarkets, Mississauga, ON, 2019, Accessed: Nov. 1, 2019. [Online]. Available: https://www.metalsupermarkets.com

[9] Alibaba, *Online Catalog*, Alibaba, Hangzhou CHN, 2019, Accessed Nov. 1, 2019. [Online]. Available: https://www.alibaba.com/product-detail/NEMA-56C-Motor-MY-singlephase_62164780998.html

[10] "Metal Fabrication Price Calculator: Metal Work: Metal Fabrication." Metal Fabrication Price Calculator | Metal Work | Metal Fabrication. Accessed: November 3, 2019. https://www.csmfg.com/calculator.html.

[11] "Db Electrical LTM0001 Winch Motor for Dump Truck *Tarp Systems Gearmotor*, Motor, Gearbox Hd, 3 and 4 Bolt Heavy Duty, Winches - Amazon Canada." Db Electrical LTM0001 Winch Motor for Dump Truck Tarp Systems Gearmotor, Motor, Gearbox Hd, 3 and 4 Bolt Heavy Duty, Winches - Amazon Canada. Accessed November 3, 2019. https://www.amazon.ca/Electrical-LTM0001-Systems-Gearmotor-Gearbox/dp/B0081S95NQ/

[12] IBC Tanks The IBC Totes Authority, *Online Catalog*, IBC Tanks, [location unknown], Accessed: Nov. 1, 2019. [Online]. Available: https://www.ibctanks.com/275gallon

Appendix A: Brainstorming

Apis Consulting held a multitude of individual and team based brainstorming sessions. A crucial component of the process of brainstorming was performing a group brainwriting exercise. The raw results can be seen in Figure A1.

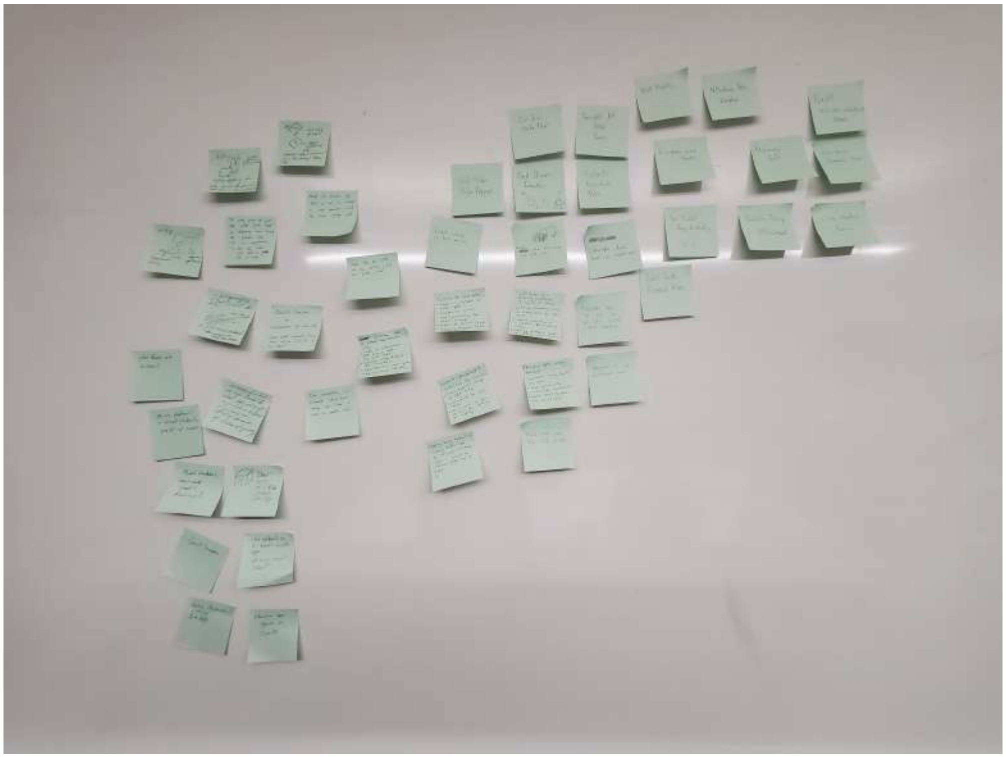


Figure A1: Rough brainwriting results

These ideas were then sorted and placed in more defined groupings to aid in understanding how each could contribute to the overall designs; this is shown in Figure A2.

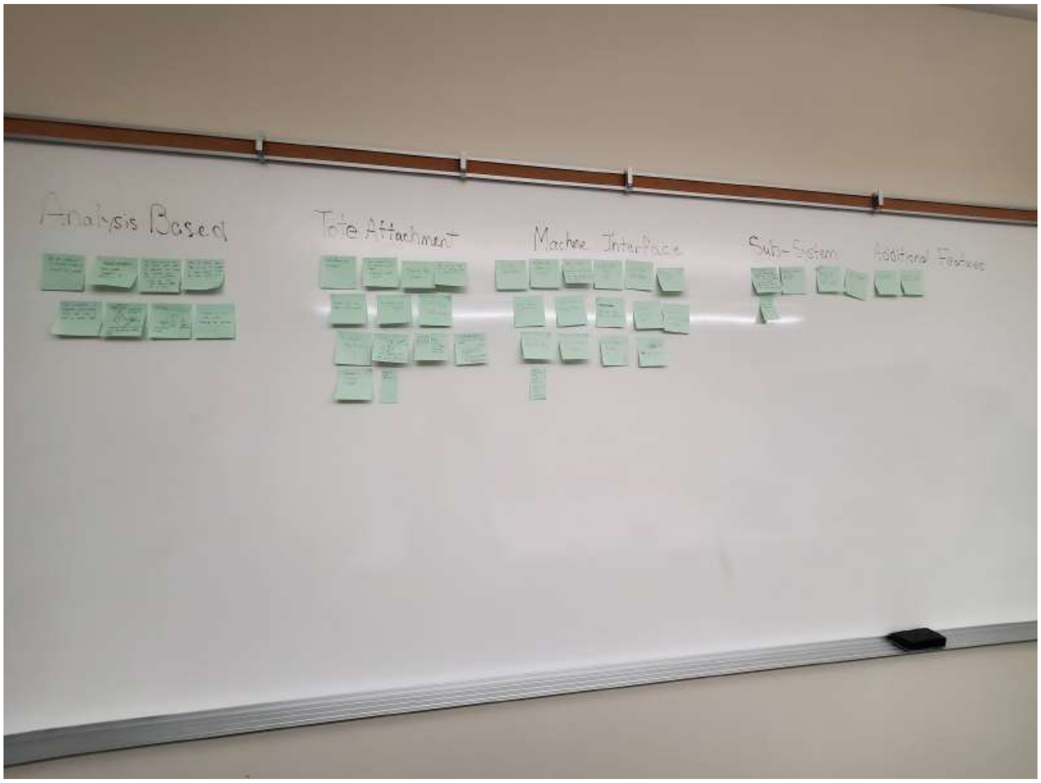


Figure A2: Organized brainwriting results

Some of the concepts that arose from the brainwriting session are summarized in Table A1.

Table A 1: Summary of Brainstorming Concepts that were not selected

Concept Name	Description	Reason for Withdrawal
Garbage Truck Arm	Similar to the mechanism used in garbage trucks to invert commercial garbage cans, the device would lift the tote over itself to invert it	Feasibility and cost concerns for a device capable of lifting a 3500 lb load over itself safely without being significantly over budget
Forklift Aided Tote Inverter Concept (Figure A3)	Stationary post with cage capable of translating up and down with the assistance of a forklift. Once at a suitable vertical position, locks engage, forklift is	Withdrawn in favour of the direct Forklift Attachment design

	withdrawn, and the machine rotates the tote	
Stationary Tote Inverter (Figure A4)	Stationary post with a cage that was able to encompass the tote, translate upwards and lock position, then rotate the tote	Feasibility questions raised about the cost associated with motor sizing and overall stability of the design
Two Post Lift with Rotation	Similar in nature to the two post lifts used in the maintenance and repair of automobiles with the additional ability of being able to rotate the tote	Budgetary constraints meant that this concept would not be feasible
Forward Inverter Forklift Attachment (Figure A5)	Tote is enclosed in the machine and locked in place. A hydraulic cylinder extends to push the tote forward and invert it. A mechanism aids in starting the rotation back to upright position.	The mechanism to aid in rotation back to upright will be placed in great stress from twisting and bending. The forward inverter brings the tote very far away from the forklift. This brings instability.
Cage Inverter Forklift Attachment (Figure A6)	Tote is enclosed in machine via the cage. Cage rotates, and fully inverts the tote.	The cage is open and there is a risk for the tote to fall out of the cage. Concept did not facilitate space for gears. Position of the shaft in the cage requires a large amount of torque.
Vertical Scissor Grab Forklift Attachment (Figure A7)	Tote is grasped by a vertical scissor style enclosure. This vertical scissor is powered by a motor through gears. Another motor is used to rotate the tote.	The concept is overly complex with the use of 2 sets of motors, and 2 sets of gear trains
Horizontal Scissor Grab Forklift Attachment (Figure A8)	Tote is grasped by a horizontal scissor style enclosure. The horizontal scissor is powered by a hydraulic cylinder to allow for open and closing. The scissor is attached to the side of the spur gear. The spur gear rotates by a motor and allows for inversion.	The horizontal scissors places the weight of the tote far from the forklift, which causes instability. Furthermore, the distance would create a large bending force.
Flat Honey Tote Tilter	Similar to the Tilter concept selected, except the stationary	Without some sort of initial angular position for the

(Figure A9)	position would be flat to reduce vertical footprint and allow for pallet jack integration	hydraulic cylinders, the lack of moment arm would prevent any form of motion

Additionally, there were some concepts that were further developed beyond the brainwriting process. In some of these cases concepts were incorporated into the Hydraulic Tilter, Rotating Frame, and Forklift Attachment’s final designs. Preliminary sketches of these concepts are shown in Figures A3 - A9.

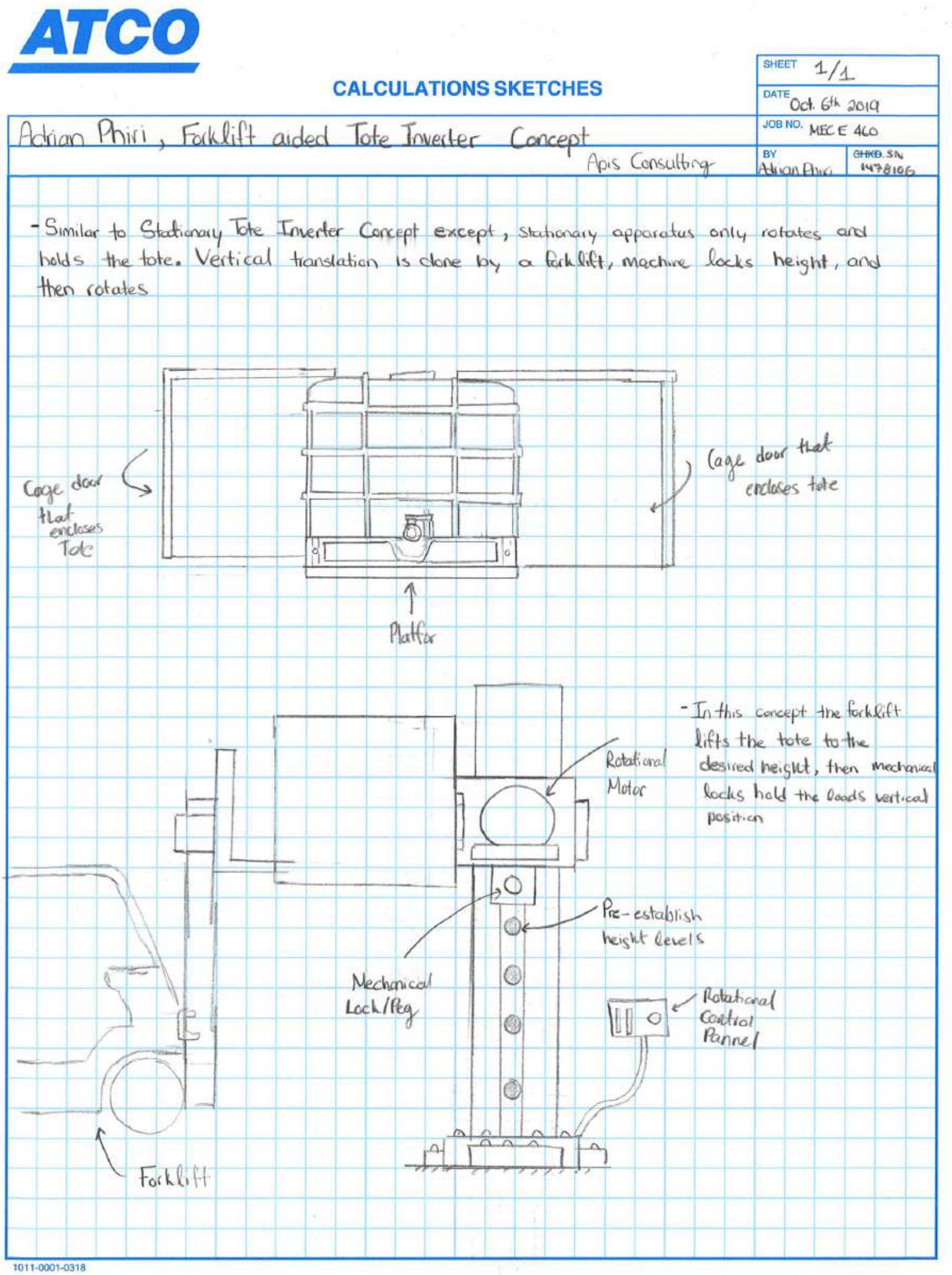


Figure A3: Forklift aided Tote Inverter concept

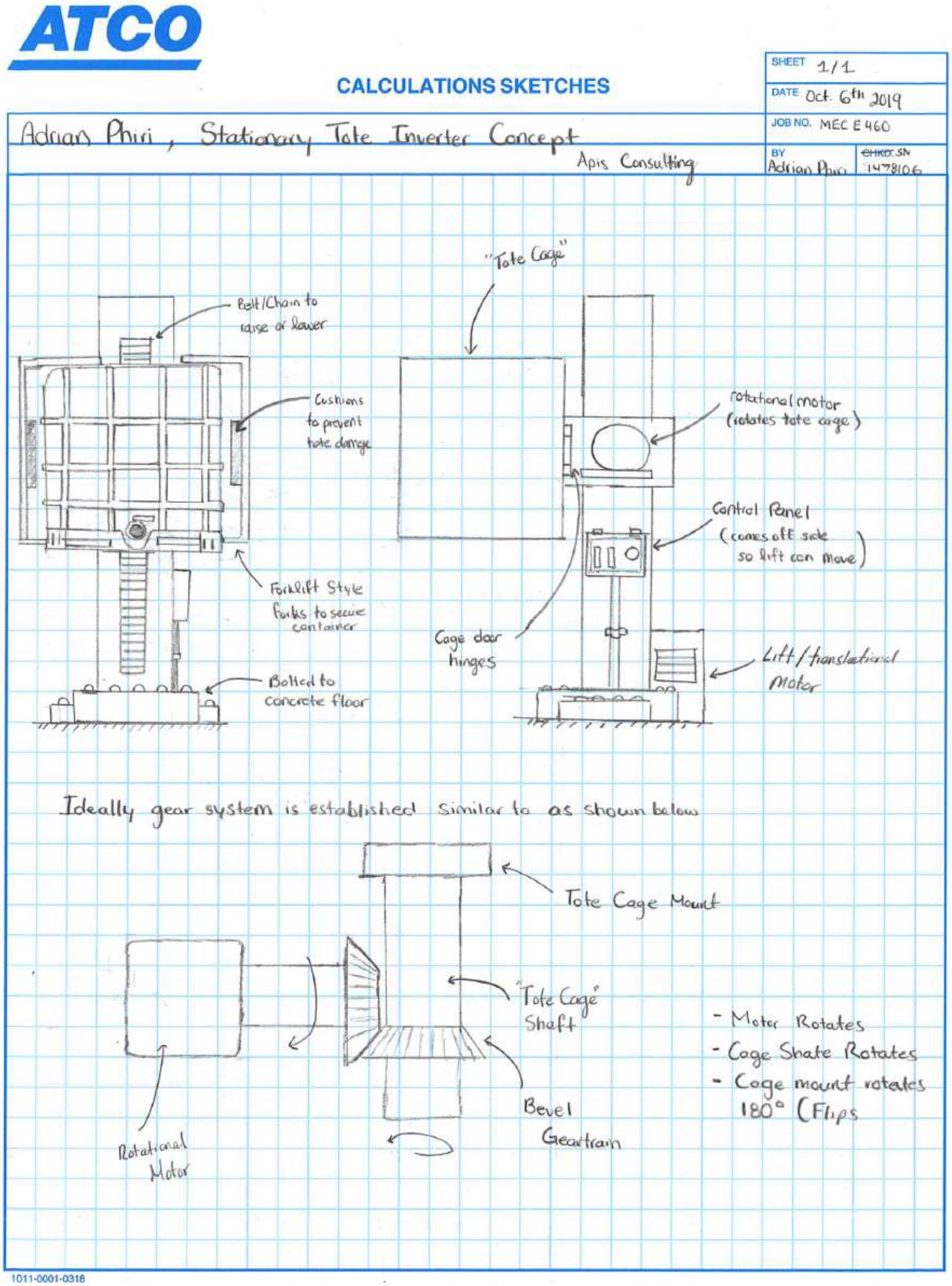


Figure A4: Stationary Tote Inverter concept

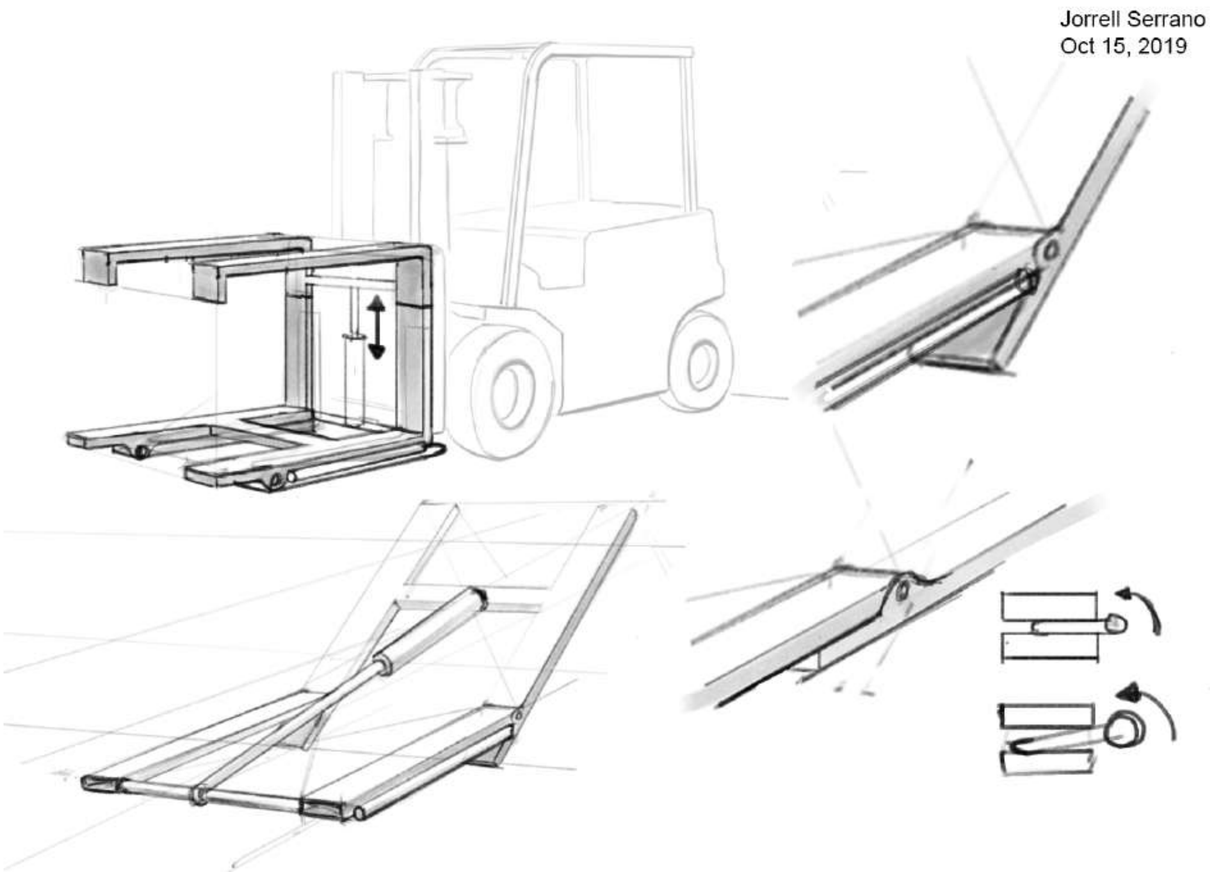


Figure A5: Forward Inverter Forklift Attachment.

Top left: full view, bottom left: concept in the middle of inversion, top right: close up of hinge, bottom right: mechanism in hinge to aid in rotation.

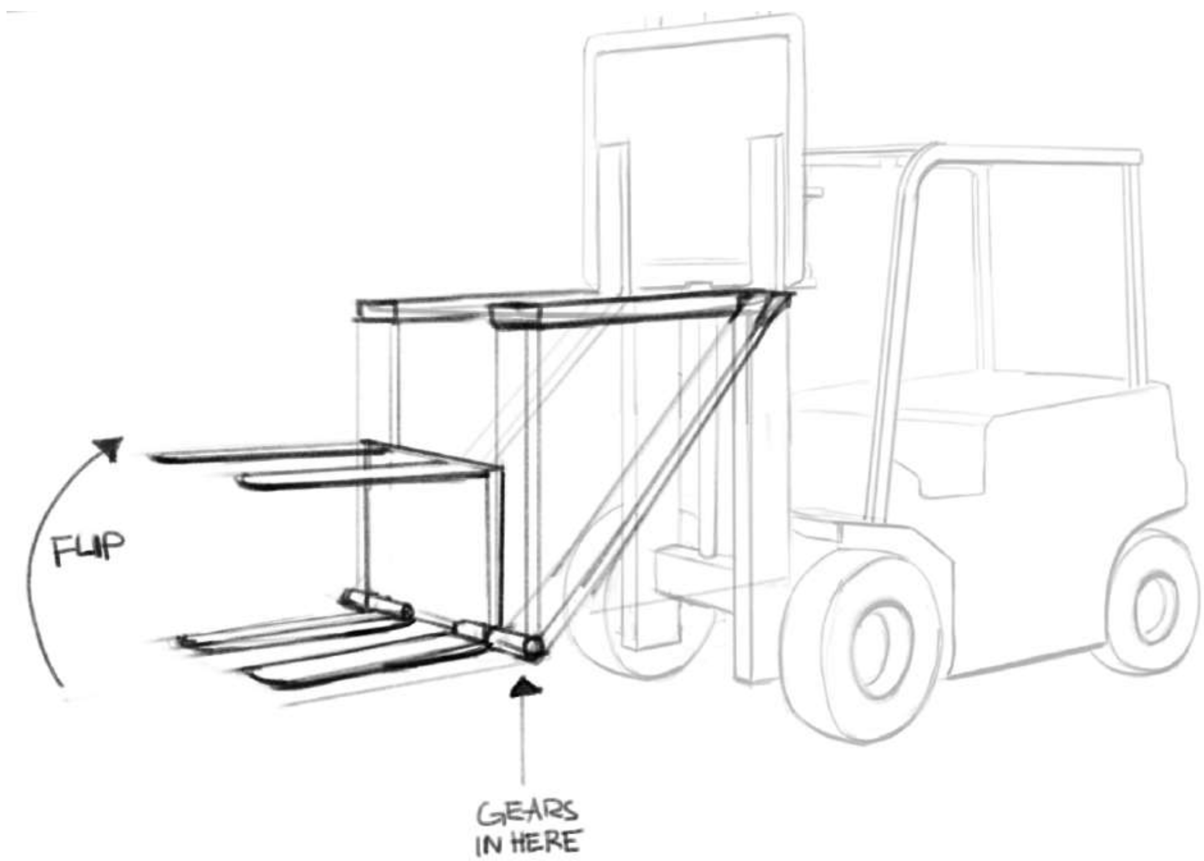


Figure A6: Cage Inverter Forklift Attachment

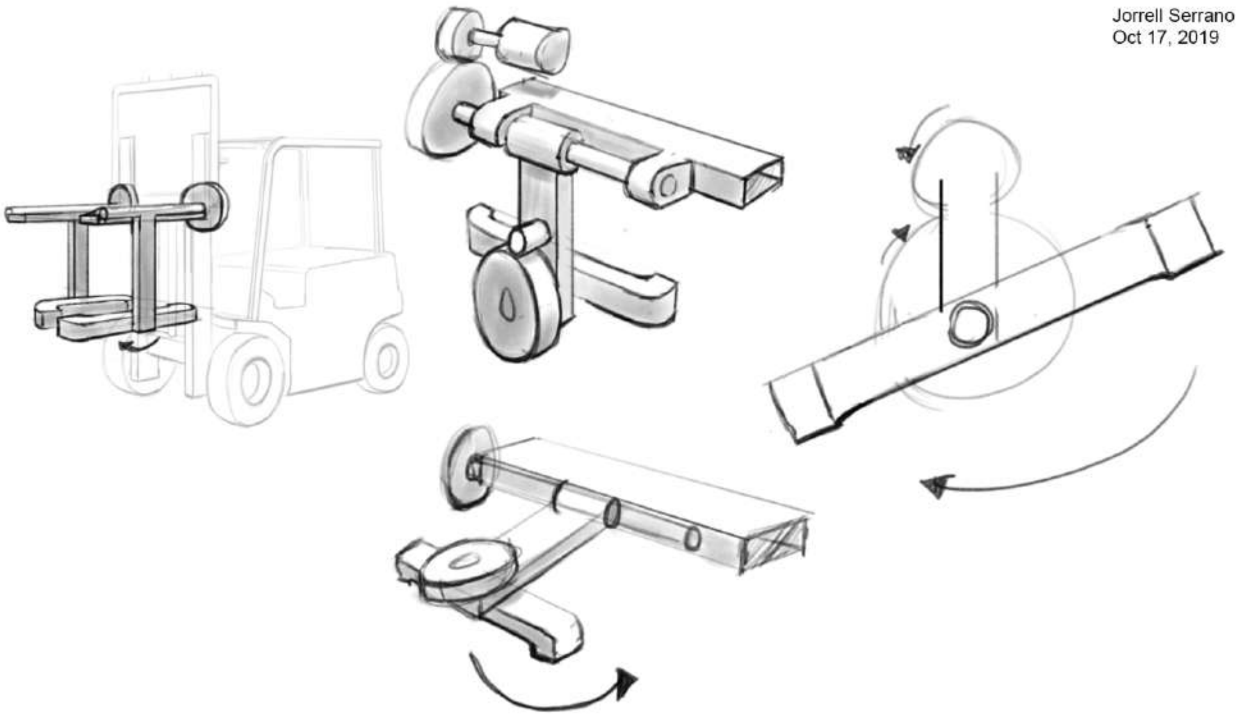


Figure A7: Vertical Scissor Grab Attachment.

Left: View of concept on a forklift. Top: close view of concept. Bottom: First motion of concept which allows grasping of tote. Right: second motion of concept which allows for inversion of tote.

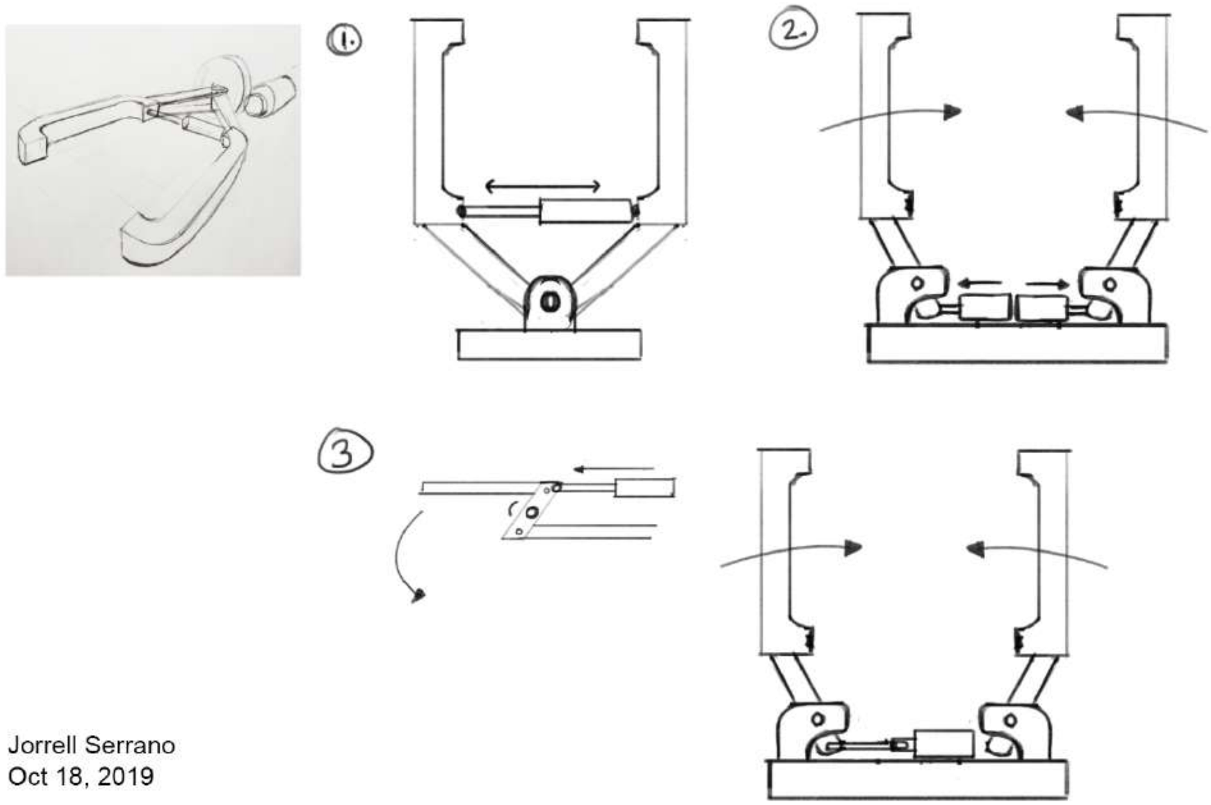


Figure A8: Horizontal Scissor Grab Forklift Attachment

Left: Isometric view of concept. 1: Grasping of tote using the retraction of the cylinder. 2: Grasping of tote using the extension of 2 cylinders. 3: Grasping of tote using the extension of one cylinder via a special connector.

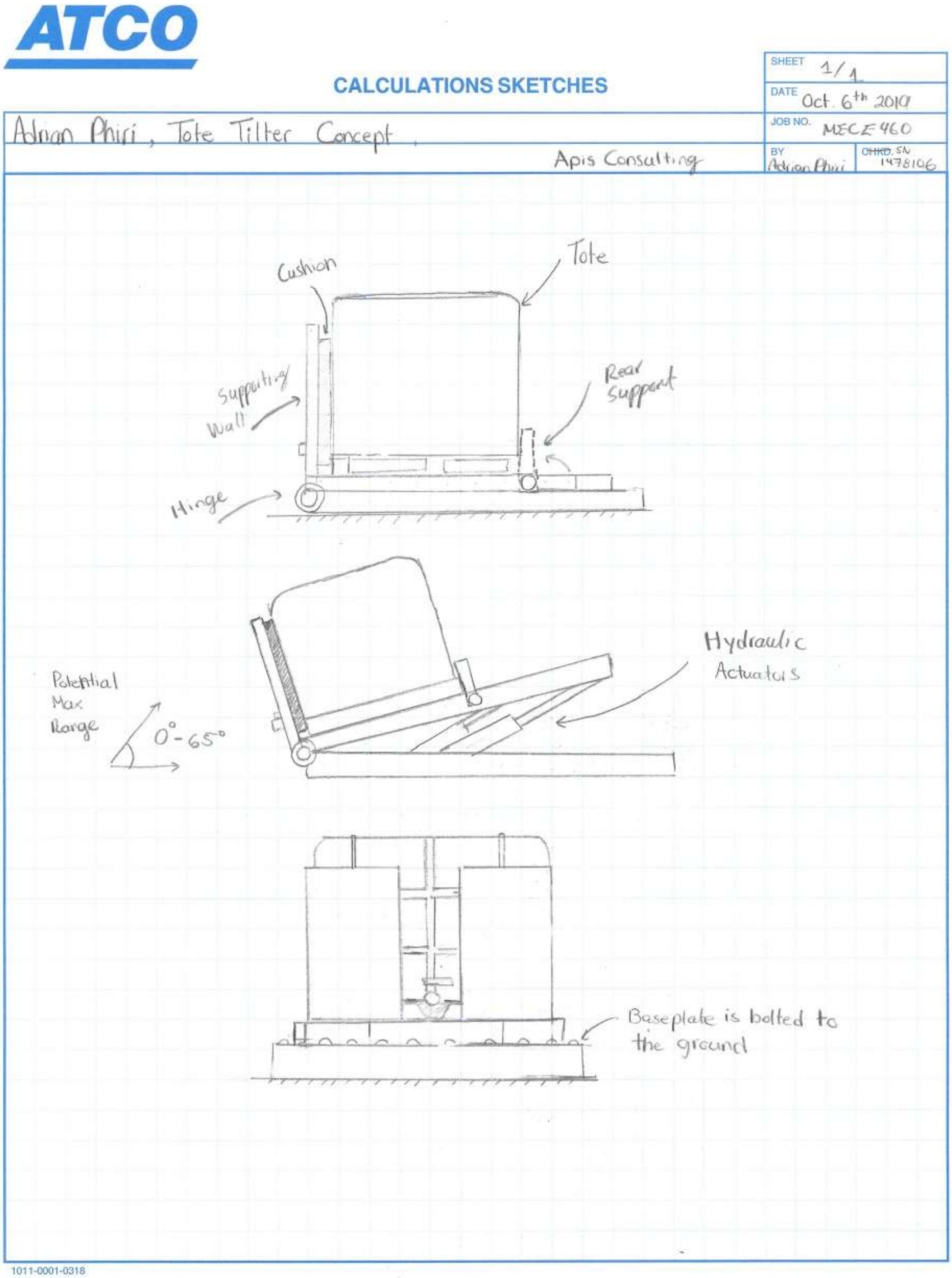


Figure A9: Flat Tote Tilter concept

Appendix B: Calculations

B.1: Concept 1 - Hydraulic Tilter

Appendix B.1 provides detailed calculations associated with the Hydraulic Tilter concept. The primary goal of this machine was to facilitate drainage of the tote's contents via the bottom spout built into the container. Although the clients scope was adjusted such that for a cleaning case the machines would be required to handle the weight of the empty tote (135 lbs), the Hydraulic Tilter was designed to support the weight of a tote full of honey (3500 lbs). With a safety factor of 1.5 this results in a maximum load of 5250 lbs.

Force Analysis for Stationary and Maximum Tilt Positions

The purpose of these calculations is to determine the forces associated with the standard operating conditions of the machine. As such, force analysis was conducted to evaluate the loads associated with critical components such as the hydraulic force, reaction forces from the hinge shaft, and the force supported by the rear supports. These values were determined for a tilt angle, θ, of 0° and 35° for the stationary and maximum angular position angle, respectively.

The Schematic and Free Body Diagrams used to conduct this analysis is collected in Figures B.1.1 and B.1.2 respectively.

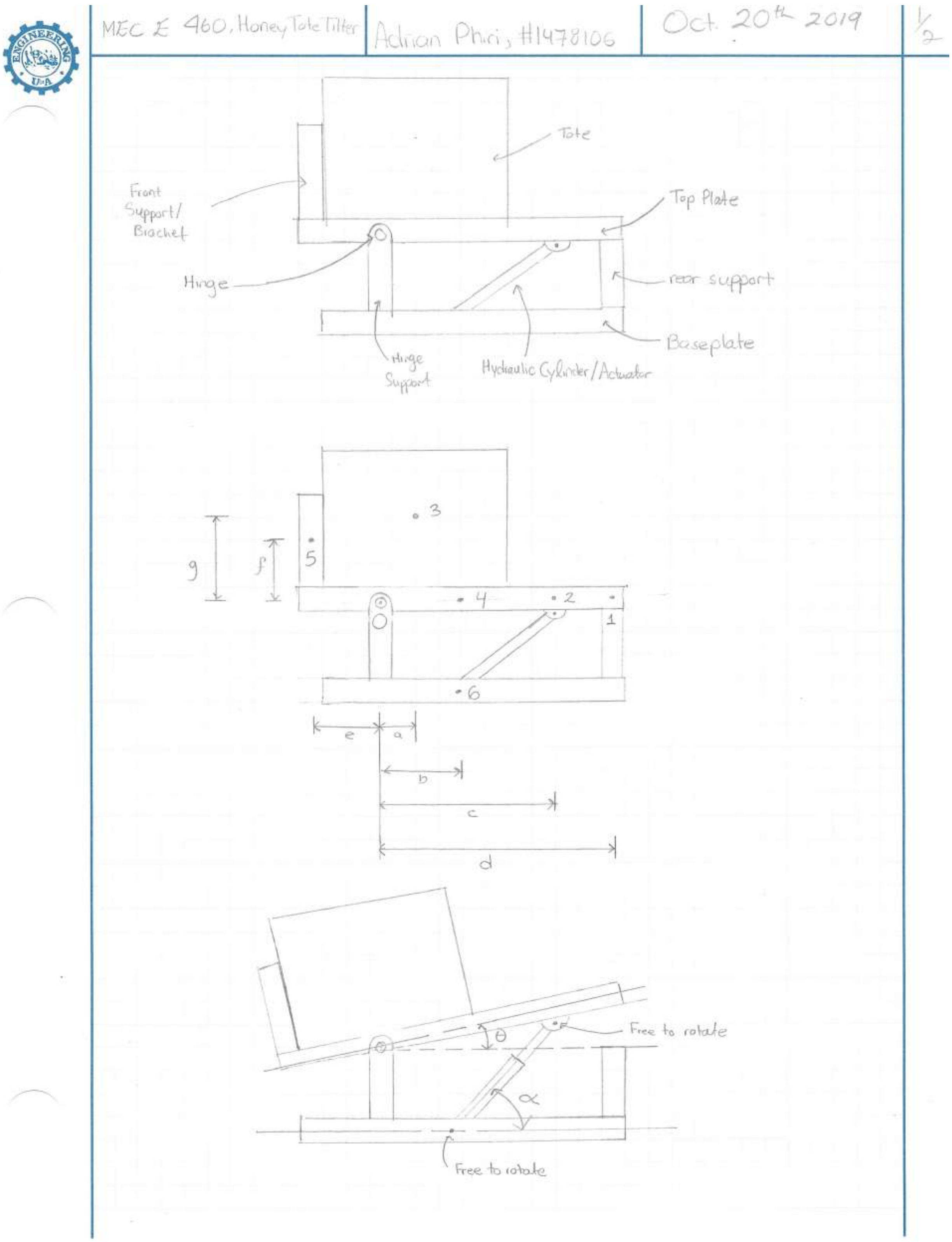
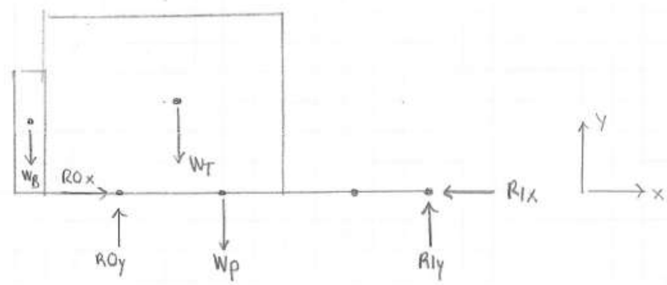


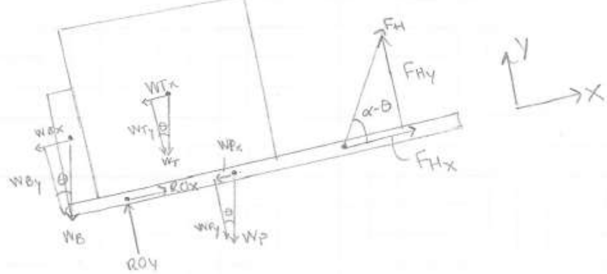
Figure B.1.1: Flat Tote Tilter concept

MEC E 460, Honey Tote Tilter | Adrian Phiri, #1478106 | Oct. 20th 2019 | 2/2

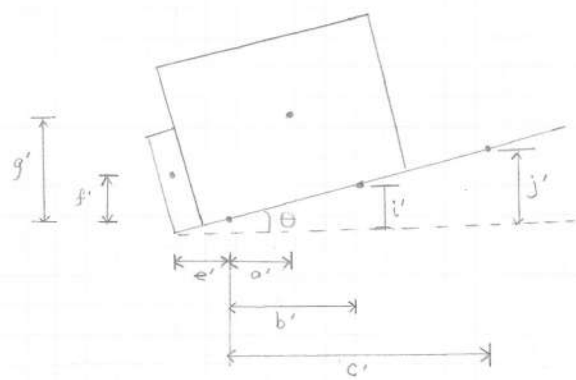
For Stationary Case,



For General Angle Case



OR



$a' = a\cos\theta - g\sin\theta$, $f' = f\cos\theta - e\sin\theta$ $\quad$ $j' = c\sin\theta$

$b' = b\cos\theta$ $\quad$ $g' = a\sin\theta + g\cos\theta$

$c' = c\cdot\cos\theta$ $\quad$ $i = b\sin\theta$

Figure B.1.2: Free body diagrams used to derive general force Equations

From Figures B.1.1 and B.1.2, the general equations were derived to determine the forces in each of the two critical cases. These derivations are shown in Figures B.1.3 - B.1.7.

ATCO

CALCULATIONS SKETCHES

MEC E 460 - Honey Tote Tilter Concept V2 & Calculations	SHEET 1/5
	DATE Oct. 20th 2019
	JOB NO. V2
	BY Adrian Phui — CHKD: SN 1478106

Objective: Determine the general equations for forces acting on the V2 Tote Tilter concept

Known:

1. Maximum worst case load associated with the tote is 3500 lb (1587.573 kg) when filled with honey of $\rho = 1420\ kg/m^3$
2. The hydraulic actuators must never have an angular displacement of 0°, otherwise it will have no moment arm and thus be unable to lift the plate
3. Base plate will have mounting brackets to hold the shaft.
4. The top plate will have bracket to attach to the shaft

Assumptions / Limitations

1. The top plate will lower at a rate at which the impact of the top plate and rear support will not produce major deformation to either component
2. The tote will remain in place during operation

1011-0001-0318

Figure B.1.3: Derivation of General Force Equations (pg. 1/5)

ATCO

CALCULATIONS SKETCHES

SHEET	2/5
DATE	Oct. 20th 2019
JOB NO.	V2
BY	Adrian Phui
CHKD: SN	1478106

MEC E 460 - Honey Tote Tilter Concept V2 & Calculations

Sketch:

Refer to Figure(s)

Analysis:

- Assume top plate rotates about point O
- a) Determine Reaction force from point A when $\theta = 0$ (Stationary Machine)
- b) Determine General Equations for F_H, O_R, A_R, F_R,
- Assume at $\theta = 0$, no forces are present that are associated with the Hydraulic Actuator

a)

Reaction Forces when $\theta = 0$

W_T = Weight of the Tote
W_P = Weight of the Top Plate
W_B = Weight of the Brace / Support
R_O = Reaction Forces at point O
R_1 = Reaction Forces at point 1
R_2 = Reaction Forces at point 2
α = Angle of Hydraulic Actuator w.r.t horizontal
θ = Angle of Top Plate w.r.t horizontal

$$\overset{+}{\rightarrow} \Sigma F_x = 0 = R_{ox} - R_{1x} \quad \Rightarrow \quad R_{ox} = R_{1x} \qquad (1)$$

$$+\uparrow \Sigma F_y = 0 = R_{oy} + R_{1y} - W_T - W_P - W_B \qquad (2)$$

$$\circlearrowleft \Sigma M_o = 0 = W_B \cdot e - W_T \cdot a - W_P \cdot b + R_{1y} \cdot d \qquad (3)$$

$$\text{From (3)}, \quad R_{1y} = \frac{1}{d}\left(W_T a + W_P b - W_B e\right)$$

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Figure B.1 4: Derivation of General Force Equations (pg. 2/5)

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CALCULATIONS SKETCHES

SHEET	3/5
DATE	Oct 20th 2019
JOB NO.	V2
BY	Adrian Phu
CHKD. SN	1478106

MEC E 460 - Honey Tote Tilter Concept V2 & Calculations

From (2),

$$R_{oy} = W_T + W_P + W_B - R_{1y} \Rightarrow R_{oy} = W_T + W_P + W_B - \frac{1}{d}\left(W_T a + W_P b - W_B e\right)$$

$$R_{oy} = W_T\left(1 - \frac{a}{d}\right) + W_P\left(1 - \frac{b}{d}\right) + W_B\left(1 + \frac{e}{d}\right)$$

<u>General Equations for O, F_H at some position α & θ</u>

$$\rightarrow \Sigma F_x = 0 = R_{Ox} + F_{Hx} - W_{Bx} - W_{Tx} - W_{Px} \quad (4)$$

$$= R_{Ox} + F_H\cos(\alpha-\theta) - W_B\sin\theta - W_T\sin\theta - W_P\sin\theta$$

$$+\uparrow \Sigma F_y = R_{Oy} + F_{Hy} - W_{By} - W_{Ty} - W_{Py} = 0 \quad (5)$$

$$= R_{Oy} + F_H\sin(\alpha-\theta) - W_B\cos\theta - W_T\cos\theta - W_P\cos\theta$$

$$(6)$$

$$(+\circlearrowleft)\Sigma M_o = 0 = W_{Bx}\cdot f + W_{By}\cdot e + W_{Tx}\cdot g - W_{Ty}\cdot a - W_{Py} b + F_{Hy} c$$

$$= 0 = W_B\sin\theta f + W_B\cos\theta e + W_T\sin\theta g - W_T\cos\theta a - W_P\cos\theta b + F_H\sin(\alpha-\theta) c$$

Using (6),

$$cF_H\sin(\alpha-\theta) = aW_T\cos\theta + W_P b\cos\theta - W_B\sin\theta\cdot f - W_B\cos\theta e - W_T\sin\theta g$$

$$F_H = \frac{1}{c\sin(\alpha-\theta)}\left[aW_T\cos\theta + bW_P\cos\theta - fW_B\sin\theta - eW_B\cos\theta - W_T\sin\theta g\right] \quad (7)$$

$$F_H = \frac{1}{c\sin(\alpha-\theta)}\left[W_T\left(a\cos\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_P b\cos\theta\right]$$

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Figure B.1.5: Derivation of General Force Equations (pg. 3/5)

ATCO

CALCULATIONS SKETCHES

SHEET	4/5
DATE	Oct 20th 2019
JOB NO.	V2
BY	Adrian Phu
CHKD: SN	1478106

MEC E 460 - Honey Tote Tilter Concept V2 & Calculations

Using (7) & (4)

$$R_{ox} = W_B\sin\theta + W_T\sin\theta + W_P\sin\theta - F_H\cos(\alpha-\theta)$$

$$= W_B\sin\theta + W_T\sin\theta + W_P\sin\theta - \left[W_T\left(a\cos\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_Pb\cos\theta\right]\frac{\cos(\alpha-\theta)}{\sin(\alpha-\theta)}$$

Using (7) & (5)

$$R_{oy} = W_B\cos\theta + W_T\cos\theta + W_P\cos\theta - F_H\sin(\alpha-\theta)$$

$$= W_B\cos\theta + W_T\cos\theta + W_P\cos\theta - \left[W_T\left(a\sin\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_Pb\cos\theta\right]\frac{\sin(\alpha-\theta)}{\sin(\alpha-\theta)}$$

Conclusions:

From the analysis of the forces acting on the system at rest, the equations for reaction forces on the pin and vertical support were determined as,

$$R_{1y} = \frac{1}{d}\left(W_Ta + W_Pb - W_Be\right) = \text{Vertical Support reaction force}$$

$$R_{oy} = W_T\left(1-\frac{a}{d}\right) + W_P\left(1-\frac{b}{d}\right) + W_B\left(1+\frac{e}{d}\right) = \text{Reaction force from Hinge}$$

General Equations for the Hydraulic & Hinge forces were also determined as functions of θ & α; where $\alpha \neq 0$ or the system would not be able to move (as the actuator would have no moment arm

$$F_H = \frac{1}{c\sin(\alpha-\theta)}\left[W_T\left(a\sin\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_Pb\cos\theta\right] = F_H(\theta,\alpha)$$

= Force provided by the hydraulic actuator / cylinder

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Figure B.1.6: Derivation of General Force Equations (pg. 4/5)

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CALCULATIONS SKETCHES

SHEET	5/5
DATE	Oct 20th 2019
JOB NO.	V2
BY	Adrian Phu
CHKD SN	1478106

MEC E 460 - Honey Tote Tilter Concept V2 & Calculations

$$R_{Ox} = W_B \sin\theta + W_T \sin\theta + W_P \sin\theta - \frac{\cos(\alpha-\theta)}{c \cdot \sin(\alpha-\theta)}\left[W_T\left(a\sin\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_P b\cos\theta\right]$$

$R_{Ox}(\theta,\alpha)$ = Reaction force in the X direction from the Hinge

$$R_{Oy} = W_B \cos\theta + W_T \cos\theta + W_P \cos\theta - \frac{1}{c}\left[W_T\left(a\sin\theta - g\sin\theta\right) - W_B\left(f\sin\theta + e\sin\theta\right) + W_P b\cos\theta\right]$$

$R_{Oy}(\theta,\alpha)$ = Reaction force in the Y direction from the Hinge

These functions can be used to evaluate variable force magnitudes at critical operating conditions of the Tote Tilter; where these conditions are associated with stationary / balanced positions (i.e where the tilter is held to drain.

1011-0001-0318

Figure B.1.7: Derivation of General Force Equations (pg. 5/5)

These formulas were than applied in SMath to determine the resulting forces using parameters associated with the size and weight of the Hydraulic Tilter’s components. These values are shown below.

Adrian Phiri & Zhe Lyu, Oct 20th 2019

Objective: Use formulas derived prior to determine the forces acting on the titler concept at the horizontal and critical angular positions

Knowns:

1. Maximum load associated with a tote full of honey with density of ρ = 1420 kg/m^3 is ~3500 lbs.

2. The hydraulic actuators must never have an angular displacement of 0 degrees, otherwise it will have no moment arm and thus be unable to lift the top plate or the tote

3. Base plate will have mounting brackets to hold the shaft (i.e. the hinge)

4. The top plat will have bracket to attach to the shaft

Assumptions/Limitations:

1. The top plate will lower at a rate at which the impact of the top plate and rear support will not produce major deformation to either component

2. The tote will remain stationary during operation
3. The machine parts are made of AISI 1020 steel and the yield strength of this material is 50991.06247 psi.

Sketch:

Refer to Figures B.1.1

Analysis:

Part 1: Use previously derived general equations to determine the reaction forces on the rear support and the hinge at the stationary position.

Part 2: Use previously derived general equations to determine the reaction forces at the hinge and the hydraulic force required to lift the top plate and the tote.

Part 1: Stationary Case

Variables: These variables were previously defined and can be seen on the derivation Figures B.1.1. For the sake of consistency, they are shown below.

Forces:

Weight of the Tote $WT := 5250\ \text{lbf}$

Weight of the Plate $WP := 742.51\ \text{lbf}$

Weight of the Bracket Pair $WB := 2 \cdot 112.33\ \text{lbf}$

Conclusion for Part 2:

The reaction forces at the hinge of the Tilter at the stationary position are,

$R0y2 = 8804.2066\ \text{lbf}$

$R0x2 = 8977.9686\ \text{lbf}$

$$R02 := \sqrt{(R0x2)^2 + (R0y2)^2} = 12574.497\ \text{lbf}$$

The force required by the Hydraulic Actuator/Cylinder at a tilt angle of 45 degrees was determined as,

$FH = -7334.3907\ \text{lbf}$ Where this value would be distributed between the two cylinders

Shear and Bending Moment Analysis for the Hinge Shaft

Given the critical nature of the Hinge Shaft about which the platform tilts, a simplified shear and bending moment analysis was conducted. This allowed for an estimation to be made regarding the size of the shaft necessary to support the loads in each position without fail; where the load under maximum angular position provides the more critical value. The Schematic and Free Body Diagrams associated with this analysis are collected in Figure B.1.8.

MEC E 460 Tiller Concept | Adrian Phiri, 1478106 | October 25th, 2019 | 1/4

Hinge Analysis

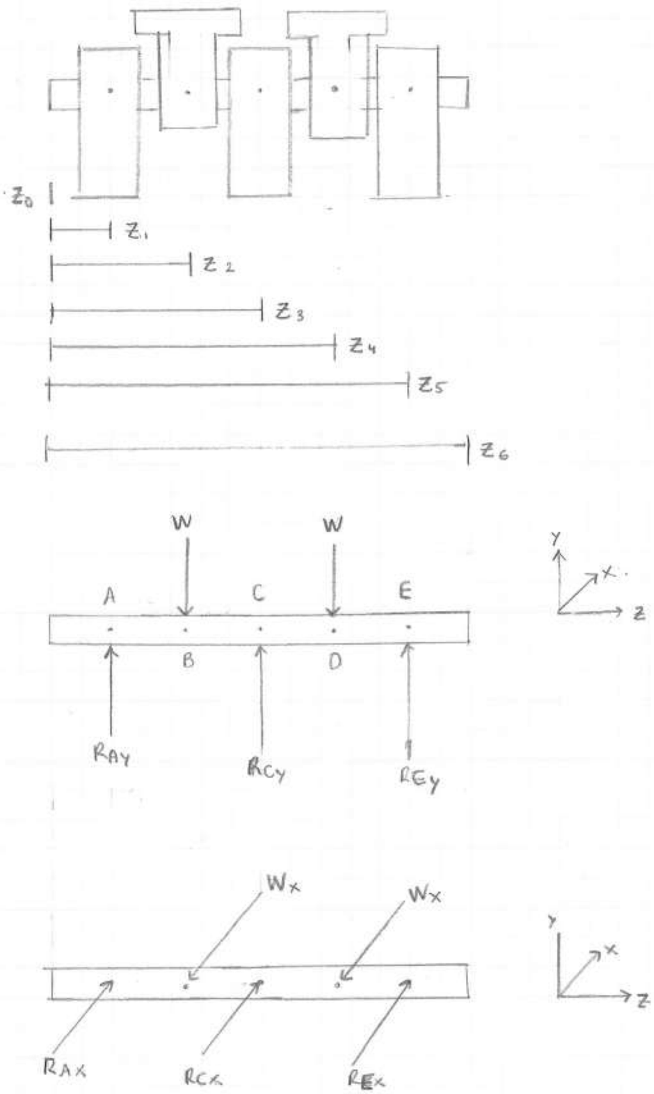


Figure B.1.8: Hinge Analysis (pg 1/4)

Using the diagrams of Figure B.1.8, the formulas for the shear and bending moment were determined via the application of the singularity equations. This can be observed in Figures B.1.9 - B.1.11.

MEC E 460 Tilter Concept	Adrian Phu, 1478106	October 25th, 2019	2/4

Objective: Prepare the general singularity equations used to determine the shear force & bending moment diagrams for the hinge at the stationary position

Known:

1. Critical reaction force values were determined using previous derivations
2. The shaft is free to rotate, acting as the hinge for the tilter. Rotation is driven by the hydraulic cylinder

Assumptions / Limitations:

1. The rod is to be evaluated as a uniform shaft
2. The weight forces acting at the two shaft mounts of the top plate will be taken as $\Sigma W_T + W_B + W_P$
3. The reaction forces from the bottom plate shaft mounts will be taken as each being 1/3 of the reaction force determined in the previous derivation
4. The Tote is loaded in the center of the machine, meaning everything the hinge support reaction forces in the x are equivalent

Sketch:

Refer to figure B.1.8

Analysis:

a) Determine general shear & bending diagram equations using singularity equations for $\theta = 0°$ (i.e stationary Tilter position)

b) Determine general shear & bending diagram equations using singularity equations for some angular position θ

Figure B.1. 9: Hinge Analysis (pg 2/4)

MEC E 460, Tiller Concept | Adrian Phiri, 1478106 | October 25^{th}, 2019 | 3/4

a)

$$+\uparrow \Sigma F_y = 0 = R_{Ay} + R_{Cy} + R_{Ey} - W_y - W_y$$ ⇒ Given the assumption of symmetric loading $R_{Ay} = R_{Cy} = R_{Ey} = R_y$

$$= 3R_y - 2W_y$$

$$\Rightarrow R_y = \frac{2W_y}{3} \quad , \quad W_y = \frac{R_{Oy}}{2}$$ from the force calculations derived prior (i.e the reaction force evenly distributed between the two top plate connection points)

$$\nearrow \Sigma F_x = 0 = R_{Ax} + R_{Cx} + R_{Ex} - W_x - W_x$$ ⇒ Given assumption of symmetric loading $R_{Ax} = R_{Cx} = R_{Ex} = R_x$

$$= 3R_x - 2W_x$$

$$\Rightarrow R_x = \frac{2W_x}{3} \quad , \quad W_x = \frac{R_{Ox}}{2}$$ ⇒ Comes from previous force calculations (i.e the reaction forces in the x direction for the tiller about the hinge shaft is evenly distributed between the two top plate connection points

• Use distribution $q(z)$, Shear $= -\int q(z)dz$, & Moment $= -\int V(z)dz$ equations to establish shear and moment diagrams

Y-Z Direction

$$q_y(z) = R_{Ay}\langle z-z_1\rangle^{-1} - W_y\langle z-z_2\rangle^{-1} + R_{Cy}\langle z-z_3\rangle^{-1} - W_y\langle z-z_4\rangle^{-1} + R_{Ey}\langle z-z_5\rangle^{-1}$$

$$V_y(z) = -\int_0^z q(z)dz$$

$$V_y(z) = -R_{Ay}\langle z-z_1\rangle^0 + W_y\langle z-z_2\rangle^0 - R_{Cy}\langle z-z_3\rangle^0 + W_y\langle z-z_4\rangle^0 - R_{Ey}\langle z-z_5\rangle^0$$

$$M_y(z) = -\int_0^z V(z)dz$$

$$M_y(z) = R_{Ay}\langle z-z_1\rangle^1 - W_y\langle z-z_2\rangle^1 + R_{Cy}\langle z-z_3\rangle^1 - W_y\langle z-z_4\rangle^1 + R_{Ey}\langle z-z_5\rangle^1$$

Figure B.1. 10: Hinge Analysis (pg 3/4)

MEC E 460 Tilter Concept | Adrian Phivi, 1478106 | October 25th 2019 | 4/4

X-Z Direction

$$q_x(z) = R_{A_x}\langle z-z_1\rangle^{-1} - W_x\langle z-z_2\rangle^{-1} + R_{C_x}\langle z-z_3\rangle^{-1} - W_x\langle z-z_4\rangle^{-1} + R_{E_x}\langle z-z_5\rangle^{-1}$$

$$V_x(z) = -R_{A_x}\langle z-z_1\rangle^{0} + W_x\langle z-z_2\rangle^{0} - R_{C_x}\langle z-z_3\rangle^{0} + W_x\langle z-z_4\rangle^{0} - R_{E_x}\langle z-z_5\rangle^{0}$$

$$M_x(z) = \underline{R_{A_x}}\langle z-z_1\rangle^{1} - \underline{W_x}\langle z-z_2\rangle^{1} + \underline{R_{C_x}}\langle z-z_3\rangle^{1} - \underline{W_x}\langle z-z_4\rangle^{1} + R_{E_x}\langle z-z_5\rangle^{1}$$

b) For the general shear & bending diagram equations for the hinge shaft at some position θ, the singularity equations are equivalent. The difference in the cases are the magnitudes of the coefficient forces

Conclusions:

For both the $\theta=0°$ & $\theta=$ Some angle cases, the distribution, shear, and moment singularity equations in the Y-Z & X-Z directions are as follows

Y-Z Direction

$$q_y(z) = R_{A_y}\langle z-z_1\rangle^{-1} - W_y\langle z-z_2\rangle^{-1} + R_{C_y}\langle z-z_3\rangle^{-1} - W_y\langle z-z_4\rangle^{-1} + R_{E_y}\langle z-z_5\rangle^{-1}$$

$$V_y(z) = -R_{A_y}\langle z-z_1\rangle^{0} + W_y\langle z-z_2\rangle^{0} - R_{C_y}\langle z-z_3\rangle^{0} + W_y\langle z-z_4\rangle^{0} - R_{E_y}\langle z-z_5\rangle^{0}$$

$$M_y(z) = \underline{R_{A_y}}\langle z-z_1\rangle^{1} - \underline{W_y}\langle z-z_2\rangle^{1} + \underline{R_{C_y}}\langle z-z_3\rangle^{1} - \underline{W_y}\langle z-z_4\rangle^{1} + R_{E_y}\langle z-z_5\rangle^{1}$$

X-Z Direction

$$q_x(z) = R_{A_x}\langle z-z_1\rangle^{-1} - W_x\langle z-z_2\rangle^{-1} + R_{C_x}\langle z-z_3\rangle^{-1} - W_x\langle z-z_4\rangle^{-1} + R_{E_x}\langle z-z_5\rangle^{-1}$$

$$V_x(z) = -R_{A_x}\langle z-z_1\rangle^{0} + W_x\langle z-z_2\rangle^{0} - R_{C_x}\langle z-z_3\rangle^{0} + W_x\langle z-z_4\rangle^{0} - R_{E_x}\langle z-z_5\rangle^{0}$$

$$M_x(z) = \underline{R_{A_x}}\langle z-z_1\rangle^{1} - \underline{W_x}\langle z-z_2\rangle^{1} + \underline{R_{C_x}}\langle z-z_3\rangle^{1} - W_x\langle z-z_4\rangle^{1} + R_{E_x}\langle z-z_5\rangle^{1}$$

Where the only major difference between the $\theta=0$ & $\theta=$ Some angle cases is the magnitude of reaction forces

Figure B.1.11: Hing Analysis (Pg 4/4)

These equations were then applied in SMath to provide Shear and Bending Moment Diagrams based upon the magnitudes of the forces in the two operating cases being evaluated. These calculations are shown below.

Shear and Bending moment Analysis for the hinge shaft

Objective:

Evaluate the stresses associated with the hinge,the Tilter

Knowns:

1. The forces acting on these critical points can be determined using using the equations previously derived (i.e. R0, FH, R1)

2. At the initial angular position the only forces acting on the hyrdaulic actuator/cylinder will be the mechanisms own self wieght

Assumptions/Limitations:

1. The shaftwill be modelled as a simple cylindircal rod for this case. Further evaluation with the potential addition of keys or steps would be considered once the FEA stage has begun.

Sketch:

Refer to Figures B.1

Analysis:

Part A) For the Stationary Case:

Shear and Bending Moment Diagrams

Forces

$$W_y := \frac{R0y}{2} = 1864.2047\ lbf$$

$$R_y := \frac{2}{3} \cdot W_y = 1242.8031\ lbf$$

$$RA_y := R_y$$

$$RC_y := RA_y$$

$$RE_y := RC_y$$

$$W_x := \frac{R0x}{2} = 0\ lbf$$

$$R_x := \frac{2}{3} \cdot W_x = 0\ lbf$$

$$RA_x := R_x$$

$$RC_x := RA_x$$

$$RE_x := RC_x$$

Distances

Reference from left end of the Hinge Shaft	$z_0 := 0\ in$
Distance to center of first base plate hinge support	$z_1 := 10\ in$
Distance to center of first top plate mount	$z_2 := 17.5\ in$
Distance to center of second base plate hinge support	$z_3 := 25\ in$
Distance to center of second top plate mount	$z_4 := 32.5\ in$
Distance to center of third base plate hinge support	$z_5 := 40\ in$

Distance from the neutral axis of the shafts Cross-sectional area $r := 1\ \text{in}$

Hinge Shaft cross-sectional area $A_c := \pi \cdot r^2$

$A_c = 3.1416\ \text{in}^2$

Hinge Shaft moment of Inertia

$$I := \frac{\pi \cdot r^4}{4}$$

Establishing Singularity Function

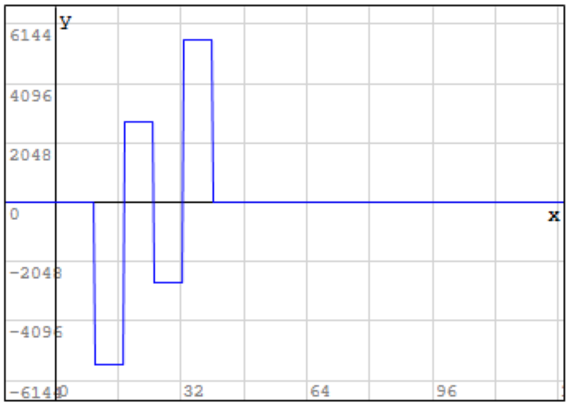
$$S(z, a, n) := \text{if } ((z - a) > 0) \wedge (n \geq 0) \quad (z-a)^n \quad \text{else} \quad 0$$

$$V_y(z) := \left(\left(-RA_y\right) \cdot S(z, z_1, 0) + W_y \cdot S(z, z_2, 0) - RC_y \cdot S(z, z_3, 0) + W_y \cdot S(z, z_4, 0) - RE_y \cdot S(z, z_5, 0)\right)$$

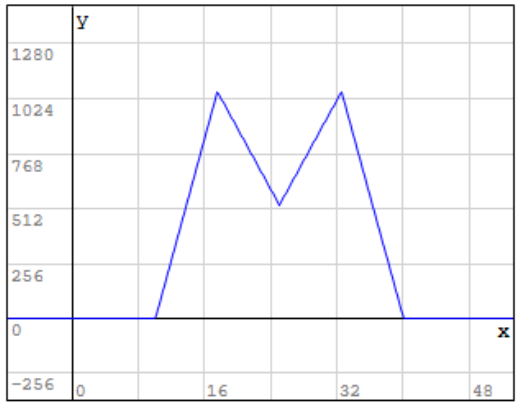
$$M_y(z) := \left(RA_y \cdot S(z, z_1, 1) - W_y \cdot S(z, z_2, 1) + RC_y \cdot S(z, z_3, 1) - W_y \cdot S(z, z_4, 1) + RE_y \cdot S(z, z_5, 1)\right)$$

$$V_x(z) := \left(\left(-RA_x\right) \cdot S(z, z_1, 0) + W_x \cdot S(z, z_2, 0) - RC_x \cdot S(z, z_3, 0) + W_x \cdot S(z, z_4, 0) - RE_x \cdot S(z, z_5, 0)\right)$$

$$M_x(z) := \left(RA_x \cdot S(z, z_1, 1) - W_x \cdot S(z, z_2, 1) + RC_x \cdot S(z, z_3, 1) - W_x \cdot S(z, z_4, 1) + RE_x \cdot S(z, z_5, 1)\right)$$

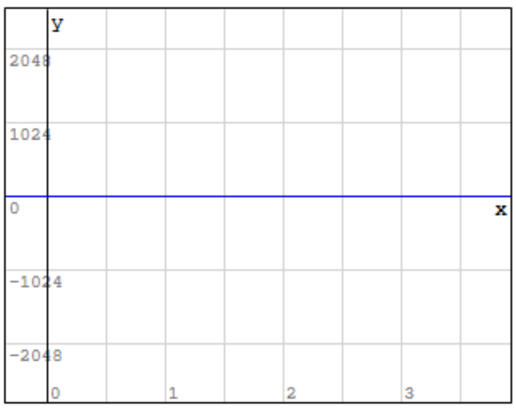


$V_y(x\ \mathrm{in})$

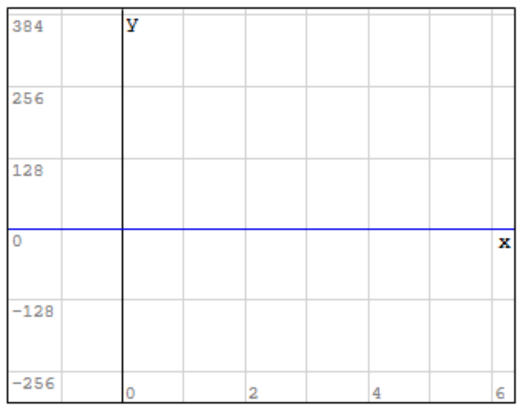


$M_y(x\ \mathrm{in})$

$$V_y\left(\frac{35}{12}\ \mathrm{ft}\right) = 5528.264\ \mathrm{N}$$

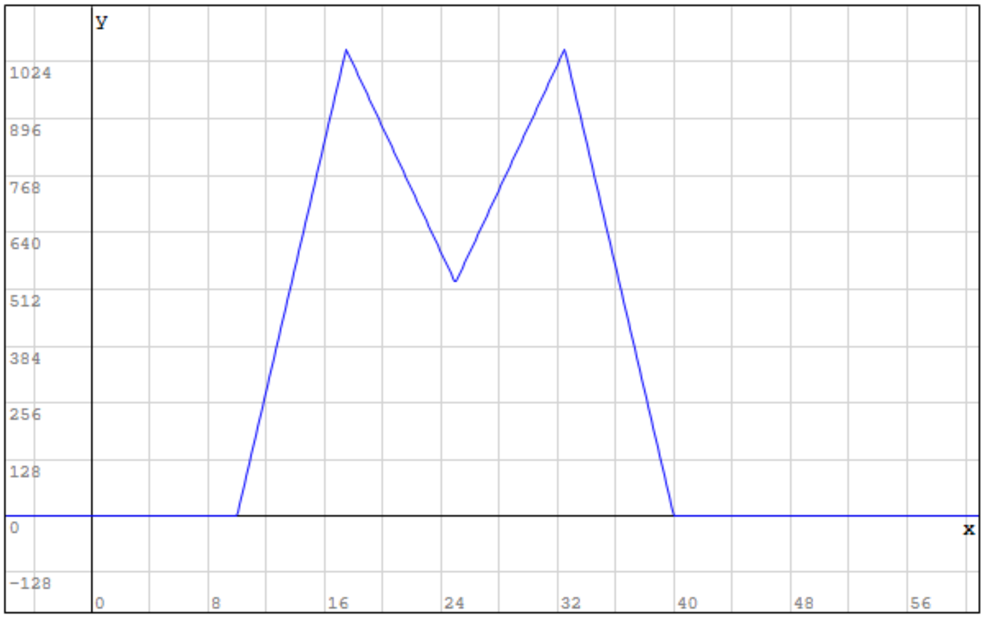


$V_x(x\ \mathrm{ft})$



$M_x(x\ \mathrm{ft})$

$$M_{resultant}(z) := \sqrt{(M_x(z))^2 + (M_y(z))^2}$$



$M_{resultant}(x\ \text{in})$

Part B) For the Critical Angular Position of θ = 35 degrees

Forces

$$W_{2y} := \frac{R0y2}{2} = 4402.1033\ \text{lbf}$$

$$R_{2y} := \frac{2}{3} \cdot W_{2y} = 2934.7355\ \text{lbf}$$

$$RA_{2y} := R_{2y}$$

$$RC_{2y} := RA_{2y}$$

$$RE_{2y} := RC_{2y}$$

$$W_{2x} := \frac{R0x2}{2} = 4488.9843\ \text{lbf}$$

$$R_{2x} := \frac{2}{3} \cdot W_{2x} = 2992.6562\ \text{lbf}$$

$$RA_{2x} := R_{2x}$$

$$RC_{2x} := RA_{2x}$$

$$RE_{2x} := RC_{2x}$$

Distances

Reference from left end of the Hinge Shaft $z_0 := 0\ \text{in}$

Distance to center of first base plate hinge support $z_1 := 10\ \text{in}$

Distance to center of first top plate mount $z_2 := 17.5\ \text{in}$

Distance to center of second base plate hinge support $z_3 := 25\ \text{in}$

Distance to center of second top plate mount $z_4 := 32.5\ \text{in}$

Distance to center of third base plate hinge support $z_5 := 40\ \text{in}$

Establishing Singularity Function

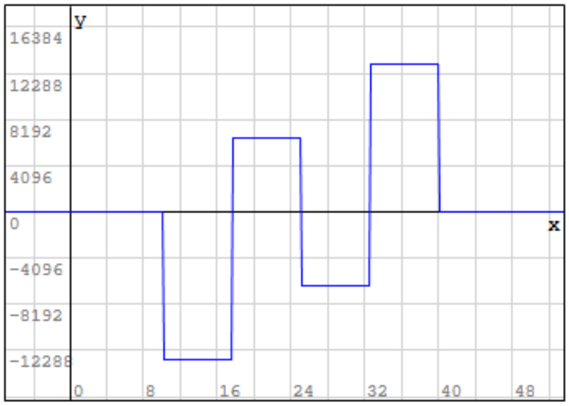
$$S(z, a, n) := \text{if } \left((z-a) > 0\right) \wedge (n \geq 0) \quad (z-a)^n \quad \text{else} \quad 0$$

$$V_{2y}(z) := \left(\left(-RA_{2y}\right) \cdot S(z, z_1, 0) + W_{2y} \cdot S(z, z_2, 0) - RC_{2y} \cdot S(z, z_3, 0) + W_{2y} \cdot S(z, z_4, 0) - RE_{2y} \cdot S(z, z_5, 0)\right)$$

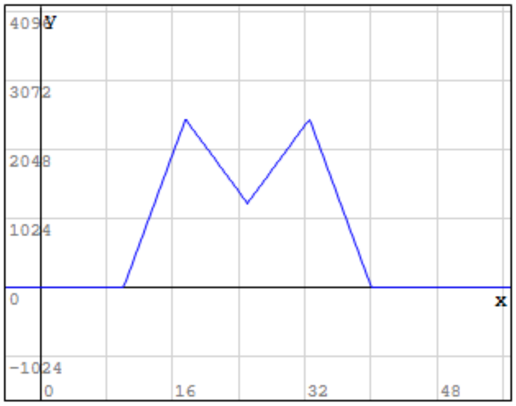
$$M_{2y}(z) := \left(RA_{2y} \cdot S(z, z_1, 1) - W_{2y} \cdot S(z, z_2, 1) + RC_{2y} \cdot S(z, z_3, 1) - W_{2y} \cdot S(z, z_4, 1) + RE_{2y} \cdot S(z, z_5, 1)\right)$$

$$V_{2x}(z) := \left(\left(-RA_{2x}\right) \cdot S(z, z_1, 0) + W_{2x} \cdot S(z, z_2, 0) - RC_{2x} \cdot S(z, z_3, 0) + W_{2x} \cdot S(z, z_4, 0) - RE_{2x} \cdot S(z, z_5, 0)\right)$$

$$M_{2x}(z) := \left(RA_{2x} \cdot S(z, z_1, 1) - W_{2x} \cdot S(z, z_2, 1) + RC_{2x} \cdot S(z, z_3, 1) - W_{2x} \cdot S(z, z_4, 1) + RE_{2x} \cdot S(z, z_5, 1)\right)$$



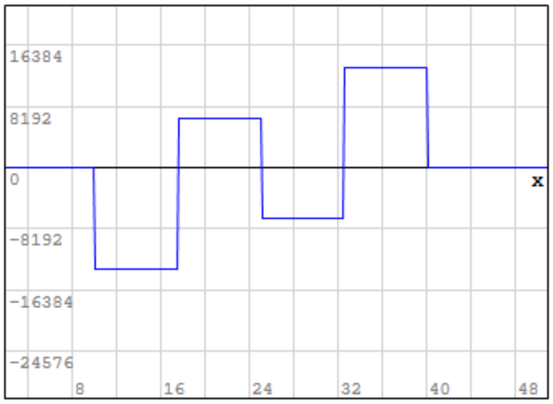
$V_{2y}(x\ \text{in})$



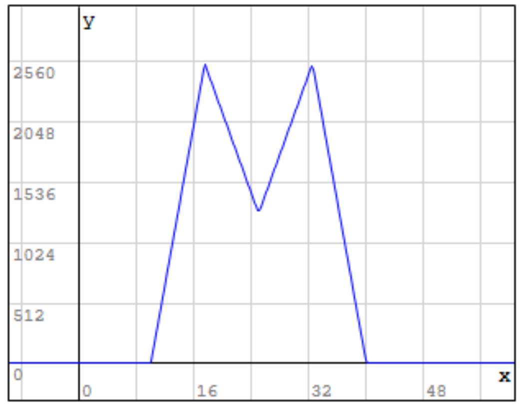
$M_{2y}(x\ \text{in})$

$$V_{2y}\left(\frac{35}{12}\ \text{ft}\right) = 13054.354\ \text{N}$$

$$V_{2y}\left(\frac{35}{12}\ \text{ft}\right) = 2934.736\ \text{lbf}$$



$V_{2x}(x\ \mathrm{in})$

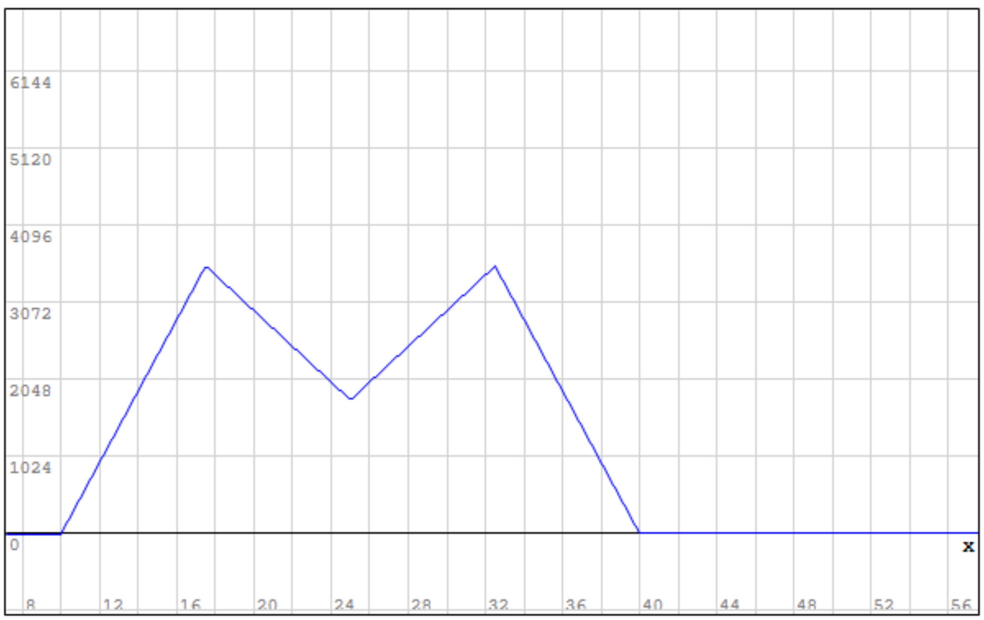


$M_{2x}(x\ \mathrm{in})$

$$V_{2x}\left(\frac{35}{12}\ \mathrm{ft}\right) = 13311.998\ \mathrm{N}$$

$$V_{2x}\left(\frac{35}{12}\ \mathrm{ft}\right) = 2992.656\ \mathrm{lbf}$$

$$M_{resultant2}(z) := \sqrt{\left(M_{2x}(z)\right)^2 + \left(M_{2y}(z)\right)^2}$$



$M_{resultant2}(x\ \mathrm{in})$

Maximum Bending mombent occurs at 17.5 and 32.5 inch

$$M_{resultant2}(17.5\ \mathrm{in}) = 31436.2424\ \mathrm{lbf\ in}$$

Stress Analysis

Stress analysis was conducted on the hinge shaft and the rear supports to determine the minimum area appropriate to support the load. The Hinge shaft analysis was conducted using values associated with the maximum angular operating position of θ = 35°. The stress analysis conducted on the rear supports was done using values associated with the stationary position, as this is the only point at which these supports would be under load.

Stress Analysis

Objective

The Following Checks the maximum bending stress with the yield strength

$$\sigma_{Design} := \frac{M_{resultant2}(17.5\ in) \cdot r}{I} = 40025.867\ psi$$

$$\sigma_{yield} := 50991.06247\ psi$$

$$\sigma_{Design} < \sigma_{yield}$$

The following Calculates the minimum radius of the shaft when the stress reaches yield strength

$$\sigma_{Bending} := \frac{Mr}{I_{area}} \qquad I_{area} := \frac{\pi r^4}{4}$$

Then the minimum radius is calculated as

$$r_{min} := \sqrt[3]{\frac{4 \cdot M_{resultant2}(17.5\ in)}{\sigma_{yield} \cdot \pi}} = 0.9225\ in$$

The designed radius of the shaft is 1 inch

$$r_{Design} > r_{min}$$

SO the designed shaft is safe to withstand the maximum bending on shaft.

Minimum Area of rear support

To calculated the minimum area of the rear support. It is assumed that the stress in the support reaches the yield strength

from

$$\sigma := \frac{F}{A} \qquad \sigma_{yield} := 50991.06247\ psi$$

Then the minimum area is calculated as

$$A_{min} := \frac{R1y}{\sigma_{yield}} = 0.0488\ in^2$$

Currently the designed area of the rear support is 58.4 in^2

$$A_{Design} := 5.84\ in^2$$

$$A_{Design} > A_{min}$$

So currently the rear is safe to withstand the load of machine at rest (flat)

B.2: Concept 2 - Rotating Frame

Appendix B.2 provides detailed calculations associated with the Rotating Frame concept. These calculations assume a tote loaded with 100 kg of material on its bottom surface. This is intended to simulate a tote with a thick layer of crystallized honey to be washed out.

Minimum Torque Analysis

The purpose of these calculations is to determine the minimum torque required to rotate the subframe and tote. The calculation reflects the condition when the tote is oriented with its bottom oriented to the side, which will require the maximum applied torque.

The free body diagrams used to conduct this analysis are shown in Figure B.2.1.

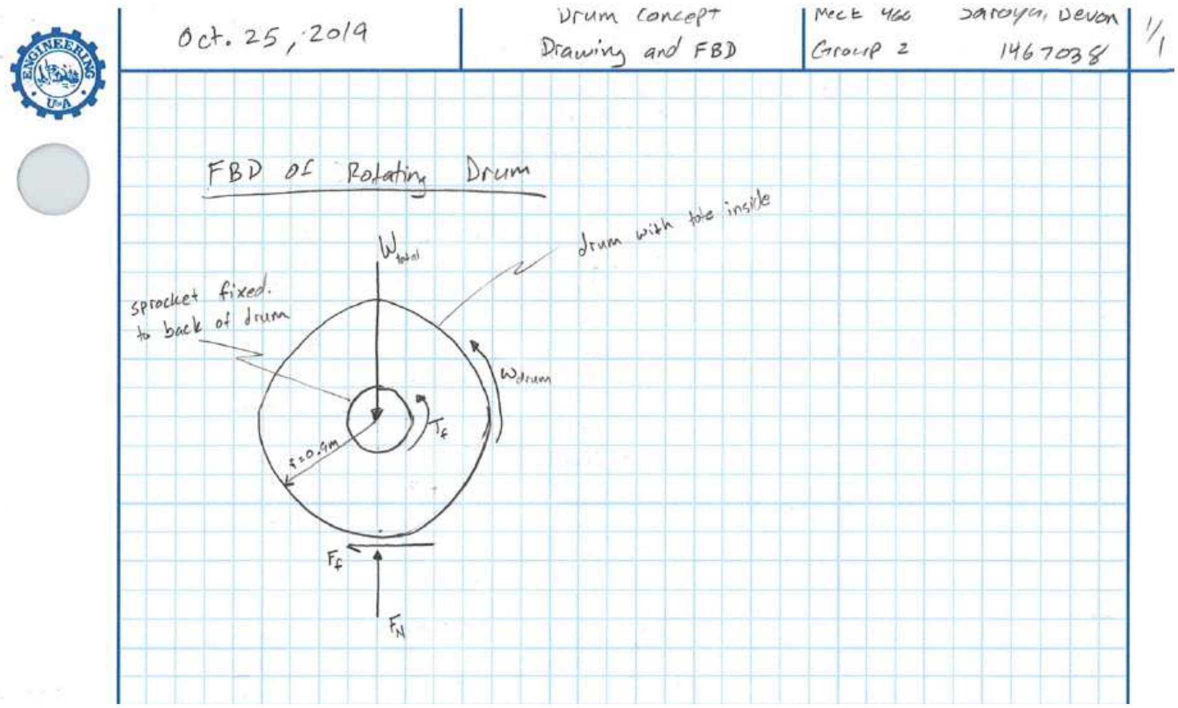


Figure B.2. 1: Free Body Diagram for torque analysis

Devon Saroya and Alex Rodd, Oct 25 2019

Objective: Determine torque and power required to rotate drum and tote based on weight of components and bearing coefficient of friction

Knowns:

1. Gravitational acceleration

$g := 9.81 \frac{m}{s^2}$

2. Empty tote mass and weight

$m_{tote} := 118.5\ kg \qquad W_{tote} := m_{tote} \cdot g = 1162.485\ N$

3. Drum mass and weight

$m_{drum} := 118.5\ kg \qquad W_{drum} := m_{drum} \cdot g = 1162.485\ N$

4. Estimated load in tote before cleaning

$m_{load} := 100\ kg \qquad W_{load} := m_{load} \cdot g = 981\ N$

5. Drum radius

$r_{drum} := 0.9\ m$

6. Load position from axis of rotation

$r_{load} := 20\ in = 0.508\ m$

7. Ball bearing coefficient of friction (From www.koyo.jtekt.co.jp)

$\mu_{BB} := 0.002$

8. Design inversion time

$t_{drum} := 20\ s$

Assumptions/Limitations:

1. Friction between drum exterior and guide rollers is negligible
2. Centre of gravity of tote and drum are located at centre of rotational axis
3. Load is equally supported by front bearings and rear bearing
4. Front bearing load is equally supported by both bearings
5. Normal force at bearings is equal to total weight of drum, tote, and load
6. Load is at fixed distance from axis of rotation

Sketch:

Refer to Figure B.2.1

Analysis:

Total weight of drum and tote with safety factor

$W_{total} := W_{tote} + W_{load} + W_{drum} = 3305.97\ N$

Normal force at bearings

$F_N := W_{total}$

Tangential force at drum-bearing interface due to bearing friction

$F_f := F_N \cdot \mu_{BB} = 6.6119\ N$

Torque required to overcome friction for drum rotation

$T_f := F_f \cdot r_{drum} = 5.9507\ N\ m$

Torque required to overcome load at one quarter rotation

$T_{load} := W_{load} \cdot r_{load} = 498.348\ N\ m$

Total torque required

$T_{total} := T_f + T_{load} = 504.2987\ N\ m$

Drum rotational velocity

$\omega_{drum} := \frac{0.5\ rev}{t_{drum}} = 1.5\ rpm$

$\omega_{drum} = 0.1571\ \frac{rad}{s}$

Minimum power required

$P_{drum} := T_{total} \cdot \omega_{drum} = 79.2151\ W$

$P_{drum} = 0.1062\ hp$

Stress Analysis

The purpose of these calculations is to determine the stress and strain conditions at the bottom of the main frame support legs. As above, this analysis assumes a tote containing 100 kg of residue at its bottom.

The free body diagram used to conduct this analysis is shown in Figure B.2.2.

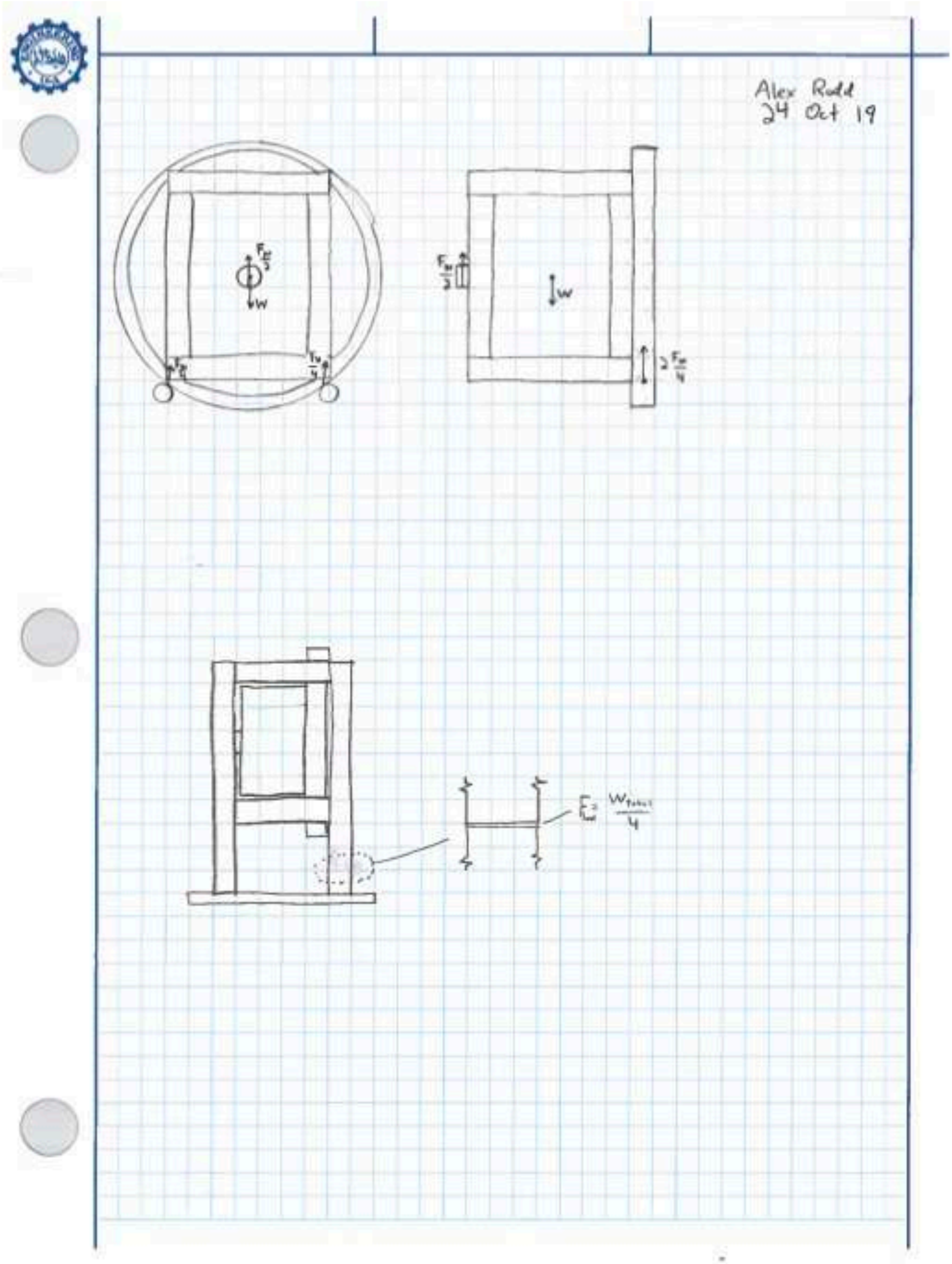


Figure B.2.2: Free Body Diagram for stress analysis

Devon Saroya and Alex Rodd, Oct 25 2019

Objective: Determine force, stress, and strain in suppport legs based on weight of assembly and material properties of steel

Knowns:

1. Gravitational acceleration

$$g := 9.81 \frac{m}{s^2}$$

2. Weight and mass of entire system

$$m_{total} := 394.72\ kg \quad W_{total} := m_{total} \cdot g = 3872.2032\ N$$

3. Mass and weight of baseplate

$$m_{plate} := 222.41\ kg \quad W_{plate} := m_{plate} \cdot g = 2181.8421\ N$$

4. Cross sectional area of legs

$$A := 4 \cdot (2\ in - 0.065\ in) \cdot 0.065\ in = 0.0003\ m^2$$

$$A_{total} := 4 \cdot A = 0.0013\ m^2$$

5. Material properties for mild steel (from www.azom.com)

5a. Young's modulus

$$E := 205\ GPa$$

5b. Yield strength

$$\sigma_y := 370\ MPa$$

Assumptions/Limitations:

1. Load is applied in the vertical direction only
2. Buckling failure is not considered
3. Force is applied equally to all four legs

Sketch: Refer to Figure B.2.2

Analysis: Total load at bottom of legs

$$W_{load} := W_{total} - W_{plate} = 1690.3611\ N$$

Stress on legs due to load

$$\sigma_{load} := \frac{W_{load}}{A_{total}} = 1.302\ MPa$$

Strain in legs due to load

$$\varepsilon_{load} := \frac{\sigma_{load}}{E} = 6.351 \cdot 10^{-6}$$

Percent of yield strength

$$\frac{\sigma_{load}}{\sigma_y} = 0.3519\ \%$$

B.3: Concept 3 - Forklift Attachment

Appendix B.3 explains in detail the calculations involved with the Forklift Attachment. The main requirement is that the Forklift Attachment must be able to lift and rotate a tote of 135 lb with a safety factor of 3. That means that the components of the Forklift Attachment must be able to withstand the stresses from a 405 lb load.

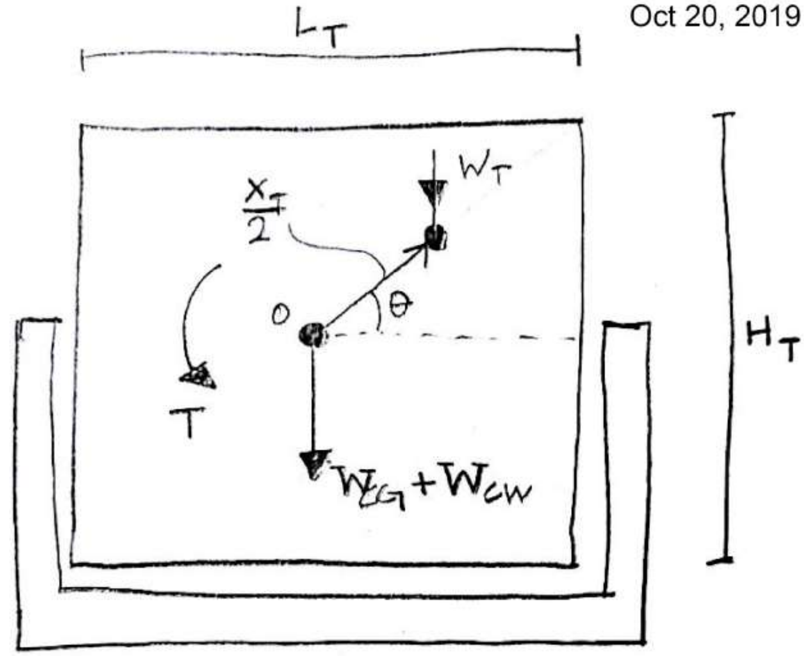
Minimum Torque Calculation

The purpose of this calculation is to find the minimum torque required to rotate the tote. The calculation is done with the critical case scenario. The center of gravity of the tote is placed halfway between the centre of rotation and the corner of the tote. This is to cover scenarios where there is crystallized honey in the tote creates an off-center weight.

Additionally, the cage and counterweight combined center of gravity is in the center of rotation. The cage and counterweight are designed this way in order to minimize the required torque. This was completed through SolidWorks.

The free body diagram for this calculation is in Figure B.3.1.

FBD by Jorrell Serrano
Oct 20, 2019



$$\sum M_O = 0 = T - W_T \frac{x_T}{2} \cos\theta = 0$$

$$\boxed{T = W_T \frac{x_T}{2} \cos\theta}$$

Figure B.3.1: Free body diagram of the tote, cage, and counterweight

Where:

- L_T is the length of the tote
- H_T is the height of the tote
- x_T is the moment arm of the weight of the tote
- W_T is the weight of the tote
- T is the torque

θ is the angle between the x_T and the horizontal

W_{CG} is the weight of the cage

W_{CW} is the weight of the counterweight

From the free body diagram, the formula for the minimum torque is derived.

Calculation by Jorrell Serrano Oct 20, 2019

Minimum Torque to Rotate the Cage and Tote

Objective

Determine the required torque to initiate rotation of the cage and the tote.

Knowns

The weight of the tote, weight of the counterweight and the weight of the cage is known.
The dimensions of the tote is known.

Assumptions

1. All of the weights are pointing downwards for the entire duration of rotation
2. The counterweight is designed so that both the center of gravity of the counterweight and the cage are at the center of rotation
3. Critical Analysis. The center of gravity of the tote is halfway between the center of rotation and the edge of the tote.
4. The gravitational constant is 9.81 m/s^2

Free Body Diagram

See figure B.3.1

Analysis

$W_T := 405\ \text{lbf}$ This is the weight of the tote

$g := 9.81\ \frac{\text{m}}{\text{s}^2}$ This is the assume gravitational constant

$L_T := 45\ \text{in}$ This is the length of the tote.

$H_T := 46\ \text{in}$ This is the height of the tote

$\theta := 0\ \text{deg}$ This is the angle where the maximum moment arm is possible

Find the moment arm of the torque

$\frac{1}{2} \cdot L_T = 22.500\ \text{in}$ $\frac{1}{2} \cdot H_T = 23.000\ \text{in}$

$$x_T := \sqrt{\left(\frac{1}{2} \cdot L_T\right)^2 + \left(\frac{1}{2} \cdot H_T\right)^2}$$

The place the farthest the center of gravity be is at one of the corners of the tote.

$$T := W_T \cdot \frac{x_T}{2} \cdot \cos(\theta)$$

$T = 736.1525\ \text{N m}$ $T = 542.958\ \text{lbf ft}$

The rest of the calculation is completed through SMath. It was found that the minimum torque to rotate the tote is 736 Nm.

Support Calculation

The next calculation to determine the forces required to hold up the tote. Again, this calculation is completed with the off center weight as explained in the minimum torque calculation. The free body diagram is shown in Figure B.3.2 for this calculation.

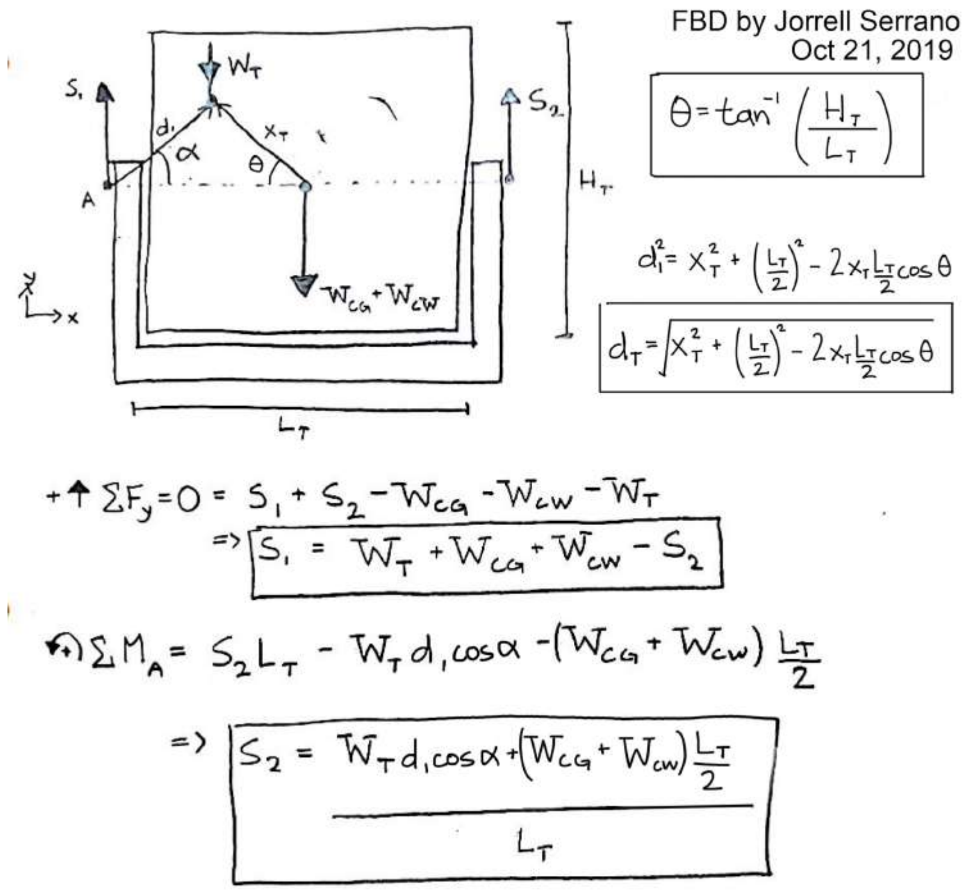


Figure B.3.2: Free body diagram of the support calculation

Where

d_1 is the moment arm from point A

L_T is the length of the tote

H_T is the height of the tote

x_T is the moment arm of the weight of the tote from the center of rotation

W_T is the weight of the tote

T is the torque

θ is the angle between the x_T and the horizontal

α is the angle between d_1 and the horizontal.

W_{CG} is the weight of the cage

W_{CW} is the weight of the counterweight

From the free body diagram, the angles of θ and α were determined. The support forces were derived.

Calculation by Jorrell Serrano Oct 21, 2019

Supporting Forces for the Tote

Objective

This calculation is for the required supporting forces from the vertical arms in order to support the tote. The weight of the tote has been assumed in the critical case scenario where there center of gravity is halfway between the center of rotation and the edge of the tote

Knowns

The weight of the tote, weight of the counterweight and the weight of the cage is known. The dimensions of the tote is known.

Assumptions

1. All of the weights are pointing downwards for the entire duration of rotation
2. The counterweight is designed so that both the center of gravity of the counterweightand the cage are at the center of rotation
3. Critical Analysis. The center of gravity of the tote is halfway between the center of rotation and the edge of the tote.
4. The gravitational constant is 9.81 m/s^2

Free Body Diagram

See figure B.3.2

Analysis

$W_T := 405\ \text{lbf}$ This is the weight of the tote.

$M_{CG} := 1417.973\ \text{kg}$

$W_{CG} := M_{CG} \cdot 9.81\ \frac{\text{N}}{\text{kg}}$ $W_{CG} = 13910.315\ \text{N}$ This is the mass & weight of the cage.

$M_{CW} := 136.101\ \text{kg}$

$W_{CW} := M_{CW} \cdot 9.81\ \frac{\text{N}}{\text{kg}}$ $W_{CW} = 1335.151\ \text{N}$ This is the mass and weight of the counterweight.

$L_T := 45\ \text{in}$ $H_T := 46\ \text{in}$ This is the length and height of the tote.

Derive the moment arm

$$x_T := \sqrt{\left(\frac{1}{2} \cdot L_T\right)^2 + \left(\frac{1}{2} \cdot H_T\right)^2}$$

This is the moment arm for the weight of the tote from the center of rotation.Critical case is when this is halfway from the center of rotation and the corner of the tote.

$$\theta := \operatorname{atan}\left(\frac{H_T}{L_T}\right) = 45.6296 \text{ deg}$$

This is the angle found between the moment arm of the weight of the tote, and the horizontal from point O. The angle is determined from the inverse tangent.

$$d_1 := \sqrt{x_T^{\,2} + \left(\frac{L_T}{2}\right)^2 - 2 \cdot x_T \cdot \frac{L_T}{2} \cdot \cos(\theta)} = 0.5842 \text{ m}$$

This is the moment arm of the weight of the tote from point A. It was determined using cosine rule.

$$\alpha := 0 \text{ deg}$$

This is the angle between the moment arm for the weight of the tote and the horizontal Critical Case when the angle is zero.

Below is the result of the equations of motion. These are the forces required to hold up the tote.

$$S_2 := \left(\frac{1}{L_T}\right) \cdot \left(\left(W_{CG} + W_{CW}\right) \cdot \frac{L_T}{2} + W_T \cdot d_1 \cdot \cos(\alpha)\right) = 8543.515 \text{ N} \qquad S_2 = 8543.515 \text{ N}$$

$$S_1 := W_T + W_{CG} + W_{CW} - S_2 = 8503.481 \text{ N} \qquad S_1 = 8503.481 \text{ N}$$

The rest of the calculation was completed in SMath. From this calculation, the two supporting forces (one for each vertical arm) is 8.54 kN, and 8.50 kN.

Rotating Shaft Calculations

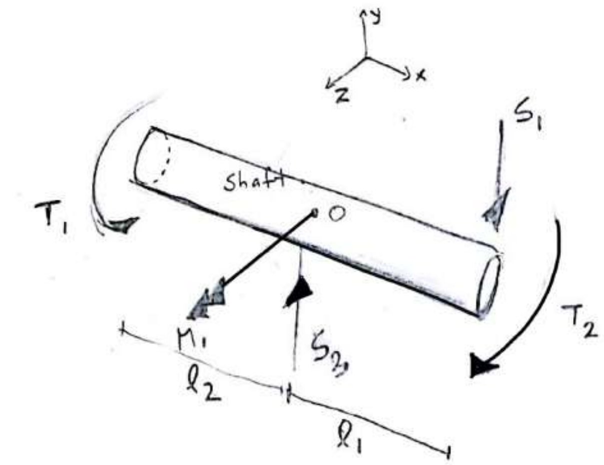
Now that the minimum torque, and support calculations have been determined, the next portion of the forklift attachment is analyzed. The next components are the rotating shafts. The outputs from the previous sections with be inputs for the rotating shaft calculations.

The key factors to determine for this analysis are the support forces from the vertical arm, as well as the stresses. The normal and bending stress must not exceed yield. Furthermore, the twist must not be too excessive.

For this analysis, the material selected is A36 steel. Therefore the stress must not exceed 250 MPa. The twist must not exceed 0.0762 degrees per inch.

The free body diagram of the shaft is in Figure B.3.3. Using the free body diagram, the supporting force and moment from the vertical arm are determined.

FBD by Jorrell Serrano
Oct 20, 2019



$$\xrightarrow{+} \sum F_x = 0 = 0$$

$$+\uparrow \sum F_y = 0 = S_3 - S_1 \Rightarrow S_1 = S_3 \quad \text{—(1)}$$

$$\sum F_z = 0 = 0$$

$$\sum (M_o)_x = 0 \quad T_1 - T_2 = 0 \Rightarrow T_1 = T_2 \quad \text{—(2)}$$

$$\sum (M_o)_y = 0$$

$$\sum (M_o)_z = M - S_1 \ell_1 = 0 \Rightarrow M_1 = S_1 \ell_1 \quad \text{—(3)}$$

$F_1 \Rightarrow$ Weight of Tote & Cage

$$\boxed{\begin{aligned} S_1 &= S_3 \\ T_1 &= T_2 \\ M_1 &= S_1 \cdot \ell_1 \end{aligned}}$$

Figure B.3. 3: Free body diagram of the shaft

Where

- T_1 is the torque from the belt drive
- S_1 is one of the supporting forces for the tote
- M_1 is the support moment from the vertical shaft
- S_3 is the support force from the vertical shaft
- T_2 is the torque required to rotate the cage, counterweight, and tote

Calculations are done twice. Once for each rotating shaft. A minimum radius is determined to withstand twisting requirements. A minimum radius is also determined to withstand bending requirements.

Rotating Shaft Calculations - Shaft 1

Calculation - Jorrell
Oct 22, 2019

Objective

This calculation is for the forces in the rotating shaft. The rotating shaft is required to withstand the torque to rotate the tote, as well as to support the tote.

Knowns

Support Forces are previously calculated from the Tote Calculations.
Torque already known from torque calculations

Assumptions

1. The shaft is a perfect cylinder
2. The shaft has even density
3. The support forces from the vertical arm are point loads and point moments
4. The support forces from the vertical arm are in the exact middle of the of the shaft
5. The weight of the shaft is neglected. It is small compared to the forces

Free Body Diagram

See figure xxx

Analysis

$S_1 := 8543.515\ N$ This is the supporting force as calculated in "Supporting Forces for the Tote"

$l_1 := 95.25\ mm$ Half the length of rotating shaft

$T_1 := 736.1525\ N\,m$ Minimum torque to rotate tote. This is from the "Minimum Torque Calculation"

$r_1 := 2.2123\ cm$ This is the radius of the shaft

$A_1 := \pi \cdot r_1^{\,2} = 1537.5807\ mm^2$ This is the cross sectional area of the shaft

$I := \frac{1}{4} \cdot \pi \cdot r_1^{\,4}$ This is the moment of inertia of the cross-sectional about the neutral axis.

$J := \frac{\pi}{2} \cdot r_1^{\,4} = 3.7627 \cdot 10^{-7}\ m^4$

$G := 75\ GPa$ Structural Steel A36 Modulus of Rigidity

Equations of Motion

$T_2 := T_1 = 736.153\ \text{N m}$

$S_3 := S_1 = 8543.515\ \text{N}$ This is the support force from vertical arm

$M_1 := S_1 \cdot l_1 = 813.770\ \text{N m}$ $M_1 = 600.206\ \text{lbf ft}$ This is support moment from the vertical arm

Singularity Function

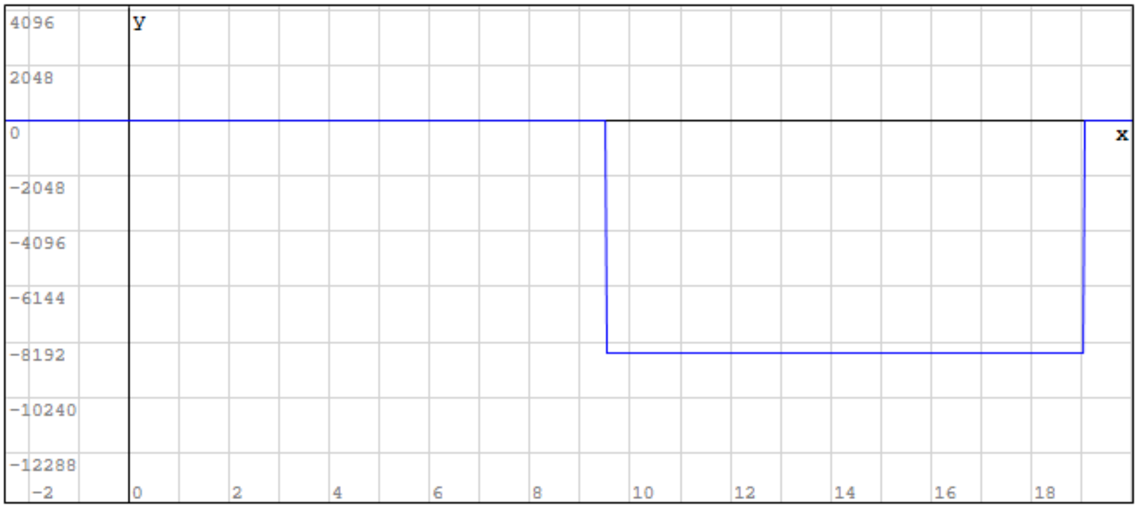
$$S(z, a, n) := \text{if } ((z - a) > 0) \wedge (n \geq 0)$$
$$(z - a)^n$$
$$\text{else}$$
$$0$$

Shear and Bending Moment Functions

$$w_x(x) := S_3 \cdot S(x, l_1, -1) - M_1 \cdot S(x, l_1, -2) - S_1 \cdot S(x, 2 \cdot l_1, -1)$$

$$V_x(x) := (-S_3) \cdot S(x, l_1, 0) - M_1 \cdot S(x, l_1, -1) + S_1 \cdot S(x, 2 \cdot l_1, 0)$$

$$M_x(x) := (S_3) \cdot S(x, l_1, 1) - (-M_1) \cdot S(x, l_1, 0) - (-S_1) \cdot S(x, 2 \cdot l_1, 1)$$



$(V_x(x\ \text{cm}))$

Algorithm to calculate Minimum Radius due to Twist

$\left(\varphi_{twist}\right) := 7 \frac{deg}{in}$

$r_{twist} := 0.05\ in$

while $\left(\varphi_{twist} > 0.0762 \frac{deg}{in}\right)$

$$\left| \begin{array}{l} r_{twist} := r_{twist} + 0.001\ in \\ J_{twist} := \frac{\pi}{2} \cdot r_{twist}^{\ 4} \\ \varphi_{twist} := \frac{T_1 \cdot 2}{J_{twist} \cdot G} \end{array} \right.$$

$r_{twist} = 2.2123\ cm$ $\varphi_{twist} = 0.0759 \frac{deg}{in}$

Algorithm to calculate Minimum Radius due to Bending

$r_{bend} := 0.01\ mm$

$A_{bend} := \pi \cdot r_{bend}^{\ 2} = 0.0003\ mm^2$

$I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4}$

$\sigma_{bend} := M_x (19.5\ cm) \cdot \frac{r_{bend}}{I_{bend}} = 2.1701 \cdot 10^{12}\ MPa$ $\sigma_{bendmax} := 250\ MPa$ This is the tensile yield

while $\left(\sigma_{bend} > \sigma_{bendmax}\right)$

$$\left| \begin{array}{l} r_{bend} := r_{bend} + 0.001\ in \\ A_{bend} := \pi \cdot r_{bend}^{\ 2} \\ I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4} \\ \sigma_{bend} := M_x (19.5\ cm) \cdot \frac{r_{bend}}{I_{bend}} \end{array} \right.$$

$r_{bend} = 2.0559\ cm$

$\sigma_{bend} = 249.752\ MPa$

From the shaft 1 calculations, the vertical arm is required to support 814 Nm, and 8.54 kN. To avoid twist, the minimum radius required is 2.21 cm. To avoid bending, the minimum radius is 2.06 cm.

Calculation by Jorrell Serrano Oct 22, 2019

Rotating Shaft Calculations - Shaft 2

Objective

This calculation is for the forces in the rotating shaft. The rotating shaft is required to withstand the torque to rotate the tote, as well as to support the tote.

Knowns

Support Forces are previously calculated from the Tote Calculations.
Torque already known from torque calculations

Assumptions

1. The shaft is a perfect cylinder
2. The shaft has uniform density
3. The support forces from the vertical arm are point loads and point moments
4. The support forces from the vertical arm are in the exact middle of the of the shaft
5. The weight of the shaft is neglected. It is small compared to the forces

Free Body Diagram

See figure xxx

Analysis

$S_1 := 8503.481\ \mathrm{N}$ This is the supporting force as calculated in "Supporting Forces for the Tote"

$l_1 := 95.25\ \mathrm{mm}$ Half the length of rotating shaft

$T_1 := 736.1525\ \mathrm{N\,m}$ Minimum torque to rotate tote. This is from the "Minimum Torque Calculation"

$r_1 := 2.2103\ \mathrm{cm}$ This is the radius of the shaft

$A_1 := \pi \cdot {r_1}^2 = 15.348\ \mathrm{cm}^2$ This is the cross sectional area of the shaft

$I := \frac{1}{4} \cdot \pi \cdot {r_1}^4$ This is the moment of inertia of the cross-sectional about the neutral axis.

$J := \frac{\pi}{2} \cdot {r_1}^4 = 3.7491 \cdot 10^{-7}\ \mathrm{m}^4$

$G := 75\ \mathrm{GPa}$ Structural Steel A36 Modulus of Rigidity

Equations of Motion

$T_2 := T_1 = 736.153\ \mathrm{N\,m}$

$S_3 := S_1 = 8503.481\ \mathrm{N}$ This is the support force from vertical arm

$M_1 := S_1 \cdot l_1 = 809.957\ \mathrm{N\,m}$ $M_1 = 597.393\ \mathrm{lbf\,ft}$ This is support moment from the vertical arm

Singularity Function

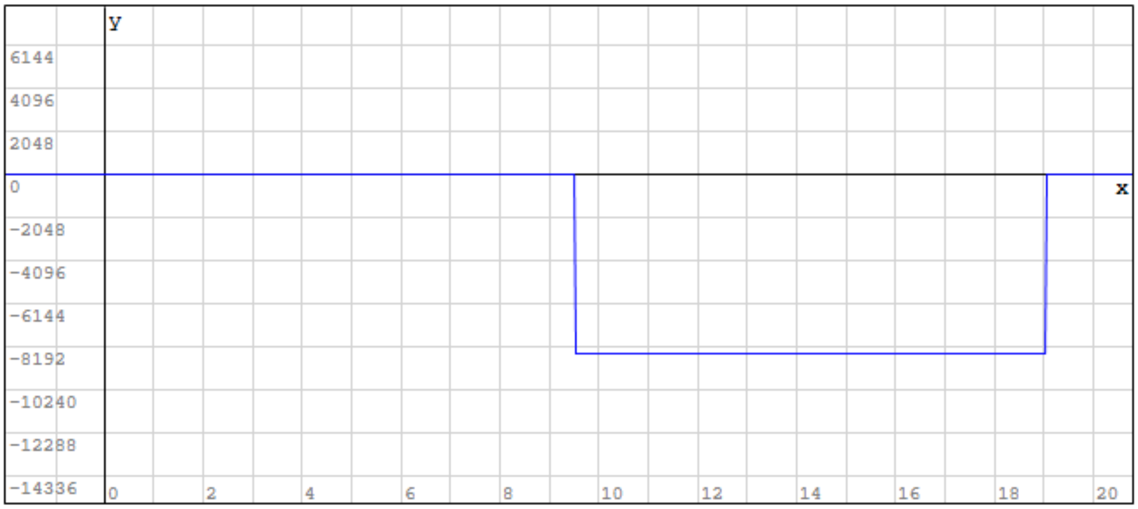
$$S(z, a, n) := \text{if } ((z-a) > 0) \wedge (n \geq 0) \quad (z-a)^n \quad \text{else} \quad 0$$

Shear and Bending Moment Functions

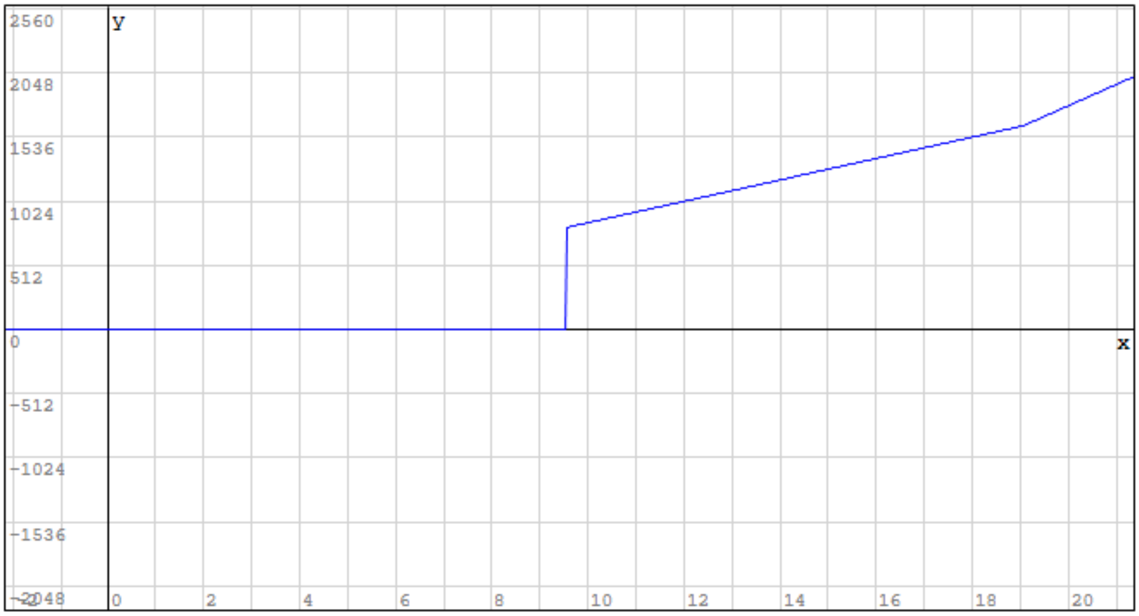
$$w_x(x) := S_3 \cdot S(x, l_1, -1) - M_1 \cdot S(x, l_1, -2) - S_1 \cdot S(x, 2 \cdot l_1, -1)$$

$$V_x(x) := (-S_3) \cdot S(x, l_1, 0) - M_1 \cdot S(x, l_1, -1) + S_1 \cdot S(x, 2 \cdot l_1, 0)$$

$$M_x(x) := (S_3) \cdot S(x, l_1, 1) - (-M_1) \cdot S(x, l_1, 0) - (-S_1) \cdot S(x, 2 \cdot l_1, 1)$$



$(V_x(x\ \mathrm{cm}))$



$M_x(x\ cm)$

Maximum Stress

$\sigma := M_x(19.5\ cm) \cdot \frac{r_1}{I} = 200.0304\ MPa$ Maximum normal stress from bending

$\tau := \frac{V_x(12\ cm)}{A_1} = -5.5404\ MPa$ Maximum Shear stress

Twist

$\varphi_1 := \frac{T_1 \cdot 2}{J \cdot G} = 0.0762\ \frac{deg}{in}$ Maximum twist should be 0.0762 °/inch

Algorithm to calculate Minimum Radius due to Twist

$$(\varphi_{twist}) := 7\ \frac{deg}{in}$$

$$r_{twist} := 0.05\ in$$

while $\left(\varphi_{twist} > 0.0762\ \frac{deg}{in}\right)$

$$r_{twist} := r_{twist} + 0.001\ in$$

$$J_{twist} := \frac{\pi}{2} \cdot r_{twist}^{\ 4}$$

$$\varphi_{twist} := \frac{T_1 \cdot 2}{J_{twist} \cdot G}$$

$$r_{twist} = 2.2123\ cm \qquad \varphi_{twist} = 0.0759\ \frac{deg}{in}$$

Algorithm to calculate Minimum Radius due to Bending

$$r_{bend} := 0.01\ mm$$

$$A_{bend} := \pi \cdot r_{bend}^{\ 2} = 3.1416 \cdot 10^{-10}\ m^2$$

$$I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4}$$

$$\sigma_{bend} := M_x(19.5\ cm) \cdot \frac{r_{bend}}{I_{bend}} = 2.16 \cdot 10^{18}\ Pa \qquad \sigma_{bendmax} := 250\ MPa$$ This is the tensile yield

while $(\sigma_{bend} > \sigma_{bendmax})$

$$r_{bend} := r_{bend} + 0.001\ in$$

$$A_{bend} := \pi \cdot r_{bend}^{\ 2}$$

$$I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4}$$

$$\sigma_{bend} := M_x(2 \cdot l_1) \cdot \frac{r_{bend}}{I_{bend}}$$

$$r_{bend} = 2.0228\ cm$$

From the shaft 2 calculations, the vertical arm is required to support 597 Nm, and 8.54 kN. To avoid twist, the minimum radius required is 2.21 cm. To avoid bending, the minimum radius is 2.02 cm.

Vertical Arm Calculations

From the previous section, the supporting force and moment that is required for the vertical arms are calculated. The analysis of the vertical arms is to determine the required moment that the top shaft needs to support, the minimum cross-sectional area, and the shear stress where the vertical arm connects to the sleeves. The normal stress from the support forces in combination with the bending moment must not exceed yield. Again, the material used for this calculation is A36 steel.

The key factors to determine for this analysis are the support forces from the vertical arm, as well as the stresses. The normal and bending stress must not exceed yield. Furthermore, the twist must not be too excessive.

The free body diagram of the shaft is in Figure B.3.4. Using the free body diagram, the supporting force and moment from the vertical arm are determined. Additionally, the critical cross-sectional area is derived.

Similar to the rotating shaft, calculations will be required for both vertical arms.

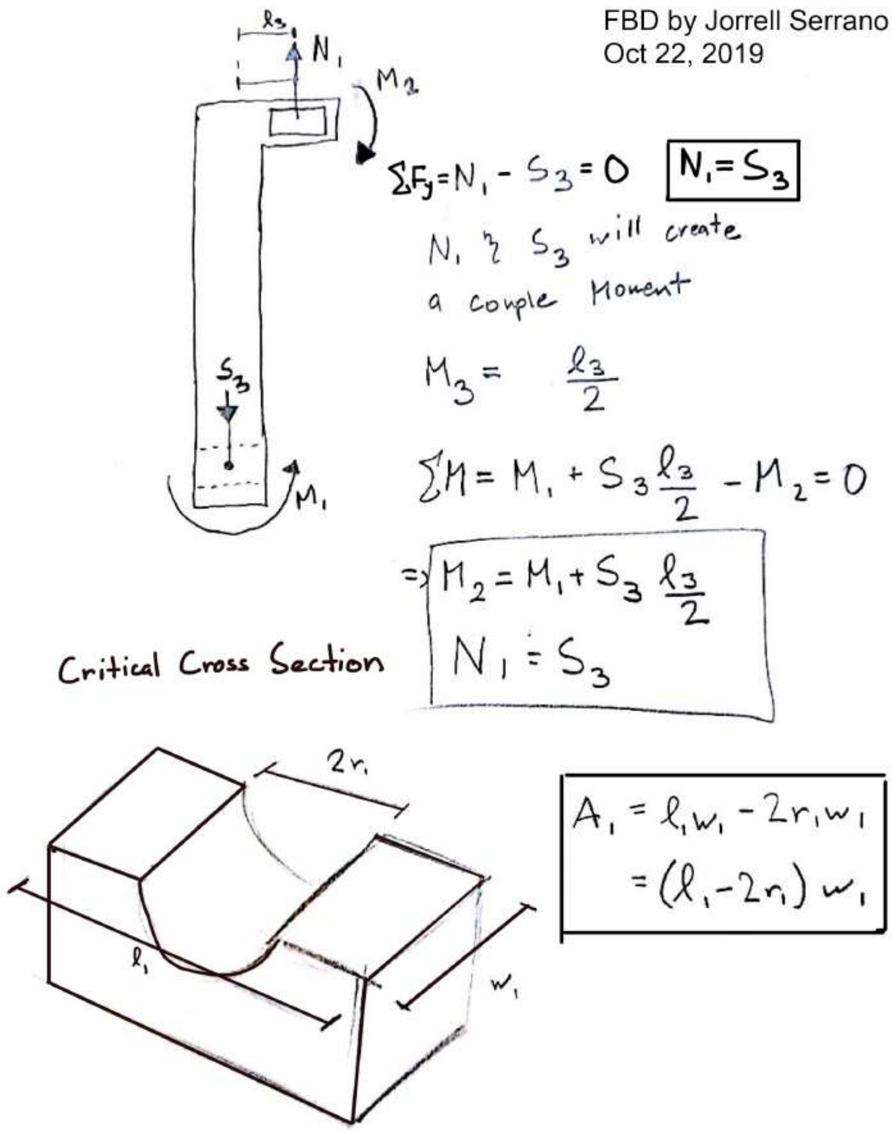


Figure B.3.4: Free body diagram of the vertical arm

Where

- l_3 is the horizontal distance between the N_1 and S_3
- N_1 is the support force from the forklift
- M_2 is the support moment from the top shafts
- S_3 is the support from from the rotating shaft
- M_1 is the support moment from rotating shaft

r_1 is the radius of the rotating shaft

w_1 is the width of the critical cross sectional area

l_1 is the length of the critical cross sectional area

Calculation by Jorrell Serrano Oct 23, 2019

Vertical Arm Calculations - Vertical Arm 1

Objective

Determine the required dimensions of the arm bar bracket in order to withstand the forces and and stresses. The critical areas of the vertical arm bar is the cross sectional area at the hole for the rotating shaft, and the cross sectional area where it attaches to the fork sleeves. The normal stress in the first area must not exceed yield. The shear stress in the second area must not exceed yield. The calculation is assumed to be in the critical case scenario where the vertical must hold the a full tote weight of 7000 lbs. The weight of 7000 lbs is with a safety factor of 2.

Knowns

The height and thickness of the vertical bar is known. The weight and moment from the rotating shaft calculation is known.

Assumptions

The vertical arm is assumed to be a rectangular shape

The density of the vertical arm is uniform

The cross section is assumed to be symmetrical about the neutral axis

The gravitational accelerations is 9.81 N/kg

The tote is assumed to be 7000lbs with a safety factor of 2 on the weight

The self weight of the vertical arm is assumed relatively small and neglected in the analysis

The assumed material is structural A36 Steel

Free Body Diagram

See figure B.3.4

Analysis

Variables

$M_1 := 736.1525\ \mathrm{N\,m}$ Support moment

$S_3 := 8543.515\ \mathrm{N}$ Support Force

$l_3 := 114.25\ \mathrm{mm}$ Distance between support force from forklift and support force

$r_1 := 2.2103\ \mathrm{cm}$ radius of hole cutout

$w_1 := 2\ \mathrm{cm}$ width of arm

$l_1 := 53.7732\ \mathrm{mm}$ length of arm. This can be changed to match the stress

Equations of Motion

$M_2 := M_1 + S_3 \cdot \frac{l_3}{2} = 1224.201\ \mathrm{N\,m}$ Moment at the top of the vertical arm. To be counteracted by the top shafts. Calculated in the top shaft calculation

$N_1 := S_3 = 8543.515\ \mathrm{N}$ Support Force from Forklift

Stress Analysis

$E := 200\ \mathrm{GPa}$
$G := 75\ \mathrm{GPa}$
$\nu := 0.32$

These are the material properties of A36 steel. The Modulus of elasticity E = 200 GPa. The modulus of rigidity is 75 GPa, and Poisson's ratio is 0.32

$\sigma_{yield} := 250\ \mathrm{MPa}$ This is the tensile yield of A36 Steel

$A_1 := \left(l_1 - 2 \cdot r_1\right) \cdot w_1 = 191.344\ \mathrm{mm}^2$ Area where there is the hole cutout. Critical Area

$A_2 := 9779\ \mathrm{mm}^2$ This is the area at the connection to the sleeves.

$I := \frac{l_1 \cdot w_1^3}{12} = 35848.8\ \mathrm{mm}^4$ The is the moment of inertia about the neutral axis

$\sigma := \frac{S_3}{A_1} + \frac{M_1 \cdot \frac{w_1}{2}}{I} = 249.9993\ \mathrm{MPa}$ This is the tensile stress of the vertical arm at the critical area. This stress accounts for the normal stress for the weight plus the normal stress from the bending moment.

$\tau := \frac{S_3}{A_2} = 0.8737\ \mathrm{MPa}$ This is the shear stress at the connection to the sleeves.

Minimum Area Calculation

$l_{calc} := 2 \cdot r_1 + 9.4\ \mathrm{mm}$

$A_{calc} := \left(l_{calc} - 2 \cdot r_1\right) \cdot w_1 = 1.88\ \mathrm{cm}^2$

$I_{calc} := \frac{l_{calc} \cdot w_1^3}{12} = 35737.3333\ \mathrm{mm}^4$

$\sigma_{calc} := \frac{S_3}{A_{calc}} + \frac{M_1 \cdot \frac{w_1}{2}}{I_{calc}} = 251.434\ \mathrm{MPa}$

while $\sigma_{calc} > \sigma_{yield}$

$$\left| \begin{aligned} & l_{calc} := l_{calc} + 0.0001\,\text{mm} \\ & A_{calc} := \left(l_{calc} - 2 \cdot r_1\right) \cdot w_1 \\ & I_{calc} := \frac{l_{calc} \cdot w_1^{\,3}}{12} \\ & \sigma_{calc} := \frac{S_3}{A_{calc}} + \frac{M_1 \cdot \frac{w_1}{2}}{I_{calc}} \end{aligned} \right.$$

$l_{calc} = 53.7732\,\text{mm}$ $\quad A_{calc} = 1.9134\,\text{cm}^2$ $\quad \sigma_{calc} = 2.5 \cdot 10^8\,\text{Pa}$

From the first vertical arm, the top shaft will need to support a moment of 54 kNm. The minimum area requirement is 1.91 $\text{cm}^{2.}$

Calculation by Jorrell Serrano Oct 23, 2019

Vertical Arm Calculations - Vertical Arm 2

Objective

Determine the required dimensions of the arm bar bracket in order to withstand the forces and and stresses. The critical areas of the vertical arm bar is the cross sectional area at the hole for the rotating shaft, and the cross sectional area where it attaches to the fork sleeves. The normal stress in the first area must not exceed yield. The shear stress in the second area must not exceed yield. The calculation is assumed to be in the critical case scenario where the vertical must hold the a full tote weight of 7000 lbs. The weight of 7000 lbs is with a safety factor of 2.

Knowns

The height and thickness of the vertical bar is known. The weight and moment from the rotating shaft calculation is known.

Assumptions

The vertical arm is assumed to be a rectangular shape

The density of the vertical arm is uniform

The cross section is assumed to be symmetrical about the neutral axis

The gravitational accelerations is 9.81 N/kg

The tote is assumed to be 7000lbs with a safety factor of 2 on the weight

The self weight of the vertical arm is assumed relatively small and neglected in the analysis

The assumed material is structural A36 Steel

Free Body Diagram

See figure B.3.4

Analysis

Variables

$M_1 := 597.393\ \text{N m}$	Support moment
$S_3 := 8503.481\ \text{N}$	Support Force
$l_3 := 50.1945\ \text{mm}$	Distance between support force from forklift and support force
$r_1 := 2.2103\ \text{cm}$	radius of hole cutout
$w_1 := 2\ \text{cm}$	width of arm
$l_1 := 50.164\ \text{mm}$	length of arm. This can be changed to match the stress

Equations of Motion

$M_2 := M_1 + S_3 \cdot \frac{l_3}{2} = 810.807\ \mathrm{N\,m}$ Moment at the top of the vertical arm. To be counteracted by the top shafts. Calculated in the top shaft calculation

$N_1 := S_3 = 8503.481\ \mathrm{N}$ Support Force from Forklift

Stress Analysis

$E := 200\ \mathrm{GPa}$
$G := 75\ \mathrm{GPa}$
$\nu := 0.32$

These are the material properties of A36 steel. The Modulus of elasticity E = 200 GPa. The modulus of rigidity is 75 GPa, and Poisson's ratio is 0.32

$\sigma_{yield} := 250\ \mathrm{MPa}$ This is the tensile yield of A36 Steel

$A_1 := \left(l_1 - 2 \cdot r_1\right) \cdot w_1 = 119.160\ \mathrm{mm}^2$ Area where there is the hole cutout. Critical Area

$A_2 := 9779\ \mathrm{mm}^2$ This is the area at the connection to the sleeves.

$I := \frac{l_1 \cdot w_1^{\,3}}{12} = 33442.6667\ \mathrm{mm}^4$ The is the moment of inertia about the neutral axis

$\sigma := \frac{S_3}{A_1} + \frac{M_1 \cdot \frac{w_1}{2}}{I} = 249.9939\ \mathrm{MPa}$ This is the tensile stress of the vertical arm at the critical area. This stress accounts for the normal stress for the weight plus the normal stress from the bending moment.

$\tau := \frac{S_3}{A_2} = 0.8696\ \mathrm{MPa}$ This is the shear stress at the connection to the sleeves.

Minimum Area Calculation

$l_{calc} := 2 \cdot r_1 + 1\ \mathrm{mm}$

$A_{calc} := \left(l_{calc} - 2 \cdot r_1\right) \cdot w_1 = 2 \cdot 10^{-5}\ \mathrm{m}^2$

$I_{calc} := \frac{l_{calc} \cdot w_1^{\,3}}{12} = 30137.3333\ \mathrm{mm}^4$

$\sigma_{calc} := \frac{S_3}{A_{calc}} + \frac{M_1 \cdot \frac{w_1}{2}}{I_{calc}} = 623.3976\ \mathrm{MPa}$

$$\text{while } \sigma_{calc} > \sigma_{yield}$$
$$\left| \begin{array}{l} l_{calc} := l_{calc} + 0.001\ mm \\ A_{calc} := \left(l_{calc} - 2 \cdot r_1\right) \cdot w_1 \\ I_{calc} := \dfrac{l_{calc} \cdot w_1^{\ 3}}{12} \\ \sigma_{calc} := \dfrac{S_3}{A_{calc}} + \dfrac{M_1 \cdot \dfrac{w_1}{2}}{I_{calc}} \end{array} \right.$$

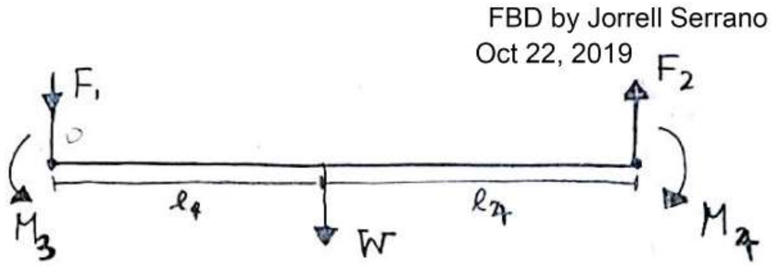
$$l_{calc} = 50.164\ mm$$

From the second vertical arm, the top shaft will need to support a moment of 846 Nm. The minimum area requirement is 1.91 cm^2.

Top Shaft Calculations

In the previous section, the required moment that the top shaft is required to support was determined. This calculation is to determine the minimum area required to withstand that moment. The fact that there are two shafts is taken into consideration in this calculation.

The free body diagram is in Figure B.3.5. The support forces are derived from the free body diagram.



$$\Sigma F_y = -F_1 + F_2 - W$$

$$\Sigma M_0 = -F_2 2\ell_1 - W\ell_1 + M_1 - M_2 = 0$$

$$\boxed{F_2 = \frac{(W\ell_4 + M_2 - M_1)}{2\ell_1} \qquad F_1 = F_2 + W}$$

Figure B.3.5: Free body diagram of the top shaft

Where

F_1 is the left support force

F_2 is the right support force

M_3 is the moment for the first vertical arm

M_4 is the moment for the second vertical arm

W is the self-weight of the shaft

CONSULTING

Calculation by Jorrell Serrano Oct 24, 2019

Top Shaft Calculations

Objective

Determine the required dimensions of the top shaft in order to withstand the forces and and stresses. The shear and bending moment in the shaft should not exceed yield.

Knowns

Both moments on each side of the shaft has been determined by the tote calculations, shaft calculations, and vertical arm calculations

Assumptions

The shaft is a perfect cylinder

The shaft has uniform density

The shaft is connected to the vertical arms on the ends of the shaft

The gravitational accelerations is 9.81 N/kg

Free Body Diagram

See figure B.3.5

Analysis

$M_3 := \frac{1224.201}{2}\ \mathrm{N\,m}$ This is moment from one of the ends. It is divided by 2 for two shafts.

$M_4 := \frac{846.009}{2}\ \mathrm{N\,m}$ This is the moment from the other end. It is divided by 2 for two shafts.

$l_4 := 489.625\ \mathrm{mm}$ This is half the length of the shaft

$\rho := 7.85\ \frac{\mathrm{Mg}}{\mathrm{m}^3}$ $\quad r_2 := 15.97\ \mathrm{mm}$ $\quad g := 9.81\ \frac{\mathrm{N}}{\mathrm{kg}}$

$A_2 := \pi \cdot r_2^{\,2} = 8.0123\ \mathrm{cm}^2$ $\quad I := \frac{1}{4} \cdot \pi \cdot r_2^{\,4}$ $\quad J := \frac{\pi}{2} \cdot r_2^{\,4} = 1.0217 \cdot 10^{-7}\ \mathrm{m}^4$

$W_s := 2 \cdot l_4 \cdot A_2 \cdot \rho \cdot g = 60.4216\ \mathrm{N}$ This is the weight of the shaft

Equations of Motion

$$F_2 := \frac{W_s \cdot l_4 + M_3 - M_4}{2 \cdot l_4} = 223.3137\ \mathrm{N}$$

$F_1 := F_2 - W_s = 162.8921\ \mathrm{N}$

Singularity Function

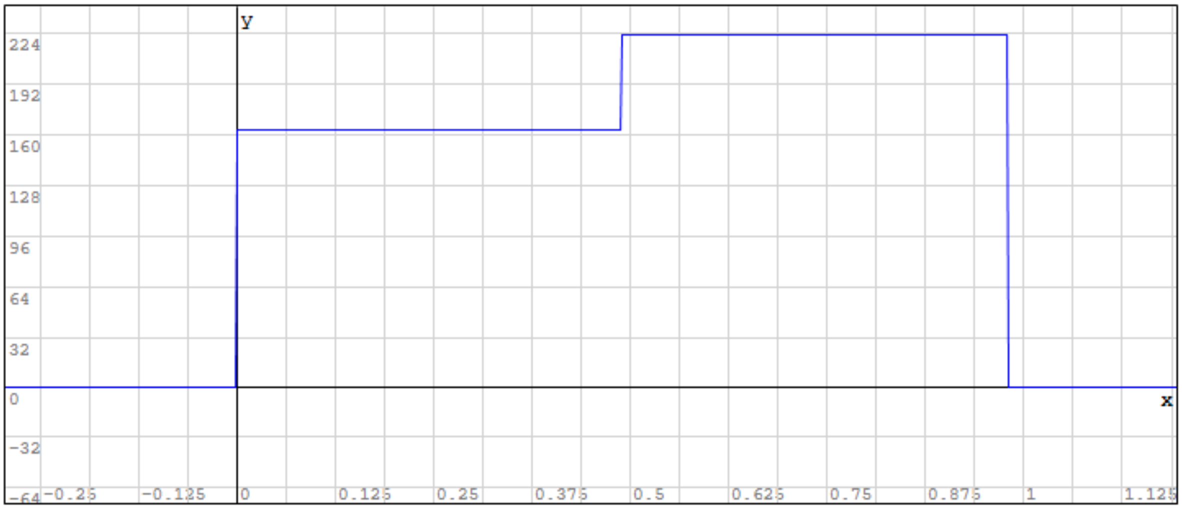
$$S(z, a, n) := \text{if } \left((z - a) > 0\right) \wedge (n \geq 0) \quad (z-a)^n \quad \text{else} \quad 0$$

Shear and Bending Moment Diagrams

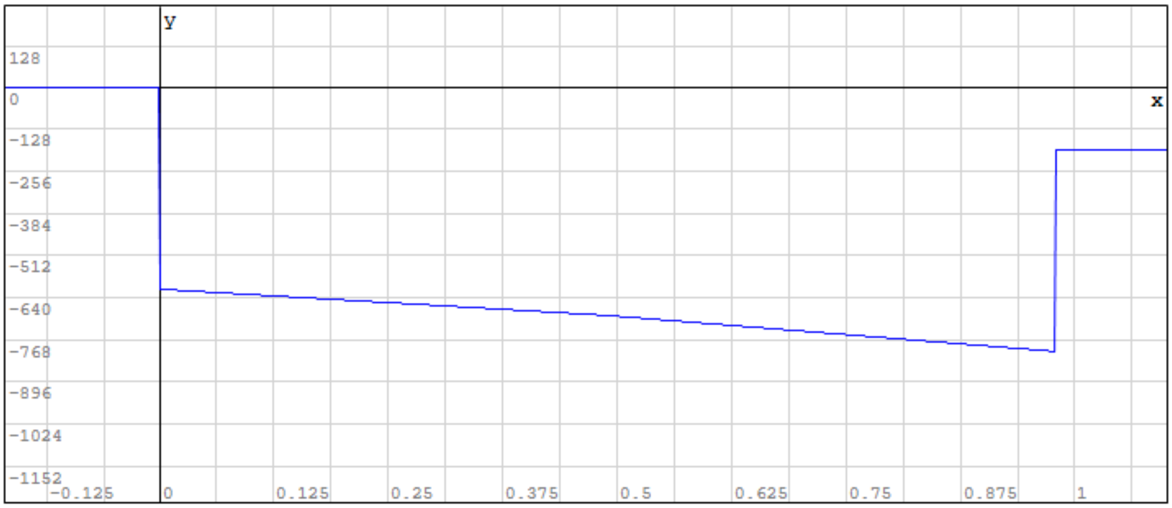
$$w(x) := -M_3 \cdot S(x, 0, -2) - F_1 \cdot S(x, 0, -1) - W_s \cdot S\left(x, l_4, -1\right) + F_2 \cdot S\left(x, 2 \cdot l_4, -1\right) + M_3 \cdot S\left(x, 2 \cdot l_4, -2\right)$$

$$V(x) := -\left(-M_3 \cdot S(x, 0, -1) - F_1 \cdot S(x, 0, 0) - W_s \cdot S\left(x, l_4, 0\right) + F_2 \cdot S\left(x, 2 \cdot l_4, 0\right) + M_3 \cdot S\left(x, 2 \cdot l_4, -1\right)\right)$$

$$M(x) := -M_3 \cdot S(x, 0, 0) - F_1 \cdot S(x, 0, 1) - W_s \cdot S\left(x, l_4, 1\right) + F_2 \cdot S\left(x, 2 \cdot l_4, 1\right) + M_3 \cdot S\left(x, 2 \cdot l_4, 0\right)$$



$V(x\ m)$



$M(x\ m)$

Maximum Stress

$$\sigma := M(979.25\ \mathrm{mm}) \cdot \frac{r_2}{I} = -250.4577\ \mathrm{MPa}$$

$$\tau := \frac{V(0.75\ \mathrm{m})}{A_2} = 0.2787\ \mathrm{MPa}$$

Algorithm to calculate Minimum Radius due to Bending

$$r_{bend} := 0.01\ \mathrm{mm}$$

$$A_{bend} := \pi \cdot r_{bend}^{\ 2} = 3.1416 \cdot 10^{-10}\ \mathrm{m}^2$$

$$I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4}$$

$\sigma_{bendmax} := 250\ \mathrm{MPa}$ This is the tensile yield

$$\sigma_{bend} := M(979.25\ \mathrm{mm}) \cdot \frac{r_{bend}}{I_{bend}} = -1.0201 \cdot 10^{12}\ \mathrm{MPa}$$

$$\text{while } \left(\left|\sigma_{bend}\right| > \sigma_{bendmax}\right)$$

$$\left| \begin{array}{l} r_{bend} := r_{bend} + 0.001\ \mathrm{mm} \\ A_{bend} := \pi \cdot r_{bend}^{\ 2} \\ I_{bend} := \frac{1}{4} \cdot \pi \cdot r_{bend}^{\ 4} \\ \sigma_{bend} := M(979.25\ \mathrm{mm}) \cdot \frac{r_{bend}}{I_{bend}} \end{array} \right.$$

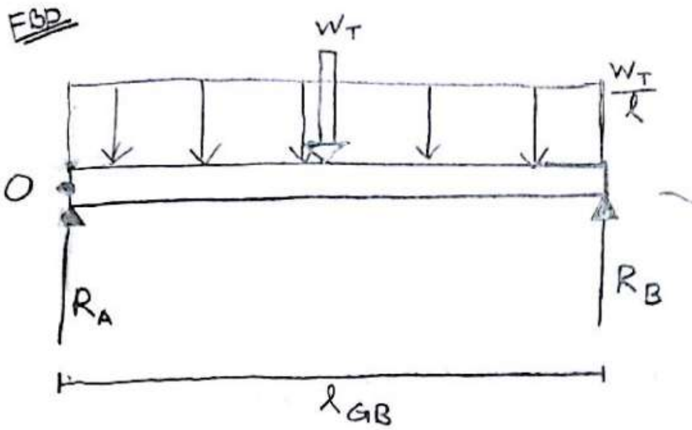
$r_{bend} = 15.98\ \mathrm{mm}$ $A_{bend} = 802.2384\ \mathrm{mm}^2$ $\sigma_{bend} = -2.4999 \cdot 10^{8}\ \mathrm{Pa}$

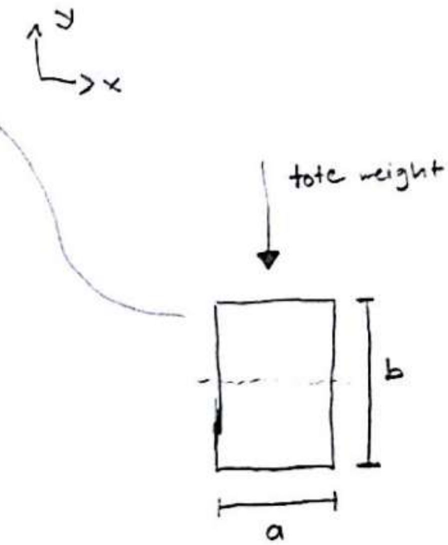
From the calculation, the minimum radius of the top shaft is 1.60 cm.

Guard Bracket Calculations

The purpose of the guard bracket is to support the tote in the event of a worst case scenario. The guard bracket is for safety purposes, and will not let the tote fall out of the cage. The free body diagram is in Figure B.3.6.

FBD by Jorrell Serrano
Oct 27, 2019





$$\uparrow \Sigma F_y = 0 = -W_T + R_A + R_B$$

$$W_T = R_A + R_B$$

$$\circlearrowleft + \Sigma M_o = 0 = R_B l_{GB} - W_T l_{GB} 0.5$$

so $$R_B = \frac{W_T l_{GB} 0.5}{l_{GB}}$$

$$\Rightarrow \boxed{R_A = \frac{W_T}{2}} \qquad \Rightarrow \boxed{R_B = \frac{W_T}{2}}$$

Figure B.3. 6: Free body diagram of the guard bracket

Where

W_T is the weight of the tote

R_A is the left support force

R_B is the right support force

l_{GB} is the length of the guard bracket

APIS CONSULTING

Calculation by Jorrell Serrano Oct 28, 2019

Guard Bracket Calculations

Objective

Determine the required dimensions of the guard bracket in order to withstand the forces and and stresses. The shear and bending moment in the shaft should not exceed yield. The guard bracket calculation is assumed to be in the critical case scenario where the guard bracket is directly under the full tote weight of 7000 lbs.

Knowns

The length and cross sectional dimensions of the guard bracket is known. The material of the guard bracket is known. The weight of the tote is known.

Assumptions

The guard bracket is assumed to be a rectangular shape

The density of the guard bracket is uniform

The cross section is assumed to be symmetrical about the neutral axis

The guard bracket is connected to the cage at the ends of the bracket.

The gravitational accelerations is 9.81 N/kg

The tote is assumed to be a constant distributed load

The self weight of the guard bracket is assumed relatively small and neglected in the analysis

The assumed material is structural A36 Steel

Free Body Diagram

See figure B.3.6

Analysis

$W_T := 405\ \text{lbf}$ This is the weight of the tote.

$l_{GB} := 1200.15\ \text{mm}$ This is the length of the guard bracket

$a_{GB} := 0.5\ \text{in}$ $b_{GB} := 0.8897\ \text{in}$ This is the width and height of the cross section of the guard bracket respectively

$\rho := 7.85\ \frac{\text{Mg}}{\text{m}^3}$ This is the density of A36 Steel $g := 9.81\ \frac{\text{N}}{\text{kg}}$ This is the graviational acceleration.

$A_{GB} := a_{GB} \cdot b_{GB} = 2.87\ \text{cm}^2$ This is the cross sectional area of the guard bracket

$I := \frac{a_{GB} \cdot b_{GB}^{\ 3}}{12} = 12213.901\ \text{mm}^4$ This is the moment of inertia of a rectangular cross sectional area about the neutral axis.

Equations of Motion

$R_A := \frac{W_T}{2} = 900.765\ N$ $R_B := R_A = 900.765\ N$

These are the reation forces. A is on the left side. B is on the right side.

Singularity Function

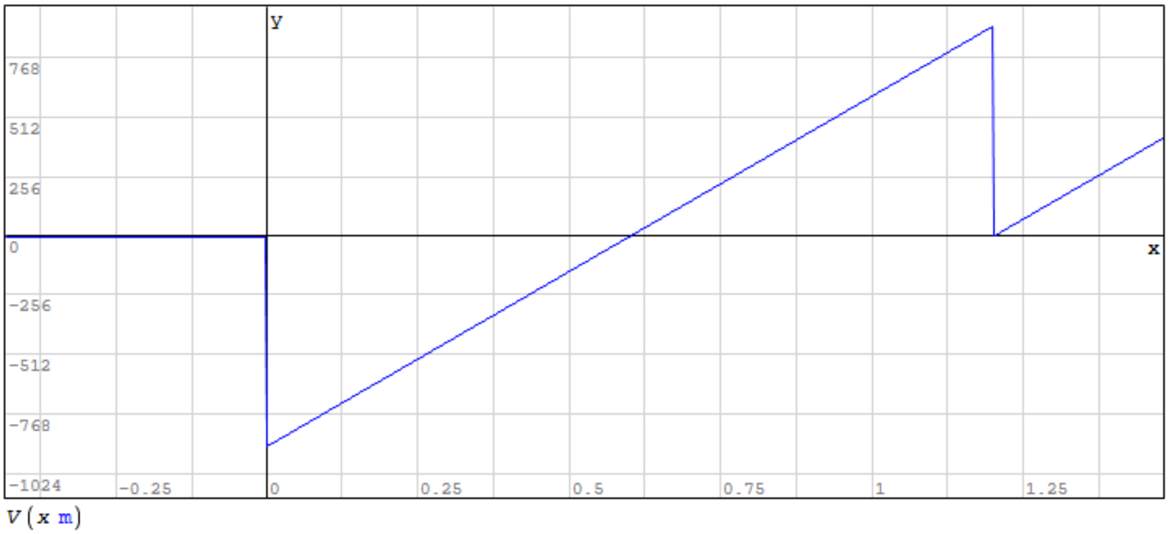
$$S(z, a, n) := \text{if } \left(\left((z-a)>0\right) \wedge (n \geq 0)\right) \quad (z-a)^n \quad \text{else} \quad 0$$

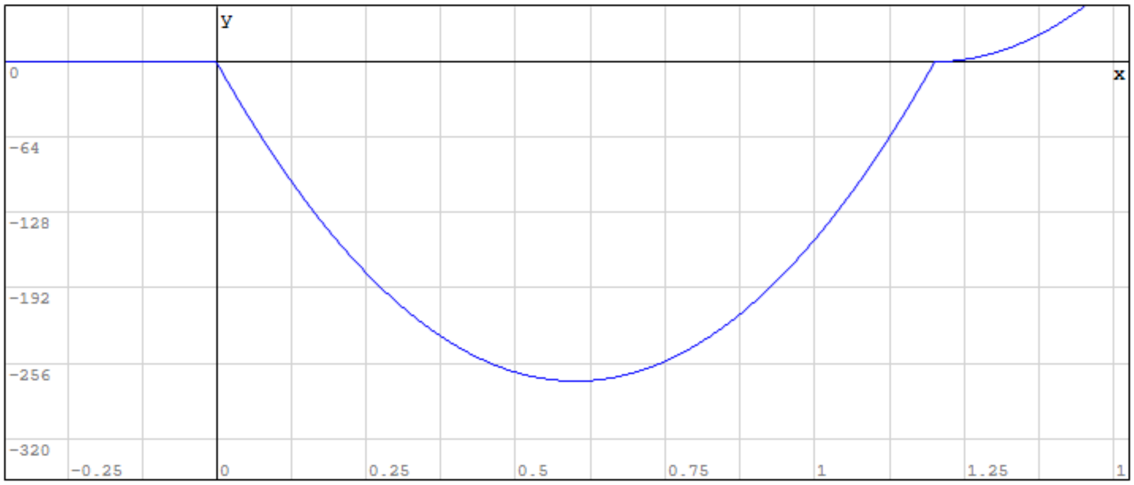
Shear and Bending Moment Diagrams

$$w(x) := R_A \cdot S(x, 0, -1) - \frac{W_T}{l_{GB}} \cdot S(x, 0, 0) + R_B \cdot S\left(x, l_{GB}, -1\right)$$

$$V(x) := -\left(R_A \cdot S(x, 0, 0) - \frac{W_T}{l_{GB}} \cdot S(x, 0, 1) + R_B \cdot S\left(x, l_{GB}, 0\right)\right)$$

$$M(x) := -\left(R_A \cdot S(x, 0, 1) - \frac{W_T}{2 \cdot l_{GB}} \cdot S(x, 0, 2) + R_B \cdot S\left(x, l_{GB}, 1\right)\right)$$





$M(x\ m)$

Maximum Stress

$$\sigma := M\left(\frac{l_{GB}}{2}\right) \cdot \frac{b_{GB} \cdot 0.5}{I} = -250.023\ MPa$$

$$\tau := \frac{V(0.00000000000001\ in)}{A_{GB}} = -3.1386\ MPa$$

Algorithm to find the minimum area needed for bending moment

The width will be constrained. Use 1 in width $a_{bend} := 0.5\ in$

$b_{bend} := 1\ in$

$A_{bend} := a_{bend} \cdot b_{bend} = 0.0003\ m^2$

$$I_{bend} := \frac{a_{bend} \cdot b_{bend}^{3}}{12} = 1.7343 \cdot 10^{-8}\ m^4$$

$\sigma_{bendmax} := 250\ MPa$ This is the tensile yield

$$\sigma_{bend} := M\left(\frac{l_{GB}}{2}\right) \cdot \frac{b_{bend} \cdot 0.5}{I_{bend}} = -197.9097\ MPa$$

while $\left(\left|\sigma_{bend}\right| < \sigma_{bendmax}\right)$

$$\begin{array}{|l} b_{bend} := \left(b_{bend} - 0.001\ mm\right) \\ A_{bend} := a_{bend} \cdot b_{bend} \\ I_{bend} := \dfrac{a_{bend} \cdot b_{bend}^{3}}{12} \\ \sigma_{bend} := M\left(\dfrac{l_{GB}}{2}\right) \cdot \dfrac{b_{bend} \cdot 0.5}{I_{bend}} \end{array}$$

$b_{bend} = 0.8897\ in$ $\sigma_{bend} = -250.0092\ MPa$

From the calculation, the minimum required area for the guard bracket is 2.87 cm^2.

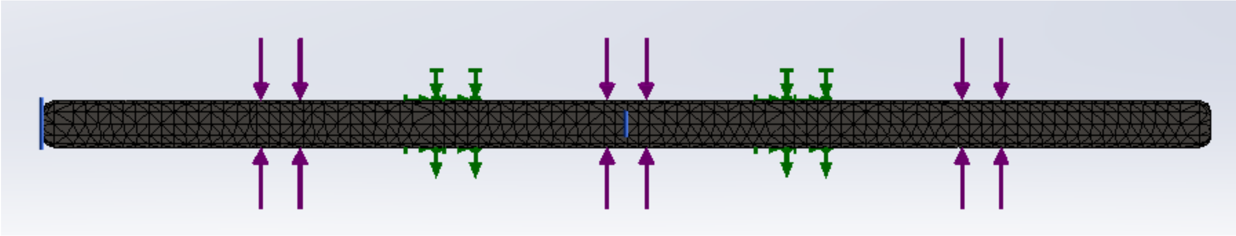
B.4: Finite Element Analysis

Concept 1: Hydraulic Tilter

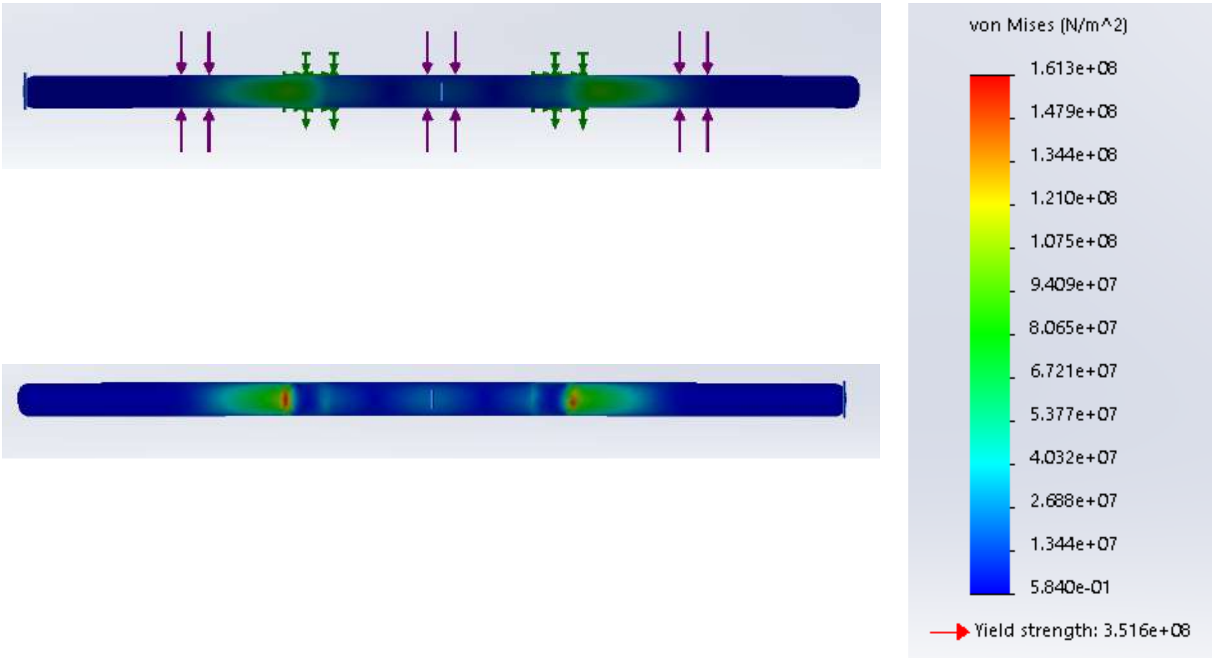
The FEA analysis was done on the hinge shaft part of the Tilter assembly. The shaft was chosen since it carried the weight of the tote directly on top.

A force of 13 kN was applied at the base of the Tilter where the tote sits. The stress is acceptable since the yield strength of the material is higher than the maximum stress. The strain is also acceptable since it is only about 0.054 % at maximum value.

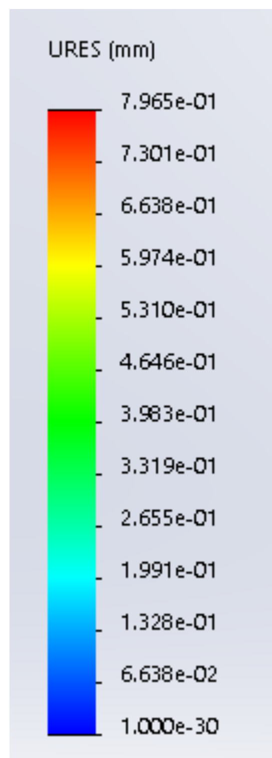
The FEA results for the shaft are shown in Figures B.4.1 – B.4.4.



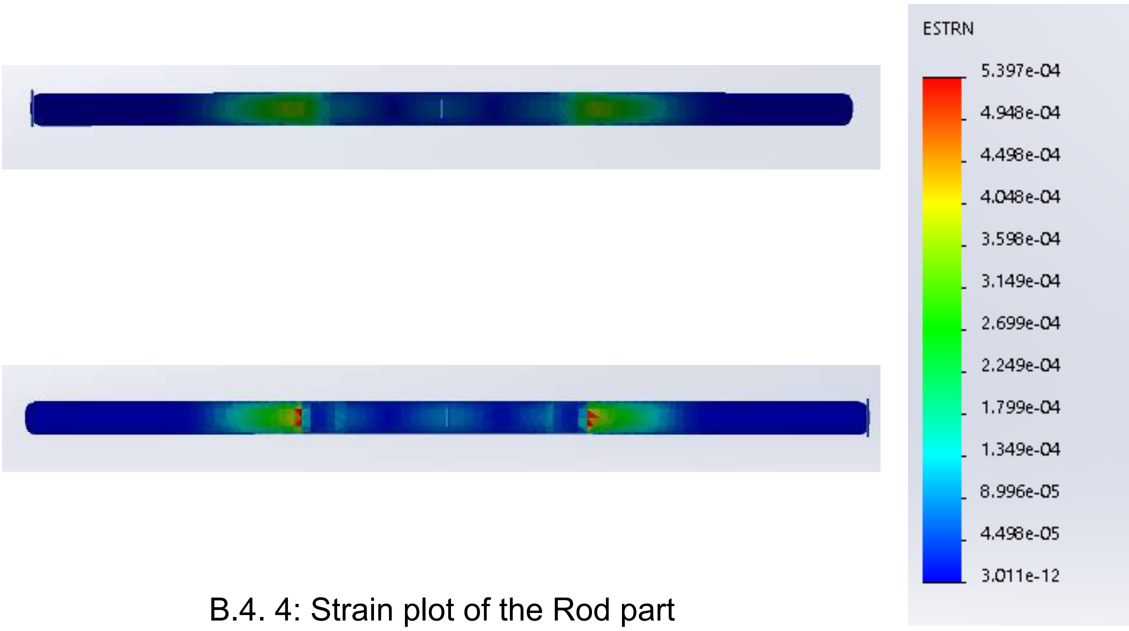
B.4.1: Manual mesh and location of forces and fixed point for Arm part.



B.4. 2: Stress plot of the Rod part



B.4. 3: Displacement plot of the Rod part



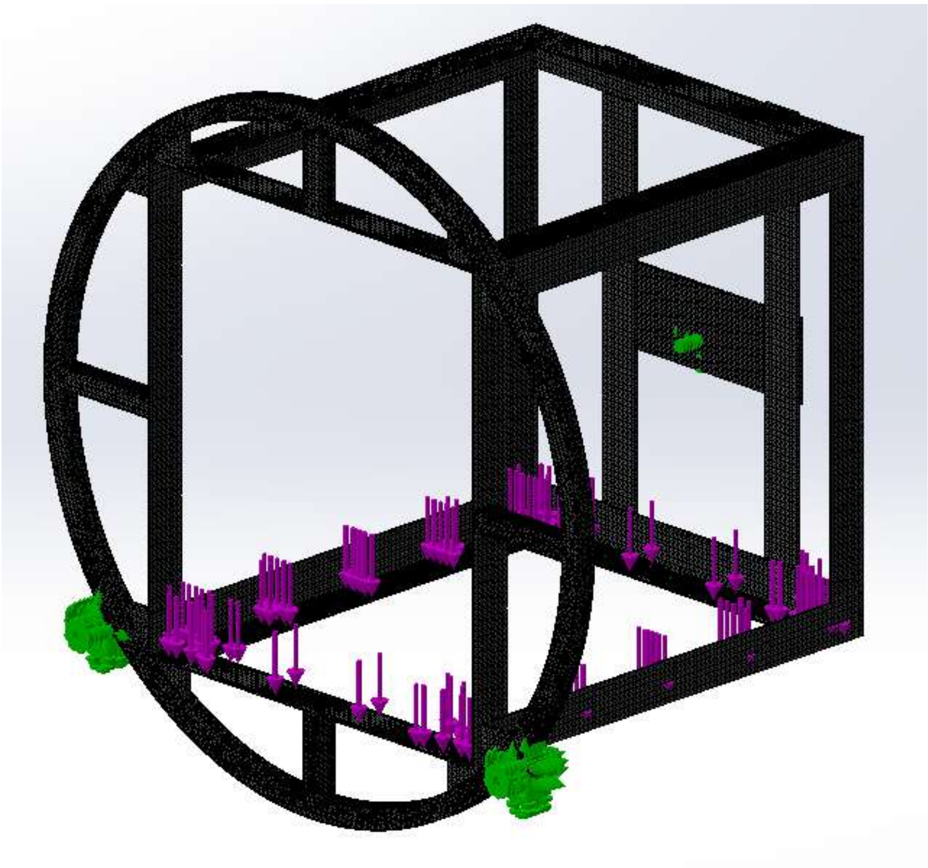
B.4. 4: Strain plot of the Rod part

Concept 2: Rotating Frame

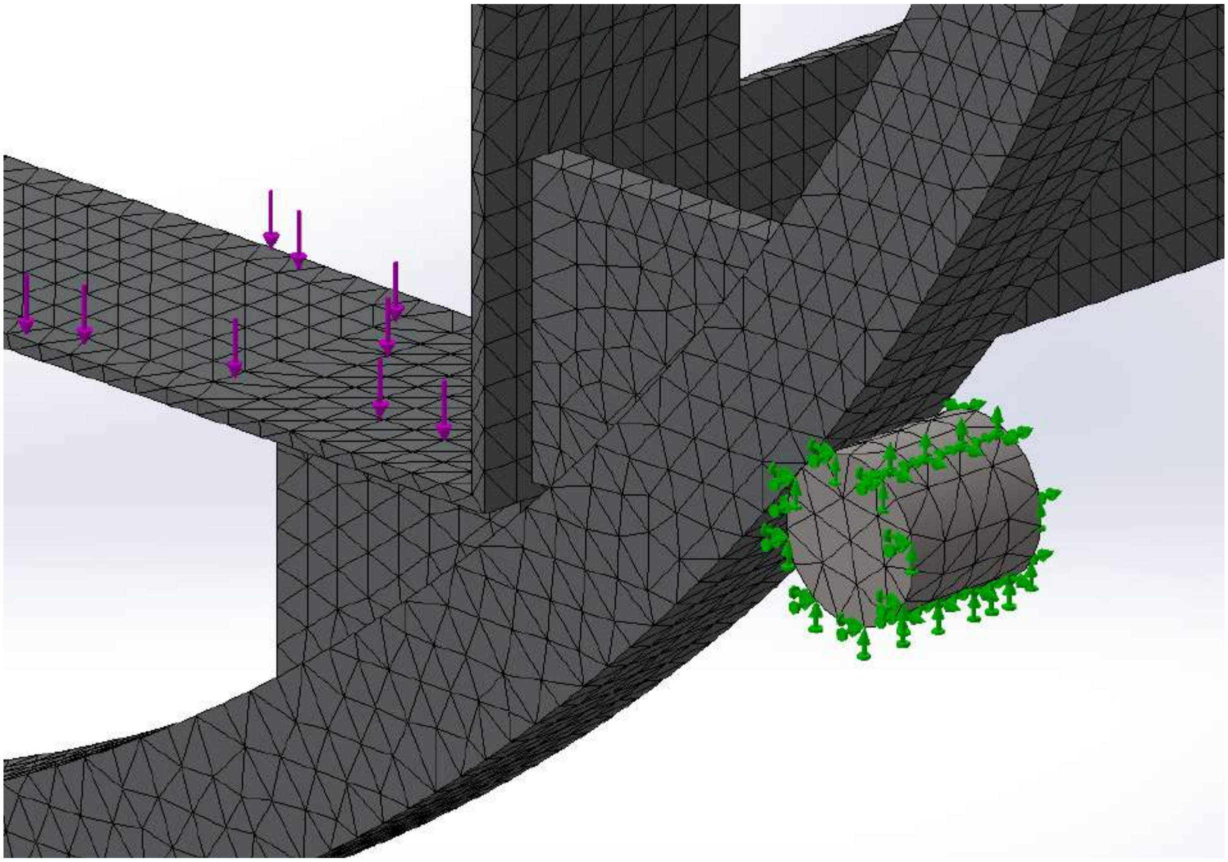
The FEA analysis was done on the Drum cage part of the Rotating Frame assembly. The Drum was chosen since it carried the weight of the tote and it is the only component that receives torque from the motor.

A Force of 9 kN was applied at the base of the drum where the tote sits. The stress is acceptable since it is a magnitude lower than the yield strength of the material. The strain is also acceptable since it is only about 0.016 % at maximum value.

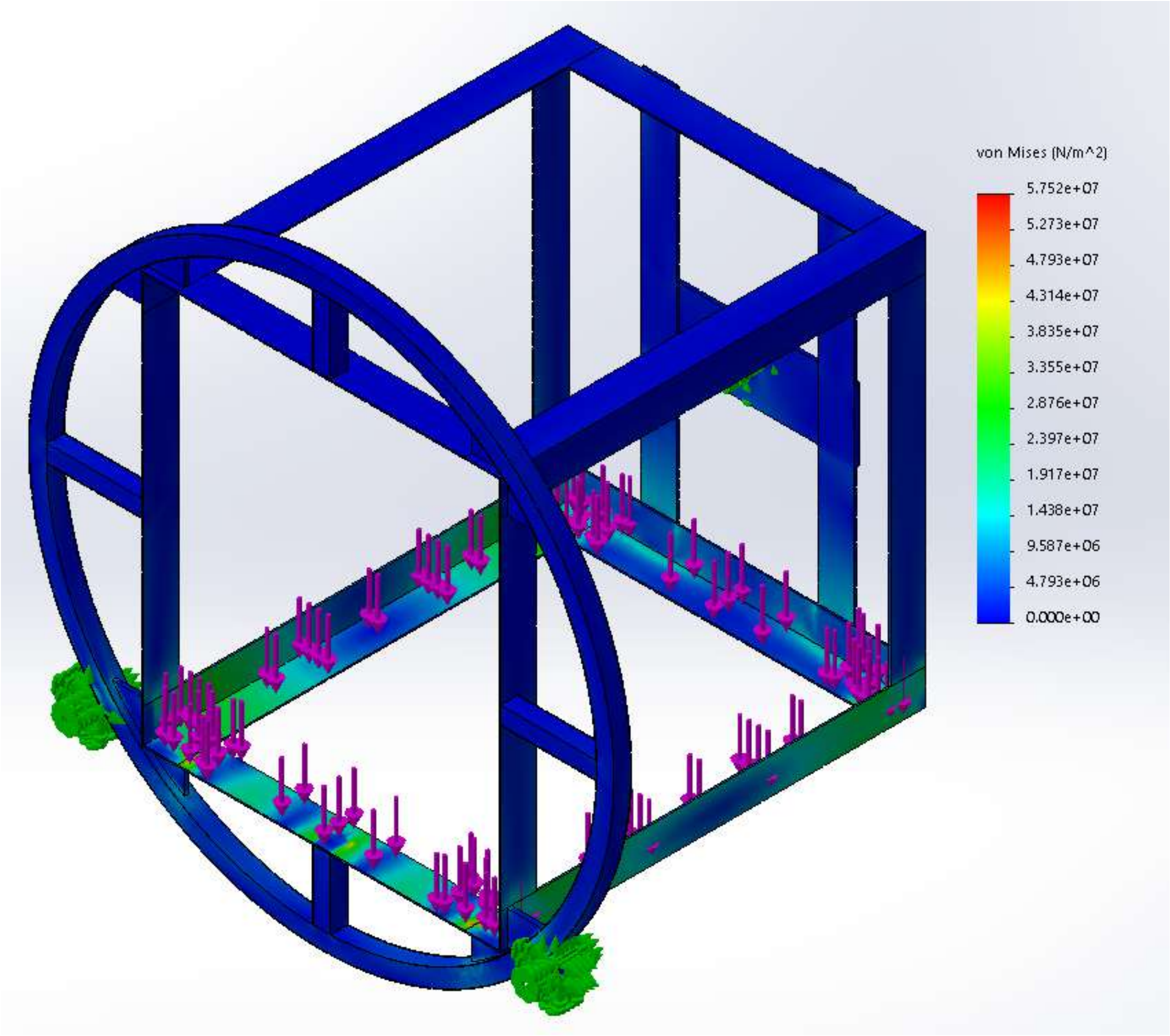
The FEA results for the Drum part are shown in Figures B.4.5 – B.4.9.



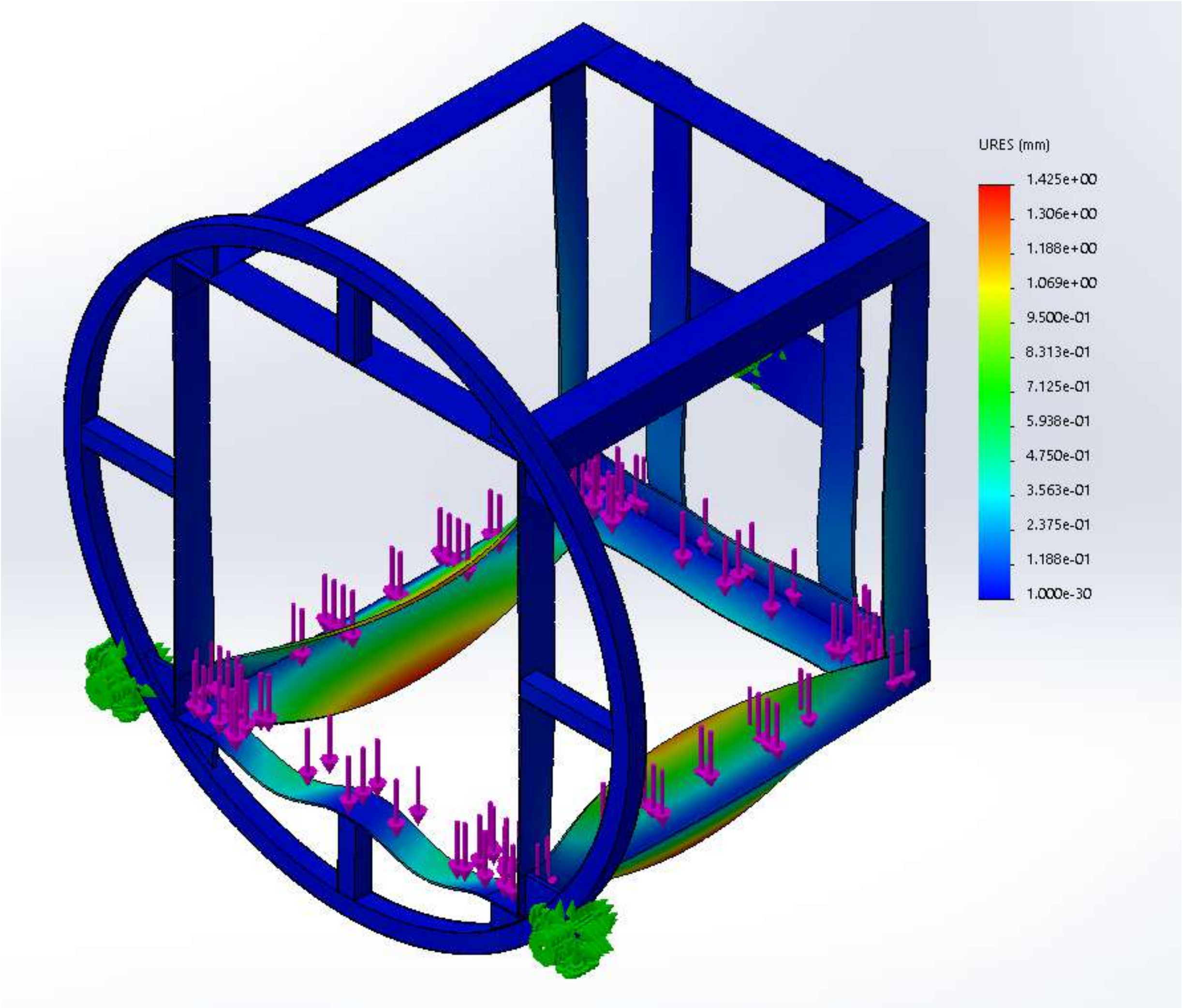
B.4.5: The fixed points and the force distribution applied to the Drum part



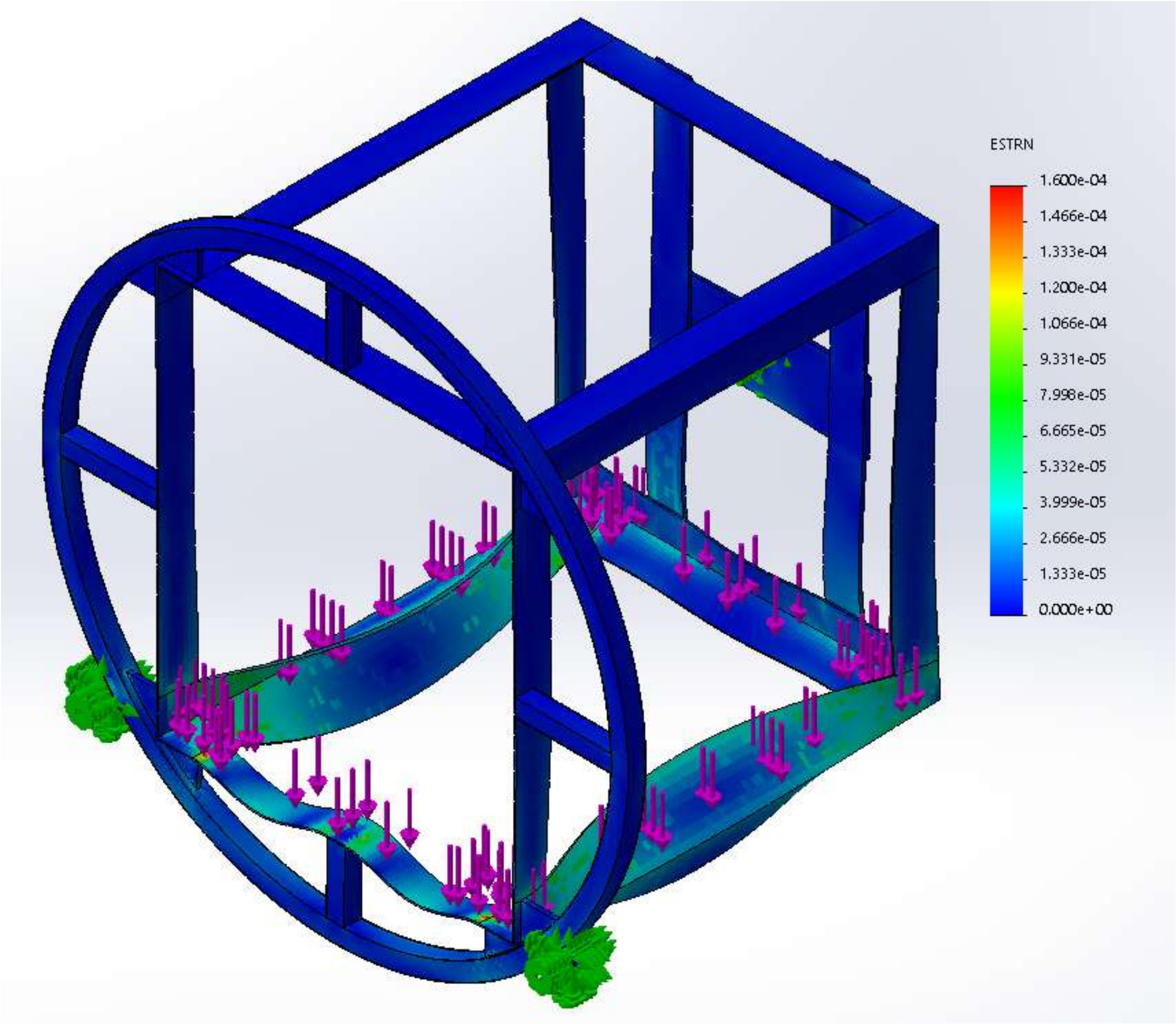
B.4.6: A close up of the manual mesh grid on Drum part



B.4.7: Stress plot of the Drum part



B.4.8: Displacement plot of the Drum part



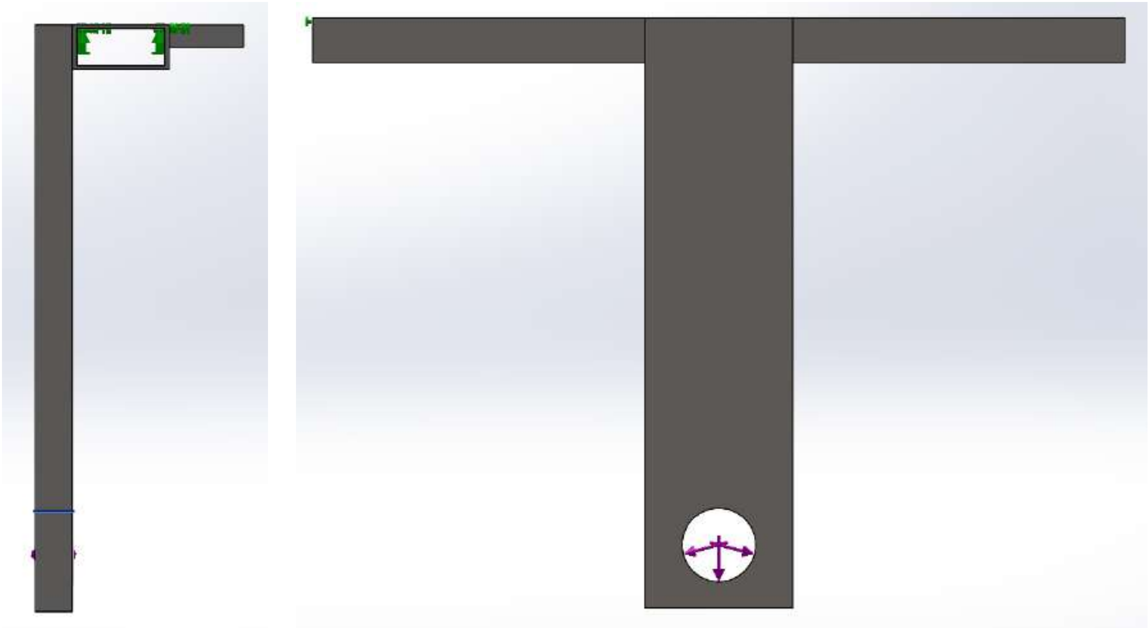
B.4. 9: Strain plot of the Drum part

Concept 3: Forklift Assembly

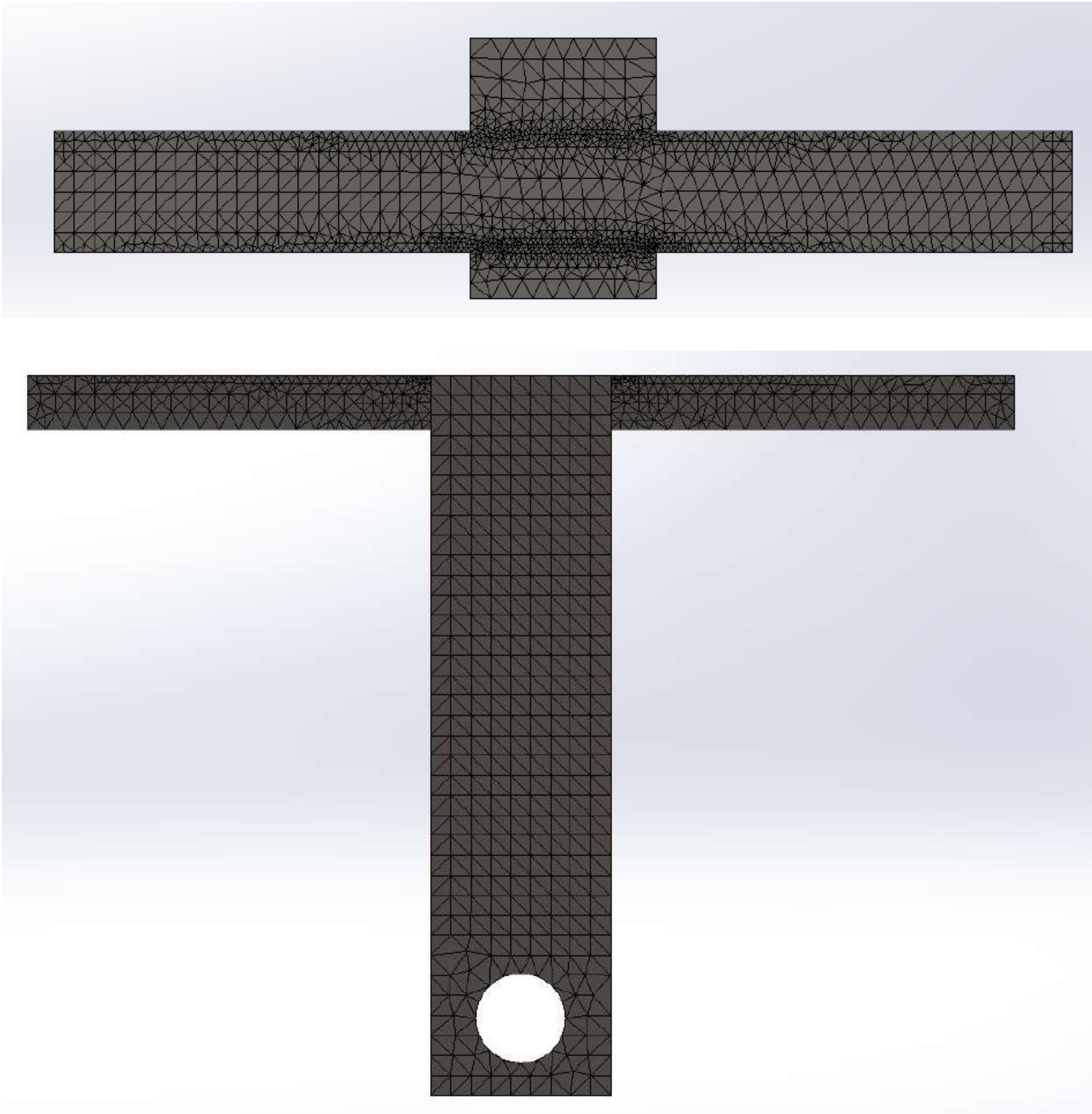
The FEA analysis was done on the Arm part of the Forklift Assembly. The Arm was chosen since it carried the weight of the cage and the tote. If the Arm failed, then the whole mechanism would fail.

A force of 8.54 kN was applied at the shaft hole. The stress is acceptable since it is a magnitude lower than the yield strength of the material. The strain is also acceptable since it is only about 0.012 % at maximum value.

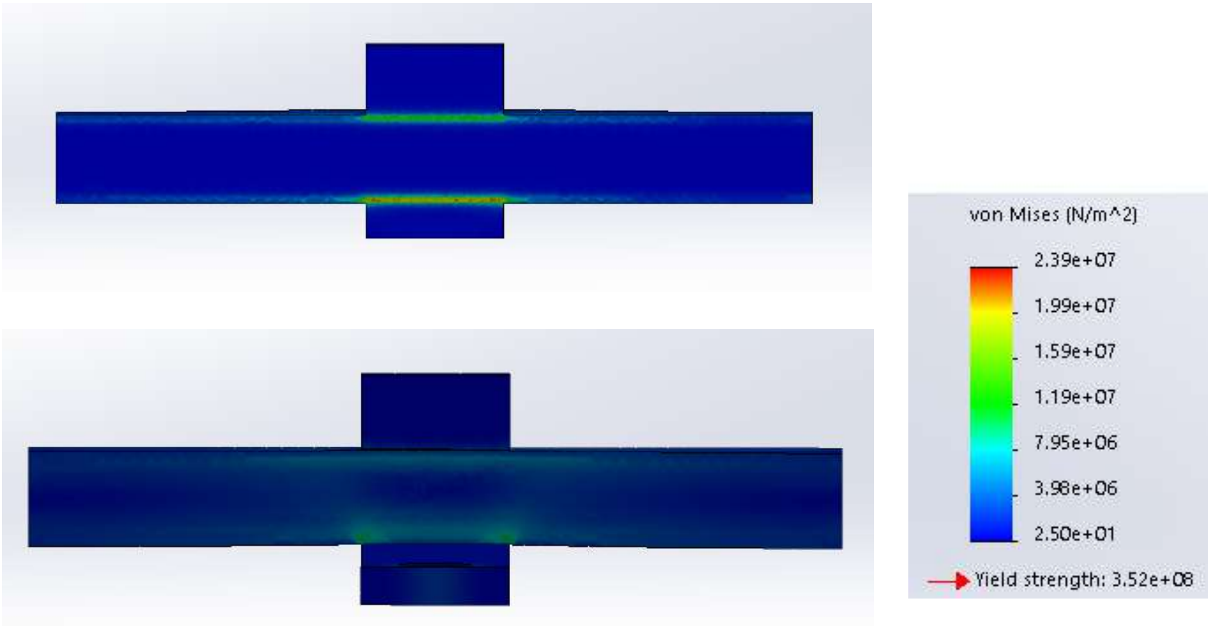
The FEA results for the Arm part are shown in Figures B.4.9 – B.4.14.



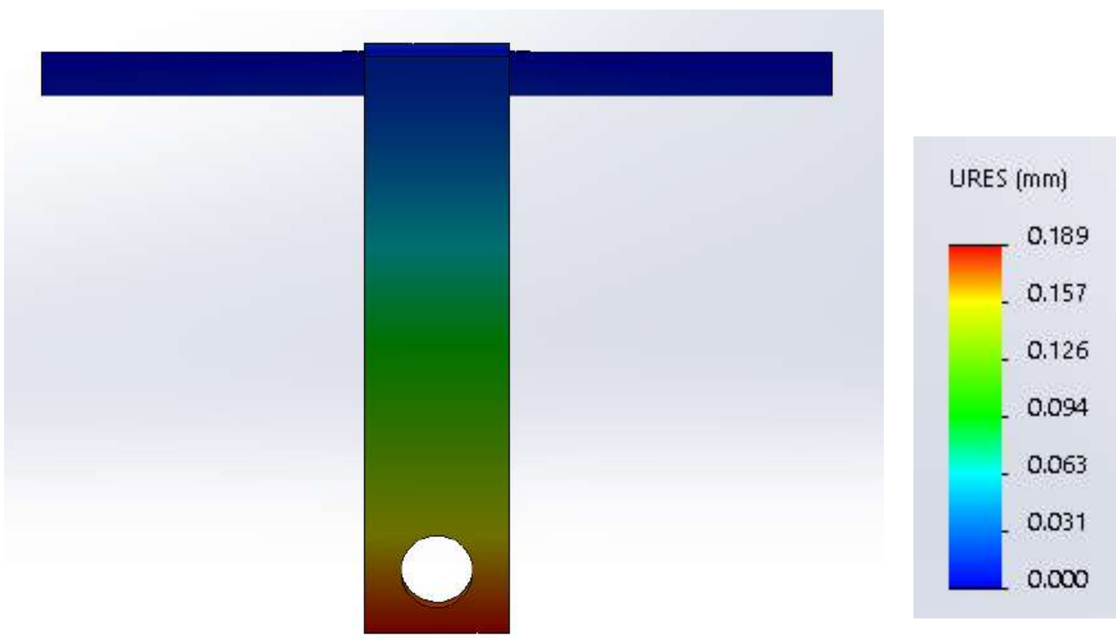
B.4. 10: The fixed points and the force distribution applied to the Arm part



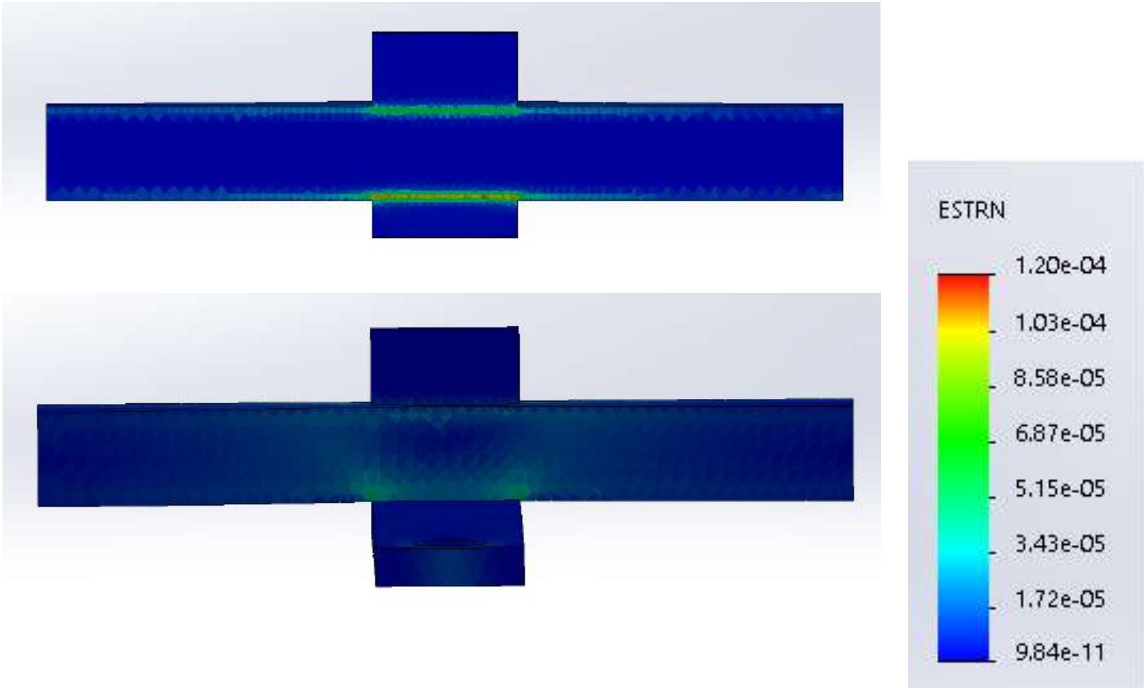
B.4. 11: The h-adaptive mesh grid on Arm part



B.4. 12: Von Mises Stress on Arm Part



B.4. 13: Displacement plot of Arm part



B.4. 14: Strain plot of Arm part

Appendix C: Cost Estimates

This appendix provides preliminary cost estimates for the Hydraulic Tilter, Rotating Frame, and Forklift Attachment concepts. Specifications that apply to the cost analysis of each concept are summarized in Table C1.

Table C1: Applicable design specifications for cost analysis

Item	Description	Specification
1.5	Overall Dimensions	At minimum the footprint of the machine will have the dimensions of the tote (~40"x45")
4.1	Manufacturing Cost	Cost of manufacturing the machine should be less than $5000. Machine should be designed to be affordable.
6.2	Repairs	Standard parts used wherever possible to simplify repairs
6.4	Total Component Count	Machine should have a simple design with fewest components necessary to reduce potential for part failure

The cost analysis for each concept is broken into material and manufacturing estimates. The cost of manufacturing for each concept design is determined by taking the hourly rate for an industry machine shop, Edmonton Fabrication Centre, and multiplying it by the estimated required manufacturing hours provided by the MEC E Shop Technicians. Standard materials and off the shelf products are selected in many cases to eliminate the cost of manufacturing custom parts. In some cases, however, some specialized fabricating is required for parts, like the ring in the Rotating Frame concept. The cost analysis for the three concepts is summarized in Table C2.

Table C2: Cost estimates for Hydraulic Tilter, Rotating Frame, and Forklift Attachment concepts

Description	Supplier	Part Number	Unit Cost (CAD)	Quantity	Total Cost (CAD)
Hydraulic Tilter					**$2408.44**
Labour	Edmonton Fabrication Centre	-	$120/hr	7.5 h	$900.00
Hydraulic Cylinder (1 ½ Bore)	Princess Auto [5]	62205K711	$140.00	2	$280.00
Mounted Sleeve Bearing	McMaster-Carr [4]	6359K29	$158.21	2	$316.42
Hydraulic Pump	Alibaba [11]	PV2R1	$85.39	1	$85.39
Base-Mount AC Motor (Nema 56C)		ML90L-4-2HP	$98.52	1	$98.52
Steel Sheet 11 Ga (48.00 x 72.00 in)	Metal Supermarkets [11]	CSH/048	$104.42	2	$200.83
Steel Rectangular Tube (3 x 1 x 0.065)		CTRT/31065	$17.11	25	$428.11
Steel Round Bar (2 in diameter, 50 in length)		HR2	$75.50	1	$99.17
Rotating Frame					**$7627.79**
Labour (Layout, fabrication, machining, welding)	Edmonton Fabrication Centre	-	$120/hr	30 hr	$2700.00
Square Tubing (2in x 2in x 0.065in)	Metal Supermarkets	CTSQ/2065	$22.87/ft	88 ft	$2012.56
Plate (3/16 in)		HP188	$31.82/ft^2	28 ft^2	$890.96
Angle (4in x 4in x 3/16in)		HA438	$37.21/ft	16 ft	$595.36
Round Bar (1in)		GSR1045/1	$33.34/ft	1 ft	$33.34
Motor Sprocket	McMaster-Carr	6280K447	$18.53	1	$18.53

(11 tooth, ANSI 41)					
Driven Sprocket (20 tooth, ANSI 41)		6280K582	$34.88	1	$34.88
Drive Chain (ANSI 41)		6261K174	$4.40/ft	10 ft	$44.00
Front Bearings (Standard duty ball)		6383K520	$20.15	2	$40.15
Rear Bearing (Standard duty ball)		6383K120	$10.07	1	$10.07
Motor (1/3hp, NEMA 56C)		59845K110	$490.81	1	$490.81
Reduction Gear (80:1, motor mount)		5887K221	$742.17	1	$742.17
Mounting Plate (0.53in, for gearbox)		5887K92	$14.96	1	$14.96
Forklift Attachment					**$3204.90**
Labour	Edmonton Fabrication Centre	-	$120/hr	15 hr	$1800.00
Motor	DB Electrical [11]	LTM0001	$250	2	$500
Vertical Beams (127.0 x 50.8 x 1778.0 mm)	China Synergy Group [10]	Plates/AISI 1020 (20#)	$70.19	4	$280.76
Horizontal Beams (127.0 x 50.8 x 1651.0 mm)		Plates/ AISI1020 (20#)	$65.17	4	$260.69
Cage Arm (127.0 x 50.8 x 1778.0 mm)		Q195-212A	$66.27	2	$132.55
Cage Fork (127.0 x 50.8 x 1651.0 mm)		Q195-212A	$71.37	2	$142.74
Arm Supporting Beam - Round Rod (152.4 x 254.0 mm)		AISI 1020 (C22,20#)	$15.75	2	$31.49

Counter Weight - Round Rod (50.8 x 1270.0 mm)		AISI 1020 (C22,20#)	$28.34	2	$56.67

The cost analysis shows that the Hydraulic Tilter is the most cost-effective concept, with an overall cost of $2408.44. The Rotating Frame is the most expensive concept with a cost of $7568.83. The Forklift Attachment falls between the other two concepts with an overall estimated cost of $3204.90.

Appendix D: Design Specification Matrix Changes

The Design Specification Matrix was updated to reflect the client's updated use case for the device. The marked-up Design Specification Matrix with the updates made after Phase I is shown in Table D1. A final version of the Design Specification Matrix is shown in Table D2.

Table D1: Marked up Design Specification Matrix

Item	Description	Specification	Authority	Importance (1-5)
1	**Dimensions**			
1.1	Height	Machine should fit and be able to operate in a room with a height of ~5.5m (18 ft)	Client	3
1.2	Weight	Weight should not exceed a dead load of 100 lbs/ft^2	NBC	5
1.3	Length of Protruding Bodies	Encapsulating mechanism will at a minimum reach the length of the tote (~45")	Client	4
1.4	Machine Tote Capacity	Operating capacity of machine should be at least one tote	Client	5
1.5	Overall Dimensions	At minimum the footprint of the machine will have the dimensions of the tote (~40"x45")	Client	3
2	**Safety Requirements**			
2.1	Method to Fix Position	Mechanism to fix position of load during lifting to prevent unexpected reduction in height	Apis	4
2.2	Mechanical Override	Mechanical override to adjust height of load in case of power loss	Apis	3
2.3	Electrical	Electrical components compliant with code	CSA C22.2 No. 301	5
2.4	Human Safety	Operator controls should not be stationed in close proximity to moving parts of the machine	CSA Z432-16	5
2.5	Emergency Stopping Device	In case of emergency operator should be able to stop the machine with an emergency stopping device	CSA 60601-1	5

2.6	Fall Protection	Operator should not be under suspended load while machine is in operation	OHSC	4
~~2.7~~	~~Load Capacity~~	~~Capable of supporting ~7000 lb load without experiencing any deformation/deflection (accounting for a 2x safety factor)~~	~~Client~~	~~5~~
2.8	Pinch Points	No exposed pinch points	CSA Z434-14	3
2.9	Operator Handling Loads	Operator should not have to lift, handle, or transport heavy or awkward loads without appropriate equipment	OHS	5
2.10	Operation Warning Signal	Indication of operation commencement and completion	CSA Z432-16	3
2.11	Cleaning Load Capacity	Capable of supporting ~ 405 lb load, with a deformation of less than 1% of a given dimension (3 x Safety Factor)	Client	5
2.12	Draining Load Capacity	Capable of supporting ~ 5250 lb load, with a deformation of less than 1% of a given dimension (1.5 X Safety Factor)	**Client**	5
3	**Food Safety**			
3.1	Contamination Prevention	Potential contaminant sources are properly enclosed (eg. lubricants)	Client	5
3.2	Food Grade Tote	Material of selected tote should comply to food standards of Canada	SFCR	5
4	**Cost**			
4.1	Manufacturing Cost	Cost of manufacturing the machine should be less than $5000. Machine should be designed to be affordable.	Client	5
4.2	Tote Selection	The tote selected should have a cheaper cost per volume than the 45 gallon food grade barrels currently used ($60)	Client	5
5	**Ergonomics**			
5.1	Control Panel	Control panel should be easily accesible and adjustable by operator	Apis	3
5.2	Vision	Clear line of sight to moving parts should be maintained at all times by operator	Apis	4
5.3	Noise Level	If noise exceeds 85 dBA hearing protection is required	OHS	3
5.4	Accessibility	Clear and intuitive machine operating process	Client	4

6	**Maintenance**			
6.1	Part Maintenance	Machine mechanisms should be easily accesible for maintenance	SFCR	2
6.2	Repairs	Standard parts used wherever possible to simplify repairs	Apis	3
6.3	Corrosion	Materials selected should be resistant to corrosion or be painted	CFIA	3
6.4	Total Component Count	Machine should have a simple design with fewest components necessary to reduce potential for part failure	Apis	3
7	**Environment**			
~~7.1~~	~~Power Consumption~~	~~Machine is designed to be electrically efficient~~	~~Apis~~	~~2~~
7.2	Operation Environment	The machine is expected to operate at temperatures between 0C ~ 35C	Apis	4
7.3	Storage Environment	The machine should withstand storage at temperatures of -40C to 40C	Apis	2
~~7.4~~	~~Manufacturing Process~~	~~Manufacturing process should be as environmentally friendly as possible~~	~~Apis~~	~~1~~
8	**Operation**			
8.1	Tote Crush Prevention	Machine should securely hold tote in any position without damaging the tote's structure	Client	5
8.2	2-Degrees of Freedom Motion	Machine should be able to lift and tilt the tote	Client	4
8.3	4-Degrees of Freedom Motion	Capable of planar and vertical translational motion, and rotational motion about one axis	Apis	4 ~~1~~
8.4	Stability	Machine should withstand tipping during operation or in case of potential impact	Apis	4
8.5	Machine Mobility	Capability of machine to be mobile	Apis	1
8.6	Water Contact	Machine will be subject to washing so all parts should be waterproof	Client	4
~~8.7~~	~~Tote Integrity~~	~~Selected totes should be stackable and able to withstand a 5 ft fall~~	~~Client~~	4
8.8	Electricity Supply	1-Phase electrical supply is used for operation	Client	3
8.9	Operational Precision Requirements	Ease of operator to insert the tote into the machine	Apis	4

8.10	Additional Training Required	Minimizes extra training required to operate the device, or its assoicated equipment (i.e. forklift)	Apis	3
8.11	Accessibility for Cleaning Tote	Easier access to tote for cleaning facilitated by machine	Client	3
8.12	Assist in Draining	Ability to tilt the tote to aid in draining of contents	Client / Apis	3

Table D2: Final Design Specification Matrix

Item	Description	Specification	Authority	Importance (1-5)
1	**Dimensions**			
1.1	Height	Machine should fit and be able to operate in a room with a height of ~5.5m (18 ft)	Client	3
1.2	Weight	Weight should not exceed a dead load of 100 lbs/ft^2	NBC	5
1.3	Length of Protruding Bodies	Encapsulating mechanism will at a minimum reach the length of the tote (~45")	Client	4
1.4	Machine Tote Capacity	Operating capacity of machine should be at least one tote	Client	5
1.5	Overall Dimensions	At minimum the footprint of the machine will have the dimensions of the tote (~40"x45")	Client	3
2	**Safety Requirements**			
2.1	Method to Fix Position	Mechanism to fix position of load during lifting to prevent unexpected reduction in height	Apis	4
2.2	Mechanical Override	Mechanical override to adjust height of load in case of power loss	Apis	3
2.3	Electrical	Electrical components compliant with code	CSA C22.2 No. 301	5
2.4	Human Safety	Operator controls should not be stationed in close proximity to moving parts of the machine	CSA Z432-16	5
2.5	Emergency Stopping Device	In case of emergency operator should be able to stop the machine with an emergency	CSA 60601-1	5

		stopping device		
2.6	Fall Protection	Operator should not be under suspended load while machine is in operation	OHSC	4
2.8	Pinch Points	No exposed pinch points	CSA Z434-14	3
2.9	Operator Handling Loads	Operator should not have to lift, handle, or transport heavy or awkward loads without appropriate equipment	OHS	5
2.10	Operation Warning Signal	Indication of operation commencement and completion	CSA Z432-16	3
2.11	Cleaning Load Capacity	Capable of supporting ~ 405 lb load, with a deformation of less than 1% of a given dimension (3 x Safety Factor)	Client	5
2.12	Draining Load Capacity	Capable of supporting ~ 5250 lb load, with a deformation of less than 1% of a given dimension (1.5 X Safety Factor)	**Client**	5
3	**Food Safety**			
3.1	Contamination Prevention	Potential contaminant sources are properly enclosed (eg. lubricants)	Client	5
3.2	Food Grade Tote	Material of selected tote should comply to food standards of Canada	SFCR	5
4	**Cost**			
4.1	Manufacturing Cost	Cost of manufacturing the machine should be less than $5000. Machine should be designed to be affordable.	Client	5
4.2	Tote Selection	The tote selected should have a cheaper cost per volume than the 45 gallon food grade barrels currently used ($60)	Client	5
5	**Ergonomics**			
5.1	Control Panel	Control panel should be easily accessible and adjustable by operator	Apis	3
5.2	Vision	Clear line of sight to moving parts should be maintained at all times by operator	Apis	4
5.3	Noise Level	If noise exceeds 85 dBA hearing protection is required	OHS	3
5.4	Accessibility	Clear and intuitive machine operating process	Client	4
6	**Maintenance**			

6.1	Part Maintenance	Machine mechanisms should be easily accessible for maintenance	SFCR	2
6.2	Repairs	Standard parts used wherever possible to simplify repairs	Apis	3
6.3	Corrosion	Materials selected should be resistant to corrosion or be painted	CFIA	3
6.4	Total Component Count	Machine should have a simple design with fewest components necessary to reduce potential for part failure	Apis	3
7	**Environment**			
7.2	Operation Environment	The machine is expected to operate at temperatures between 0C ~ 35C	Apis	4
7.3	Storage Environment	The machine should withstand storage at temperatures of -40C to 40C	Apis	2
8	**Operation**			
8.1	Tote Crush Prevention	Machine should securely hold tote in any position without damaging the tote's structure	Client	5
8.2	2-Degrees of Freedom Motion	Machine should be able to lift and tilt the tote	Client	4
8.3	4-Degrees of Freedom Motion	Capable of planar and vertical translational motion, and rotational motion about one axis	Apis	4
8.4	Stability	Machine should withstand tipping during operation or in case of potential impact	Apis	4
8.5	Machine Mobility	Capability of machine to be mobile	Apis	1
8.6	Water Contact	Machine will be subject to washing so all parts should be waterproof	Client	4
8.8	Electricity Supply	1-Phase electrical supply is used for operation	Client	3
8.9	Operational Precision Requirements	Ease of operator to insert the tote into the machine	Apis	4
8.10	Additional Training Required	Minimizes extra training required to operate the device, or its associated equipment (i.e. forklift)	Apis	3
8.11	Accessibility for Cleaning Tote	Easier access to tote for cleaning facilitated by machine	Client	3
8.12	Assist in Draining	Ability to tilt the tote to aid in draining of contents	Client / Apis	3

Appendix E: Project Management

E.1: Gantt Chart

A Gantt chart was created at the beginning of the project to address key project tasks and deadlines. The Gantt chart and associated schedule estimate serves as the basis for project work, and has been adjusted as tasks are completed and further discussed. Actual timelines for the tasks defined in the Gantt chart have been tracked as the project has progressed. The Gantt Chart is shown in Figure E.1.1.

ID	Task Mode	Task Name	Duration	Predec	Baseline Start	Baseline Finish	Actual Start	Actual Finish	Resource Initials	% Complete	Deadline
1		**Phase 1**	**52 days**		**Mon 9/9/19**	**Fri 10/4/19**	**Mon 9/9/19**	**Wed 10/30/19**		**100%**	**NA**
2		Library assignment	8 days		Fri 9/13/19	Thu 9/19/19	Fri 9/13/19	Fri 9/20/19	DS,ZL,AP,JS	100%	Fri 9/13/19
3		Project selection	2 days		Mon 9/9/19	Tue 9/10/19	Mon 9/9/19	Tue 9/10/19	T2	100%	Tue 9/10/19
4		Letter of intent	2 days	3	Wed 9/11/19	Thu 9/12/19	Wed 9/11/19	Thu 9/12/19	DS	100%	NA
5		Project scheduling	3 days		Mon 9/9/19	Wed 9/11/19	Mon 9/9/19	Wed 9/11/19	AR	100%	NA
6		Initial client meeting	1 day		Mon 9/16/19	Mon 9/16/19	Mon 9/16/19	Mon 9/16/19	T2	100%	NA
7		Initial advisor meeting	1 day		Mon 9/16/19	Mon 9/16/19	Mon 9/16/19	Mon 9/16/19	T2,ZC	100%	NA
8		Preliminary research	8 days	6,7	Tue 9/17/19	Tue 9/24/19	Tue 9/17/19	Tue 9/24/19	T2	100%	NA
9		**Report 1**	**50 days**		**Mon 9/16/19**	**Wed 9/25/19**	**Wed 9/11/19**	**Wed 10/30/19**		**100%**	Mon 9/30/19
10		Identify design scope	30 days	6,7	Tue 9/17/19	Mon 9/23/19	Tue 9/17/19	Wed 10/16/19	AJ,AP	100%	NA
11		Market research	38 days	10	Thu 9/19/19	Mon 9/23/19	Thu 9/19/19	Sat 10/26/19	AP,JS	100%	NA
12		Literature research	38 days	10	Thu 9/19/19	Mon 9/23/19	Thu 9/19/19	Sat 10/26/19	T2	100%	NA
13		Design specification matrix	37 days	6,7	Tue 9/17/19	Tue 9/17/19	Tue 9/17/19	Wed 10/23/19	T2	100%	NA
14		Preliminary calculations	35 days	10	Thu 9/19/19	Wed 9/25/19	Thu 9/19/19	Wed 10/23/19	T2	100%	NA
15		Team charter	1 day		Mon 9/23/19	Mon 9/23/19	Mon 9/23/19	Mon 9/23/19	T2	100%	NA
16		IP agreement	1 day	6	Tue 9/17/19	Tue 9/17/19	Tue 9/17/19	Tue 9/17/19	AJ,AP	100%	NA
17		Cost estimation	1 day		Fri 9/27/19	Fri 9/27/19	Fri 9/27/19	Fri 9/27/19	AP,JS	100%	NA
18		Draft report	9 days		Wed 9/11/19	Sun 9/22/19	Wed 9/11/19	Thu 9/19/19	JS,ZL	100%	NA
19		Edit report	2 days	18	Mon 9/23/19	Fri 9/27/19	Tue 10/29/19	Wed 10/30/19	T2	100%	NA
20		Cover letter	1 day	18	Mon 9/23/19	Mon 9/23/19	Mon 9/23/19	Mon 9/23/19	DS,JS	100%	NA
21		Submit report	0 days	19,20	Mon 9/30/19	Mon 9/30/19	Mon 9/30/19	Mon 9/30/19	JS	100%	NA
22		Survey 1	1 day		Fri 10/4/19	Fri 10/4/19	Fri 10/4/19	Fri 10/4/19	T2	100%	Fri 10/4/19
23		**Phase 2**	**40 days**		**Mon 9/30/19**	**Fri 11/8/19**	**Mon 9/30/19**	**NA**		**99%**	**NA**
24		Design brainstorming	6 days		Mon 9/30/19	Mon 10/7/19	Mon 9/30/19	Sat 10/5/19	T2	100%	NA
25		Concept proposals	1 day	24	Tue 10/8/19	Fri 10/11/19	Mon 10/7/19	Mon 10/7/19	T2	100%	NA
26		Brain-writing	1 day		Mon 10/7/19	Mon 10/7/19	Mon 10/7/19	Mon 10/7/19	T2	100%	NA
27		Concept modelling (Rev 0)	5 days	25	Mon 10/14/1	Sun 10/13/1	Tue 10/8/19	Sat 10/12/19	ZL,AJ,AR	100%	NA
28		Concept modelling (Rev 1)	1 day		Mon 10/21/1	Mon 10/28/1	Mon 10/21/...	Mon 10/21/19	ZL,AJ,AR	100%	NA
29		Concept calculations	15 days	25	Mon 10/14/1	Fri 10/18/19	Mon 10/14/...	Mon 10/28/19	AJ,AP	100%	NA
30		Component research	5 days	25	Mon 10/14/1	Fri 10/18/19	Mon 10/14/...	Fri 10/18/19	AR,DS	100%	NA
31		Vendor research	12 days	25	Fri 10/18/19	Tue 10/22/1	Mon 10/21/...	Fri 11/1/19	JS,ZL	100%	NA
32		Cost evaluation	12 days	25	Mon 10/21/1	Thu 10/24/1	Mon 10/21/...	Fri 11/1/19	AJ,AP	100%	NA
33		Material selection	12 days	25	Thu 10/24/1	Fri 10/25/19	Mon 10/21/...	Fri 11/1/19	AJ,AP	100%	NA
34		**Report 2**	**8 days**		**Mon 9/30/19**		**Mon 10/28/...**	**Mon 11/4/19**		**100%**	Mon 11/4/19
35		Design evaluation matrix	5 days	13	Sat 10/12/19	Sat 10/12/19	Mon 10/28/...	Fri 11/1/19	JS,ZL	100%	NA
36		Draft report	5 days		Wed 10/2/19	Wed 10/23/1	Mon 10/28/...	Fri 11/1/19	AJ,AP	100%	NA
37		Edit report	1 day	36	Thu 10/24/1	Fri 11/1/19	Fri 11/1/19	Fri 11/1/19	AR,DS	100%	NA
38		Executive summary	5 days	36	Sun 10/27/1	Tue 10/29/1	Mon 10/28/...	Fri 11/1/19	JS,ZL	100%	NA
39		Cover letter	5 days	36	Wed 10/30/1	Wed 10/30/1	Mon 10/28/...	Fri 11/1/19	AJ,AP	100%	NA
40		Submit report	1 day	37,38,	Mon 11/4/19	Mon 11/4/19	Mon 11/4/19	Mon 11/4/19	JS	100%	NA
41		Survey 2	1 day		Fri 11/8/19	Fri 11/8/19	Fri 11/8/19	NA	T2	0%	Fri 11/8/19
42		**Phase 3**	**45 days**		**Wed 10/30/1**	**Fri 12/13/19**	**NA**	**NA**		**0%**	**NA**
43		**Final solid model**	**10 days**		**NA**	**NA**	**NA**	**NA**		**0%**	**NA**
44		Implement tote securement	10 days	27	Sat 11/2/19	Thu 11/14/1	NA	NA	JS,ZL	0%	NA
45		Implement floor mount	10 days	27	Sat 11/2/19	Thu 11/14/1	NA	NA	AJ,AP	0%	NA
46		Optimize weight and cost	10 days	27	Sat 11/2/19	Thu 11/14/1	NA	NA	AR,DS	0%	NA
47		Final component selection	10 days	30	Tue 11/5/19	Fri 11/8/19	NA	NA	JS,ZL	0%	NA
48		Final calculations	15 days	29	Wed 10/30/1	Tue 11/19/1	NA	NA	T2	0%	NA
49		Final FEA	15 days	29	Wed 10/30/1	Tue 11/19/1	NA	NA	AJ,AP	0%	NA
50		**Report 3**					**NA**	**NA**		**0%**	Mon 12/2/19
51		Drawing package	7 days	46	Mon 11/18/1	Tue 11/26/1	NA	NA	JS,ZL	0%	NA
52		Design compliance matrix	3 days	35	Tue 11/5/19	Thu 11/7/19	NA	NA	AJ,AP	0%	NA
53		Draft report	14 days		Tue 11/5/19	Fri 11/22/19	NA	NA	AR,DS	0%	NA
54		Edit report	5 days	53	Mon 11/25/1	Fri 11/29/19	NA	NA	JS,ZL	0%	NA
55		Executive summary	2 days	53	Tue 11/26/1	Thu 11/28/1	NA	NA	AJ,AP	0%	NA
56		Cover letter	1 day	53	Tue 11/26/1	Tue 11/26/1	NA	NA	AR,DS	0%	NA
57		Submit report	0 days	54,55,	Mon 12/2/19	Mon 12/2/19	NA	NA	JS	0%	NA
58		Poster	4 days		Mon 12/2/19	Wed 12/4/19	NA	NA	JS,ZL	0%	NA
59		Prepare Presentation	4 days		Mon 12/2/19	Thu 12/5/19	NA	NA	T2	0%	NA
60		Deliver Presentation	1 day		Fri 12/6/19	Fri 12/6/19	NA	NA	T2	0%	NA
61		Survey 3	1 day		Fri 12/13/19	Fri 12/13/19	NA	NA	T2	0%	Fri 12/13/19

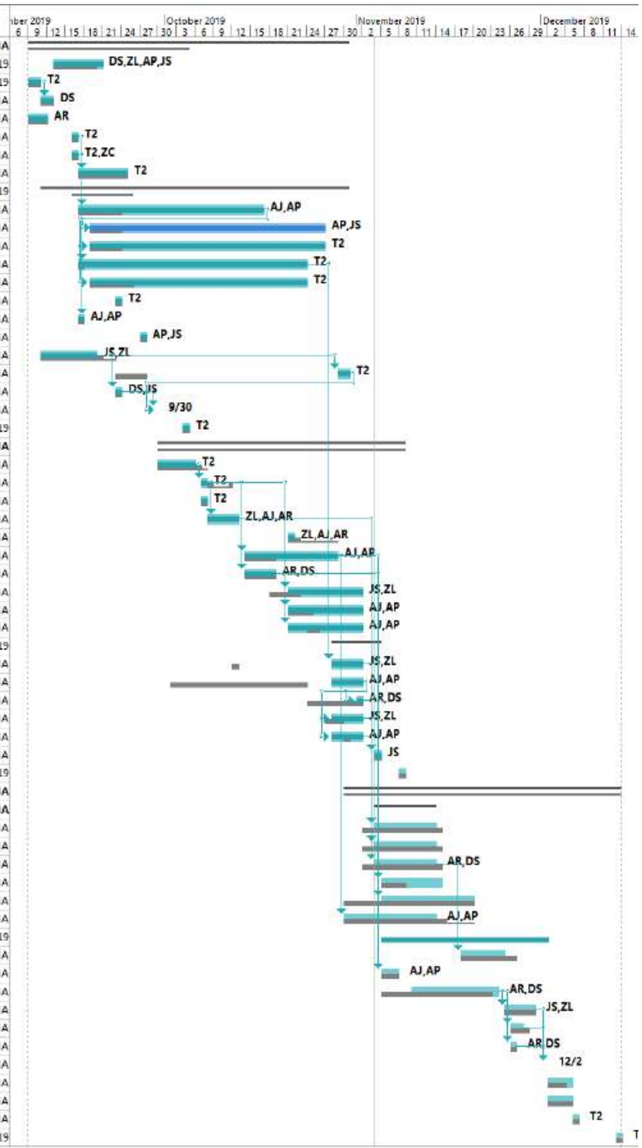


Figure E.1.1: Updated Gantt Chart

E.2: Timesheets

All team members have recorded the time spent working on tasks throughout the project. A summary of the weekly hours spent by each person in Phase I and II is shown in Table E.2.1. A description of every task performed by each team member, along with the date and time spent working on it are recorded in Figure E.2.1 - E.2.6.

Table E.2. 1: Weekly breakdown of hours spent per team member

			Aditya Jain	Adrian Phiri	Alex Rodd	Devon Saroya	Jorrell Serrano	Zhe Lyu
Phase 1	**Week 1**	Sep 9-15	3:45	10:35	4:45	5:00	4:45	3:56
	Week 2	Sep 16-22	4:00	6:15	5:00	8:00	6:45	3:56
	Week 3	Sep 23-29	11:00	9:35	11:30	6:30	19:15	8:41
Phase 2	**Week 4**	Sep 30 - Oct 6	4:15	4:15	6:15	0:00	6:00	0:00
	Week 5	Oct 7-13	4:45	6:55	6:00	7:30	8:15	8:00
	Week 6	Oct 14-20	3:40	7:40	4:00	4:00	1:30	5:30
	Week 7	Oct 21-27	3:00	11:10	7:30	10:00	13:45	7:30
	Week 8	Oct 28 - Nov 3	15:30	30:35	31:00	28:30	30:30	22:49
TOTAL			**49:55**	**87:00**	**76:00**	**69:30**	**90:45**	**60:22**

Day	Date	Start	End	Duration (H:mm)	Task Comments/Description	Week	Weekly Total (H:mm)
Phase I							
Monday	9/9/2019	2:00 PM	5:45 PM	3:45	First meeting	1	3:45
Tuesday	9/10/2019			0:00			
Wednesday	9/11/2019			0:00			
Thursday	9/12/2019			0:00			
Friday	9/13/2019			0:00			
Saturday	9/14/2019			0:00			
Sunday	9/15/2019			0:00			
Monday	9/16/2019	2:00 PM	6:00 PM	4:00	Client and advisor meeting	2	4:00
Tuesday	9/17/2019			0:00			
Wednesday	9/18/2019			0:00			
Thursday	9/19/2019			0:00			
Friday	9/20/2019			0:00			
Saturday	9/21/2019			0:00			
Sunday	9/22/2019			0:00			
Monday	9/23/2019	2:00 PM	6:00 PM	4:00	Meeting, advisor meeting	3	8:00
Tuesday	9/24/2019			0:00			
Wednesday	9/25/2019			0:00			
Thursday	9/26/2019			0:00			
Friday	9/27/2019	12:00 PM	4:00 PM	4:00	Phase 1 report work		
Saturday	9/28/2019			0:00			
Sunday	9/29/2019			0:00			
Monday	9/30/2019	2:00 PM	5:00 PM	3:00	Group Meeting, Final Report Review		
Phase II							
Tuesday	10/1/2019			0:00		4	7:15
Wednesday	10/2/2019			0:00			
Thursday	10/3/2019			0:00			
Friday	10/4/2019	1:00 PM	2:15 PM	1:15	Brainstorming		
Saturday	10/5/2019			0:00			
Sunday	10/6/2019	1:45 PM	4:45 PM	3:00	Brainstorming		
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Team Meeting, Brain Writing Excercise, Concept development and selection	5	4:45
Tuesday	10/8/2019			0:00			
Wednesday	10/9/2019			0:00			
Thursday	10/10/2019	5:00 PM	6:45 PM	1:45	solid model for forklift		
Friday	10/11/2019			0:00			
Saturday	10/12/2019			0:00			
Sunday	10/13/2019			0:00			
Monday	10/14/2019			0:00		6	3:40
Tuesday	10/15/2019			0:00			
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019			0:00			
Friday	10/18/2019			0:00			
Saturday	10/19/2019	4:00 AM	4:45 AM	0:45	redesign forklift		
Sunday	10/20/2019	9:50 AM	12:45 PM	2:55	forklift calculation		
Monday	10/21/2019	2:00 AM	5:00 AM	3:00	meeting	7	3:00
Tuesday	10/22/2019			0:00			
Wednesday	10/23/2019			0:00			
Thursday	10/24/2019			0:00			
Friday	10/25/2019			0:00			
Saturday	10/26/2019			0:00			
Sunday	10/27/2019			0:00			
Monday	10/28/2019	3:00 AM	5:00 AM	2:00	group meeting	8	12:30
Tuesday	10/29/2019			0:00			
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019			0:00			
Friday	11/1/2019	2:00 AM	5:00 AM	3:00	work session		
Saturday	11/2/2019	2:00 AM	5:00 AM	3:00	forklift images and description		
Sunday	11/3/2019	9:30 AM	2:00 PM	4:30	work session		
Monday	11/4/2019	2:00 PM	5:00 PM	3:00			

Figure E.2.1: Timesheet for Aditya Jain

Day	Date	Start	End	Duration (H:mm)	Task Comments/ Description	Week	Weekly Total (H:mm)
				Phase I			
Monday	9/9/2019	2:00 PM	5:45 PM	3:45	First Team Meeting and Letter of Intent Discussion	1	10:35
Tuesday	9/10/2019	3:00 PM	3:45 PM	0:45	Finalization of Letter of Intent and Project Selection		
Wednesday	9/11/2019	3:00 PM	5:30 PM	2:30	Establishing Preliminary Organization of Project Content. Initial Patent/Standard Search. Determining Questions for Client		
Thursday	9/12/2019			0:00			
Friday	9/13/2019	10:00 AM	12:00 PM	2:00	Completed Library Assignment 1 (True Time Frame was 10:00 - 11:00 AM and 12:00 - 1:00 PM)		
Saturday	9/14/2019			0:00			
Sunday	9/15/2019	8:00 AM	9:35 AM	1:35	Created Time Sheet Draft 1 Excel Document for Team		
Monday	9/16/2019	2:00 PM	5:30 PM	3:30	Team Meeting, Client Meeting, and Advisor Meeting	2	6:15
Tuesday	9/17/2019			0:00			
Wednesday	9/18/2019	9:00 AM	9:20 AM	0:20	Phase 1 Report Work		
Thursday	9/19/2019	3:50 PM	5:45 PM	1:55	Phase 1 Report Work		
Friday	9/20/2019			0:00			
Saturday	9/21/2019			0:00			
Sunday	9/22/2019	6:00 PM	6:30 PM	0:30	Discussed Group Objectives and Issues for Next Meeting		
Monday	9/23/2019	2:00 PM	6:15 PM	4:15	leeting. Determined Questions for Client, Worked on DSM, Worked on Gantt Chart, Spoke with	3	6:35
Tuesday	9/24/2019			0:00			
Wednesday	9/25/2019			0:00			
Thursday	9/26/2019			0:00			
Friday	9/27/2019			0:00			
Saturday	9/28/2019	7:30 PM	9:50 PM	2:20	Worked on Phase 1 Report Finalization		
Sunday	9/29/2019			0:00			
Monday	9/30/2019	2:00 PM	5:00 PM	3:00	Group Meeting, Final Report Review		
				Phase II			
Tuesday	10/1/2019			0:00		4	7:15
Wednesday	10/2/2019			0:00			
Thursday	10/3/2019			0:00			
Friday	10/4/2019	1:00 PM	2:15 PM	1:15	Brainstorming		
Saturday	10/5/2019			0:00			
Sunday	10/6/2019	1:45 PM	4:45 PM	3:00	Brainstorming		
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Team Meeting, Brain Writing Excercise, Concept development and selection	5	6:55
Tuesday	10/8/2019			0:00			
Wednesday	10/9/2019			0:00			
Thursday	10/10/2019	5:00 PM	6:45 PM	1:45	Tilter Concept definition		
Friday	10/11/2019			0:00			
Saturday	10/12/2019			0:00			
Sunday	10/13/2019	9:35 AM	11:45 AM	2:10	Reviewed Phase 1 Report, Determined fundamental Concept calculations, Reviewed communication mehtods, further concept review.		
Monday	10/14/2019			0:00		6	7:40
Tuesday	10/15/2019			0:00			
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019	4:30 PM	5:30 PM	1:00	Met with Dr. Chen to appraise him of Project's current standing		
Friday	10/18/2019			0:00			
Saturday	10/19/2019	4:00 AM	4:45 AM	0:45	Tilter concept design calculations		
Sunday	10/20/2019	9:50 AM	3:45 PM	5:55	Work on the Tilter concept calculations (True Duration was from 9:50 AM - 12:30 PM and from 2:45 PM - 3:30 PM, and 4:05 PM - 7:05 PM)		
Monday	10/21/2019	2:00 PM	5:30 PM	3:30	Team Meeting	7	11:10
Tuesday	10/22/2019			0:00			
Wednesday	10/23/2019			0:00			
Thursday	10/24/2019	5:00 PM	6:30 PM	1:30	Worked on Tilter Force Calcs on Smath		
Friday	10/25/2019	3:45 PM	8:20 PM	4:35	Worked on Shear and Moment Singularity Equations for Tilter Hinge Shaft (True Duration 3:45 - 6:35, 9:45 - 11:30)		
Saturday	10/26/2019	6:00 PM	19:00	1:00	Formatting Calculations		
Sunday	10/27/2019	9:30 PM	10:05 PM	0:35	Correcting Formula Derivations for Forces		
Monday	10/28/2019	1:00 PM	5:30 PM	4:30	Shop Visit and Group Meeting	8	26:35
Tuesday	10/29/2019			0:00			
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019			0:00			
Friday	11/1/2019	12:00 PM	7:30 PM	7:30	Worked on Phase 2 Report with Individually (12:00 - 2:00) and with the Team (2:00 - 7:30)		
Saturday	11/2/2019	11:30 AM	2:50 PM	3:20	Worked on Phase 2 Report. True Duration 11:30 - 1:30, 3:20 - 3:30, 7:20 - 8:30		
Sunday	11/3/2019	9:30 AM	8:45 PM	11:15			
Monday	11/4/2019	1:00 PM	5:00 PM	4:00	Team meeting		

Figure E.2.2: Timesheet for Adrian Phiri

Day	Date	Start	End	Duration (H:mm)	Task Comments/ Description	Week	Weekly Total (H:mm)
				Phase I			
Monday	9/9/2019	2:00 PM	5:45 PM	3:45	First meeting	1	4:45
Tuesday	9/10/2019			0:00			
Wednesday	9/11/2019	4:00 PM	5:00 PM	1:00	Email client		
Thursday	9/12/2019			0:00			
Friday	9/13/2019			0:00			
Saturday	9/14/2019			0:00			
Sunday	9/15/2019			0:00			
Monday	9/16/2019	2:00 PM	6:00 PM	4:00	Client and advisor meeting	2	5:00
Tuesday	9/17/2019			0:00			
Wednesday	9/18/2019	5:00 PM	5:30 PM	0:30	Email client		
Thursday	9/19/2019			0:00			
Friday	9/20/2019	5:00 PM	5:30 PM	0:30	Email client		
Saturday	9/21/2019			0:00			
Sunday	9/22/2019			0:00			
Monday	9/23/2019	2:00 PM	6:00 PM	4:00	Meeting, advisor meeting	3	8:30
Tuesday	9/24/2019			0:00			
Wednesday	9/25/2019	5:00 PM	5:30 PM	0:30	Email client		
Thursday	9/26/2019			0:00			
Friday	9/27/2019	12:00 PM	4:00 PM	4:00	Create Gantt chart		
Saturday	9/28/2019			0:00			
Sunday	9/29/2019			0:00			
Monday	9/30/2019	2:00 PM	5:00 PM	3:00	Group Meeting, Final Report Review	4	9:15
				Phase II			
Tuesday	10/1/2019	7:00 PM	8:00 PM	1:00	Idea Generation		
Wednesday	10/2/2019			0:00			
Thursday	10/3/2019	4:00 PM	5:00 PM	1:00	Idea Generation		
Friday	10/4/2019	1:00 PM	2:15 PM	1:15	Brainstorming		
Saturday	10/5/2019			0:00			
Sunday	10/6/2019	1:45 PM	4:45 PM	3:00	Brainstorming		
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Team Meeting, Brain Writing Excercise, Concept development and selection	5	6:00
Tuesday	10/8/2019			0:00			
Wednesday	10/9/2019	6:00 PM	9:00 PM	3:00	SolidWorks concept modeling for drum		
Thursday	10/10/2019			0:00			
Friday	10/11/2019			0:00			
Saturday	10/12/2019			0:00			
Sunday	10/13/2019			0:00			
Monday	10/14/2019	2:00 PM	6:00 PM	4:00	Meeting, advisor meeting	6	4:00
Tuesday	10/15/2019			0:00			
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019			0:00			
Friday	10/18/2019			0:00			
Saturday	10/19/2019			0:00			
Sunday	10/20/2019			0:00			
Monday	10/21/2019	2:00 PM	6:00 PM	4:00	Meeting, advisor meeting	7	7:30
Tuesday	10/22/2019	8:00 AM	9:00 AM	1:00	Video meeting with Ryan		
Wednesday	10/23/2019	1:00 PM	3:30 PM	2:30	Work on drum concept calcs and Gantt chart		
Thursday	10/24/2019			0:00			
Friday	10/25/2019			0:00			
Saturday	10/26/2019			0:00			
Sunday	10/27/2019			0:00			
Monday	10/28/2019	2:00 PM	6:00 PM	4:00	Meeting, advisor meeting	8	28:00
Tuesday	10/29/2019			0:00			
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019			0:00			
Friday	11/1/2019	12:00 PM	5:00 PM	5:00	Meeting, worked on phase 2 report		
Saturday	11/2/2019	2:00 PM	8:00 PM	6:00	Drum concept SW		
Sunday	11/3/2019	10:00 AM	11:00 PM	13:00	Phase 2 report work session		
Monday	11/4/2019	2:00 PM	5:00 PM	3:00	Team meeting and phase 2 report final review		

Figure E.2.3: Timesheet for Alexander Rodd

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Day	Date	Start	End	Duration (H:mm)	Task Comments/ Description	Week	Weekly Total (H:mm)
				Phase I			
Monday	9/9/2019	2:00 PM	3:00 PM	1:00	First Team Meeting and Letter of Intent Discussion		
Tuesday	9/10/2019			0:00			
Wednesday	9/11/2019			0:00		1	5:00
Thursday	9/12/2019			0:00			
Friday	9/13/2019	4:00 PM	5:30 PM	1:30	Researching preliminary designs and determing questions to ask client		
Saturday	9/14/2019	12:00 PM	1:00 PM	1:00	General research		
Sunday	9/15/2019	2:00 PM	3:30 PM	1:30	Preliminary standards and patent search		
Monday	9/16/2019	2:00 PM	6:00 PM	4:00	Group meeting, client kickoff meeting, faculty advisor kickoff meeting		
Tuesday	9/17/2019			0:00			
Wednesday	9/18/2019			0:00			
Thursday	9/19/2019			0:00		2	8:00
Friday	9/20/2019	6:00 PM	6:30 PM	0:30	Phase 1 report		
Saturday	9/21/2019	4:00 PM	6:00 PM	2:00	Phase 1 report - design specification matrix		
Sunday	9/22/2019	6:00 PM	7:30 PM	1:30	Phase 1 report - design specification matrix		
Monday	9/23/2019	2:00 PM	6:00 PM	4:00	Group meeting - working on phase 1 report; advisor meeting		
Tuesday	9/24/2019	8:00 PM	9:30 PM	1:30	Phase 1 report reviewing and editing		
Wednesday	9/25/2019			0:00			
Thursday	9/26/2019			0:00		3	6:30
Friday	9/27/2019			0:00			
Saturday	9/28/2019	9:30 PM	10:00 PM	0:30	e 1 report reviewing and editing (performed from 5:30am to 6:00am in Norway, inputted time in M		
Sunday	9/29/2019	2:00 PM	2:30 PM	0:30	e 1 report reviewing and editing (performed from 10pm to 10:30pm in Norway, inputted time in M		
Monday	9/30/2019			0:00			
				Phase II			
Tuesday	10/1/2019			0:00			
Wednesday	10/2/2019			0:00			
Thursday	10/3/2019			0:00		4	0:00
Friday	10/4/2019			0:00			
Saturday	10/5/2019			0:00			
Sunday	10/6/2019			0:00			
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Team meeting		
Tuesday	10/8/2019			0:00			
Wednesday	10/9/2019			0:00			
Thursday	10/10/2019			0:00		5	7:30
Friday	10/11/2019	4:30 PM	6:00 PM	1:30	Working on concept design drawing and description		
Saturday	10/12/2019			0:00			
Sunday	10/13/2019	9:30 AM	12:30 PM	3:00	Team meeting		
Monday	10/14/2019			0:00			
Tuesday	10/15/2019			0:00			
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019	4:30 PM	5:30 PM	1:00	Meeting with Dr. Chen	6	4:00
Friday	10/18/2019	6:45 PM	8:00 PM	1:15	Design concept analysis and calculations		
Saturday	10/19/2019			0:00			
Sunday	10/20/2019	5:15 PM	7:00 PM	1:45	Design concept analysis and calculations		
Monday	10/21/2019	2:00 PM	5:00 PM	3:00	Team meeting and meeting with Dr. Chen		
Tuesday	10/22/2019	8:00 AM	9:00 AM	1:00	Video meeting with Ryan		
Wednesday	10/23/2019	1:00 PM	5:00 PM	4:00	gn concept calculations and Gantt Chart updates (actual time was 1:00pm-3:30pm & 7:30pm-9:00		
Thursday	10/24/2019			0:00		7	10:00
Friday	10/25/2019	6:30 PM	8:30 PM	2:00	SMath calculations		
Saturday	10/26/2019			0:00			
Sunday	10/27/2019			0:00			
Monday	10/28/2019	1:00 PM	5:00 PM	4:00	Meeting with Dan from machine shop, team meeting, meeting with Dr. Chen		
Tuesday	10/29/2019			0:00			
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019			0:00		8	24:30
Friday	11/1/2019	1:30 PM	7:30 PM	6:00	Team meeting and Phase 2 report work session		
Saturday	11/2/2019	9:30 PM	11:00 PM	1:30			
Sunday	11/3/2019	9:30 AM	10:30 PM	13:00	Phase 2 report work session (actual time was 9:30am-8:30pm, 11:00pm-1:00am)		
Monday	11/4/2019	1:00 PM	5:00 PM	4:00	Team meeting and phase 2 report final review		

Figure E.2.4: Timesheet for Devon Saroya

Day	Date	Start	End	Duration (H:mm)	Task Comments/ Description	Week	Weekly Total (H:mm)
				Phase I			
Monday	9/9/2019	7:15 AM	9:00	1:45	Letter of Intent		
Tuesday	9/10/2019			0:00			
Wednesday	9/11/2019	7:30 PM	20:00	0:30	Email to Bee Project Groups	1	4:45
Thursday	9/12/2019	10:00	11:00	1:00	Research for Group Library Assignment + PDF combining and submitting		
Friday	9/13/2019			0:00			
Saturday	9/14/2019			0:00			
Sunday	9/15/2019	7:30 PM	9:00 PM	1:30	General Research		
Monday	9/16/2019	14:00	17:30	3:30	Group Meeting + Client Kickoff Meeting + Advisor Meeting		
Tuesday	9/17/2019	11:00 AM	12:15 PM	1:15	Phase 1 Report - Market Analysis		
Wednesday	9/18/2019	11:45 AM	12:15 PM	0:30	Market Reseach + Competition Price Quoting		
Thursday	9/19/2019	11:15 AM	12:45 PM	1:30	Phase 1 Report - Market Analysis	2	6:45
Friday	9/20/2019			0:00			
Saturday	9/21/2019			0:00			
Sunday	9/22/2019			0:00			
Monday	9/23/2019	1:30 PM	6:00 PM	4:30	Cover Letter + Group Meeting		
Tuesday	9/24/2019			0:00			
Wednesday	9/25/2019	11:00 AM	1:30 PM	2:30	Phase 1 Report + Market Research		
Thursday	9/26/2019	11:00 PM	11:15 PM	0:15	Calling Australia for Market Prices, Mate	3	16:15
Friday	9/27/2019	11:30	12:30 PM	1:00	Phase 1 Report		
Saturday	9/28/2019	3:30 PM	8:15 PM	4:45	Phase 1 Report		
Sunday	9/29/2019	5:30 PM	8:45 PM	3:15	Phase 1 Report		
Monday	9/30/2019	2:00 PM	5:00 PM	3:00	Group Meeting		
				Phase II			
Tuesday	10/1/2019			0:00			
Wednesday	10/2/2019	9:00 PM	10:00 PM	1:00	Idea Generation		
Thursday	10/3/2019	18:00	9:00 PM	3:00	Idea Generation	4	9:00
Friday	10/4/2019	8:00 PM	10:00 PM	2:00	Idea Generation		
Saturday	10/5/2019			0:00			
Sunday	10/6/2019			0:00			
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Team Meeting		
Tuesday	10/8/2019	8:00 PM	9:00 PM	1:00	Concept Sketches		
Wednesday	10/9/2019	9:00 PM	10:00 PM	1:00	Concept Sketches		
Thursday	10/10/2019			0:00		5	8:15
Friday	10/11/2019	11:45 AM	12:00 PM	0:15	Concept Description		
Saturday	10/12/2019			0:00			
Sunday	10/13/2019	9:30 AM	12:30 PM	3:00	Team Meeting		
Monday	10/14/2019			0:00			
Tuesday	10/15/2019	11:00	12:30 PM	1:30	Concept Calculations		
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019			0:00		6	1:30
Friday	10/18/2019			0:00			
Saturday	10/19/2019			0:00			
Sunday	10/20/2019			0:00			
Monday	10/21/2019	2:00 PM	5:00 PM	3:00	Team Meeting		
Tuesday	10/22/2019	2:30 PM	4:45 PM	2:15	Concept Calculations		
Wednesday	10/23/2019	8:00 AM	10:30 AM	2:30	Client Meeting + Concept Calculations		
Thursday	10/24/2019			0:00		7	13:45
Friday	10/25/2019	12:00 PM	1:00 PM	1:00	Concept Calculations		
Saturday	10/26/2019	1:00 PM	6:00 PM	5:00	Concept Calculations		
Sunday	10/27/2019			0:00			
Monday	10/28/2019	2:00 PM	5:00 PM	3:00	Team Meeting		
Tuesday	10/29/2019	11:15 AM	12:15 PM	1:00	Concept Calculations		
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019	7:30 PM	10:15 PM	2:45	Concept Calculations	8	27:30
Friday	11/1/2019	11:30 AM	6:00 PM	6:30	Phase 2 Report		
Saturday	11/2/2019	6:00 PM	21:00	3:00	Phase 2 Report		
Sunday	11/3/2019	9:30 AM	8:45 PM	11:15			
Monday	11/4/2019	2:00 PM	5:00 PM	3:00			

Figure E.2.5: Timesheet for Jorrell Serrano

Day	Date	Start	End	Duration (H:mm)	Task Comments/ Description	Week	Weekly Total (H:mm)
				Phase I			
Monday	9/9/2019			0:00		1	3:56
Tuesday	9/10/2019			0:00			
Wednesday	9/11/2019			0:00			
Thursday	9/12/2019			0:00			
Friday	9/13/2019			0:00			
Saturday	9/14/2019	10:00 PM	11:00 PM	1:00	Design Sepcification Matrix		
Sunday	9/15/2019	2:00 PM	4:56 PM	2:56	Design Sepcification Matrix		
Monday	9/16/2019			0:00		2	3:56
Tuesday	9/17/2019			0:00			
Wednesday	9/18/2019			0:00			
Thursday	9/19/2019			0:00			
Friday	9/20/2019			0:00			
Saturday	9/21/2019	10:00 PM	11:00 PM	1:00			
Sunday	9/22/2019	2:00 PM	4:56 PM	2:56	Report Writing		
Monday	9/23/2019	2:00 PM	6:00 PM	4:00	Group Meeting	3	5:41
Tuesday	9/24/2019	9:00 PM	10:41 PM	1:41	Safety Factor Calculation		
Wednesday	9/25/2019			0:00			
Thursday	9/26/2019			0:00			
Friday	9/27/2019			0:00			
Saturday	9/28/2019			0:00			
Sunday	9/29/2019			0:00			
Monday	9/30/2019	2:00 PM	5:00 PM	3:00	Group Meeting		
				Phase II			
Tuesday	10/1/2019			0:00		4	3:00
Wednesday	10/2/2019			0:00			
Thursday	10/3/2019			0:00			
Friday	10/4/2019			0:00			
Saturday	10/5/2019			0:00			
Sunday	10/6/2019			0:00			
Monday	10/7/2019	2:00 PM	5:00 PM	3:00	Group Meetting	5	8:00
Tuesday	10/8/2019			0:00			
Wednesday	10/9/2019			0:00			
Thursday	10/10/2019			0:00			
Friday	10/11/2019			0:00			
Saturday	10/12/2019	1:00 PM	6:00 PM	5:00	Solidwork Modelling		
Sunday	10/13/2019			0:00			
Monday	10/14/2019			0:00		6	5:30
Tuesday	10/15/2019			0:00			
Wednesday	10/16/2019			0:00			
Thursday	10/17/2019	4:30 PM	6:00 PM	1:30	Group Meeting		
Friday	10/18/2019			0:00			
Saturday	10/19/2019			0:00			
Sunday	10/20/2019	8:00 PM	12:00 AM	4:00	Preliminary Calculation		
Monday	10/21/2019	2:00 PM	5:00 PM	3:00	Group Meeting	7	7:30
Tuesday	10/22/2019	12:30 PM	2:00 PM	1:30	Preliminary Calculation		
Wednesday	10/23/2019			0:00			
Thursday	10/24/2019			0:00			
Friday	10/25/2019	7:00 PM	10:00 PM	3:00	V2 Of Tilt Model		
Saturday	10/26/2019			0:00			
Sunday	10/27/2019			0:00			
Monday	10/28/2019			0:00		8	19:49
Tuesday	10/29/2019			0:00			
Wednesday	10/30/2019			0:00			
Thursday	10/31/2019	7:00 PM	10:16 PM	3:16	Revise of SW		
Friday	11/1/2019			0:00			
Saturday	11/2/2019	1:00 PM	6:23 PM	5:23	Revise Of SW Tilt		
Sunday	11/3/2019	9:30 AM	8:40 PM	11:10	Revise Of SW Tilt		
Monday	11/4/2019	2:00 PM	5:00 PM	3:00			

Figure E.2.6: Timesheet for Zhe Lyu

D.3: Meeting Minutes

Meeting Minutes were taken during team meetings to document discussions and work that occurred. Important information received during meetings with our client and faculty advisor were also recorded in the meeting minutes to document it for later reference. Action items were also set with deadlines to hold team members accountable to assigned tasks. Copies of the Meeting Minutes documents are provided below.

Group 2 Team Meeting

Minutes SEP. 16, 2019 2:00-5:30 DICE 1-240

Meeting Called By	Devon
Type of meeting	Team meeting, client kickoff meeting, and faculty advisor kickoff meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Adrian Phiri, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Aditya Jain, Connie Philips, Zengtao Chen

Agenda topics

1 HOUR CLIENT MEETING PREPARATION ALL

Discussion	Prepared for kickoff meeting with client. Discussed questions to ask client and brainstormed ideas to present to client.
Conclusions	Final list of questions to ask client during kickoff meeting

1 HOUR CLIENT KICKOFF MEETING ALL

Discussion	Introductory meeting with client, Connie Philips. Defined project scope and asked questions to clarify our responsibilities.
Discussed IP provision.	
Conclusions	Good discussion and clarification on some aspects of the project scope. Some grey areas of project definition that need to
be filled in by speaking to the beekeeper, Ryan.	

Action items	Person responsible	Deadline
Send contact info for beekeeper, Ryan, and Bee Made packing house.	Connie Philips	Sep. 18, 2019

1 HOUR TASK IDENTIFICATION AND ASSIGNMENT ALL

Discussion	Assigned tasks to all team members to complete for the phase 1 report (specified in document on drive called “Phase 1
Role Assignments”. Also prepared for kick off meeting with faculty advisor.	
Conclusions	Everyone to complete 75% of their assigned tasks by next Monday (Sep. 23) so we have time to look at each other’s work
and make changes as needed. Created final list of questions and topics to discuss with faculty advisor.	

Action items	Person responsible	Deadline
Rules and responsibilities section of report and Gantt chart will be done as team	All	Sep. 23, 2019
Complete a large portion of assigned tasks for the Phase 1 report for the next team meeting on Monday	All	Sep. 23, 2019

0.5 HOUR FACULTY ADVISOR KICKOFF MEETING ALL

Discussion	Met with faculty advisor, Zengtao Chen. Introduced project scope. Discussed Dr. Chen’s background and extent of how
he will be able to advise us.	

APIS CONSULTING

Conclusions	Set up weekly meeting with Dr. Chen at 4:30 on Mondays. He set out expectations for us during meetings with him.

Group 2 Team Meeting

Minutes SEPTEMBER 30TH 2019 2:00 PM - 5:00 PM DICE 1-240

Meeting Called By	Apis
Type of meeting	General
Note taker	Adrian Phiri
Timekeeper	Adrian Phiri
Attendees	Adrian Phiri, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Aditya Jain, Zengtao Chen

Agenda topics

2. hours 20 Minutes Phase 1 Report Finalization

Discussion	Reviewed Phase 1 Report for final submission to client and on eclass
Conclusions	Added additional graphs and figures, fixed in text citations, worked on grammar and word choice.

Action items	Person responsible	Deadline
Submit Report	Jorrell	5:00 PM
Send to Clients	Alex	Oct 1st

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Group 2 Team Meeting

Minutes OCT. 7, 2019 2:00-5:00 DICE 1-240

Meeting Called By	APIS CONSULTING
Type of meeting	Team meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Adrian Phiri, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Aditya Jain

Agenda topics

3 HOURS Phase 2 Sticky Note Brainstorming ALL

Discussion	Carried out brainstorming activity where each member wrote down several ideas for the machine on sticky notes and then
all sticky notes were put up on the wall. The team went through each sticky note and discussed everyone's ideas and sorted them	
based on a few categories. Then went through each idea in depth and discussed how it could be implemented.	
Conclusions	Narrowed down our ideas into 3 concepts (stationary drum device, forklift-based device, and tilter) that we will move
forward with in Phase 2 and present to the client, Ryan, for review. Everyone paired up to create preliminary isometric drawings with	
descriptions and preliminary solid models to present next team meeting. Will talk more about analysis for the concepts next team meeting.	

Action items	Person responsible	Deadline
Email Ryan about setting up a meeting to take place within the next week and a half	Alex	Oct. 8
Email Ryan about verifying the direction of our concepts	Alex	Oct. 14
Detailed preliminary drawings/solid models of chosen concepts, and descriptions	Aditya & Jorrell, Alex & Devon, Adrian & Zhe	Oct. 14
Upload pictures of sticky note brainstorming exercise to drive	Jorrell	Oct. 8
Update Dr. Chen about our meeting today - send meeting minutes and picture of sticky notes	Alex	Oct. 8
Check building hours for next Monday (Thanksgiving)	Devon	Oct. 8

Group 2 Team Meeting

Minutes OCT. 13, 2019 9:30-12:00 DICE 1-240

Meeting Called By	Apis Consulting
Type of meeting	Team meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Adrian Phiri, Jorrell Serrano, Alexander Rodd, Zhe Lyu

Agenda topics

1 HOUR Phase 1 Graded Report Review DEVON, ADRIAN, JORRELL, ZHE

Discussion Discussed comments left by the grader for our Phase 1 report. Identified areas of improvement and things to keep in mind for Phase 2 and 3 reports (summarized below).

Conclusions Keep grader's comments in mind when writing phase 2,3 reports. Lessons learned will help us improve our future submissions.

Action items	Person responsible	Deadline
Set meeting for Sunday before Phase 2 submission to put together report and review everyone's sections for cohesiveness	Devon	Nov. 3
Enter time sheet info!!	Aditya, Alex	Oct. 15
Alex to CC us on all future communications with clients, advisor, course instructor	Alex	ongoing

Things to keep in mind based on comments from Phase 1 report:

- ensure to introduce each subsection with a sentence or two
- be very explicit in intent, selection, etc.
  - explicitly explain why we did what we did
- table headings go above tables (figure headings go below figures)
- no need to say "figure is below"
- don't put key parts of the report in the Appendix (eg. project management section)
- detailed team review of the report on the Sunday before submission to make sure everything is cohesive
- explicitly mention key takeaways from each figure, table, calculation etc
- ensure timesheet info is up to date so we don't lose marks for project management section again
- specify tote selection at beginning of report and why we chose that specific one (client requirements, dimensions, food grade)
- Roman numerals for table of contents. Report introduction should start on pg 1
- Appendices should start on a new page

1.5 HOURS Design Concept Calculation Brainstorming DEVON, ADRIAN, JORRELL, ZHE, ALEX

Discussion Brainstormed the calculations we should carry out for each design concept. Narrowed down the calculations we decided are necessary for the Phase 2 report.

Conclusions Divided up the calculations among team members to perform for the next team meeting (Monday, Oct. 21). Calculations split between pairs for each concept design. Will review calculations next team meeting and present to Dr. Chen.

Action items	Person responsible	Deadline
Design calculations in SMATH for each design concept	All	Oct. 21, 2019
Email to Ryan presenting our 3 concept designs - summaries of each design, use case of each design (tilt, rotate, etc), a few pictures of each design to clearly convey the design.	All	Oct. 15, 2019

Group 2 Team Meeting

Minutes OCT. 21, 2019 2:00-5:00 DICE 1-240

Meeting Called By	Apis Consulting
Type of meeting	Team meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Adrian Phiri, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Aditya Jain, Dr. Chen

Agenda topics

1 HOUR — Project Management & Updating Gantt Chart — DEVON, ADRIAN, JORRELL, ALEX, LYU, ADITYA

Discussion	Went through each completed item on the Gantt Chart and reflected on our actual timelines for each task. Will be updating Gantt Chart to also show our actual timelines next to our estimated timelines (project management section of the report).
Conclusions	We will be updating the Gantt Chart with actual timelines during each team meeting moving forward. This will allow us to keep on top of project management for the report on a weekly basis.

Action items	Person responsible	Deadline
Update Gantt Chart: include actual timelines, have # of days include weekends, change names to initials, add tasks	Alex, Devon	Oct. 28
Update Gantt Chart for Phase 3 report during team meeting next week (to include in project management section of Phase 2 report)	All	Oct. 28
Reach out to Ryan to set up meeting/video chat	Alex	Oct. 21

1.5 HOURS — Design Concept Calculation Updates — DEVON, ADRIAN, JORRELL, ALEX, LYU, ADITYA

Discussion	Everyone updated the team on the progress of their calculations. Went through the calculations each person has performed and offered suggestions for further analysis.
Conclusions	Will continue working on calculations this week on our concept designs. Also need to meet with Roger Marchand from the machine shop to discuss estimates on manufacturing costs for our designs (need to set meeting this week or early next week).

Action items	Person responsible	Deadline
Finish calculations and revised models	All	Oct. 28
Set up meeting with machine shop for Oct. 28	Alex	Oct. 21
FEA for other 2 concept designs	Aditya	Oct. 28

1 HOUR — Meeting With Dr. Chen — DEVON, ADRIAN, JORRELL, ALEX, LYU, ADITYA, DR. CHEN

Discussion	Updated Dr. Chen on our progress with calculations since our meeting last Thursday. Asked Dr. Chen some questions regarding force analysis on the tilter concept. Asked Dr. Chen about need for FEA for phase 2 report → do we need to include FEA for all concepts if we do it for one of them?
Conclusions	Dr. Chen was unsure about the need to include FEA for all concepts in phase 2 report if we include it for one of them. Safety factor of 2 is sufficient if we are certain of loading conditions and potential shock (use SF of 3 if there are uncertainties of a concept)

Action items	Person responsible	Deadline
Add a mechanism to secure the shaft with a pin/collar/other for the forklift concept	Jorrell, Aditya	Oct. 28

Group 2 Team Meeting

Minutes OCT. 23, 2019 8:00-9:00 DICE 1-250

Meeting Called By	Apis Consulting
Type of meeting	Client video chat meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Jorrell Serrano, Alexander Rodd, Zhe Lyu

Agenda topics

1 HOUR Video Chat Meeting With Client DEVON,, JORRELL, ALEX, LYU

Discussion	Had a skype meeting with the client, Ryan, to present our concepts and get answers to questions we had about the scope of the project.
Conclusions	During our conversation with Ryan, we learned the tote only needs to be flipped when it is empty and the tote can be drained by tilting the forks on the forklift forward. This simplifies our design. We decided with Ryan that the forklift tilter design would be the best design to move forward with in phase 3. Answers from Ryan to our questions are recorded in a document in the Phase 2 folder.

Action items	Person responsible	Deadline
Set up meeting with Ryan to kick off Phase 3 (probably for before reading week)	Alex	Nov. 4
Modify calculations for fork lift concept design to reflect the fact that only an empty tote needs to be flipped.	Jorrell	Oct. 28

APIS CONSULTING

Group 2 Team Meeting

Minutes OCT. 28, 2019 2:00-5:00 DICE 1-240

Meeting Called By	Apis Consulting
Type of meeting	Weekly Team Meeting
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Adrian Phiri, Aditya Jain, Dr. Chen

Agenda topics

1 HOUR — Project Management & Updating Gantt Chart — DEVON, JORRELL, ALEX, LYU, ADRIAN

Discussion Reviewed action items from last week and each person updated the team on their progress for their tasks. Went through each completed item on the Gantt Chart and reflected on our actual timelines for each task. Looked at work that needs to be done for the Phase 2 report and timelines for the next week. Adrian, Alex, and I updated the rest of the team on our meeting with Dan from the machine shop regarding rough cost estimates for our 3 concepts - tilter concept cheapest, drum substantially more expensive to fabricate.

Conclusions Decided to meet as a group on Friday to work on the Phase 2 report together, to have more cohesion in our report this time. We will also meet on Sunday to work on the report and edit it. We will meet during our regular meeting time next Monday to do one last read through of our report before submitting it and then start on Phase 3 work. Need to do research on suppliers for our materials and off the shelf components. Add these costs with estimated shop fabrication costs (found by multiplying estimated duration by shop rate of ~$120) to determine overall cost estimate of each concept.

Action items	Person responsible	Deadline
Set up meeting with Ryan to kick off Phase 3 for Thursday or Friday	Alex	Nov. 4
Book room for meeting on Friday from 1-5	Devon	Oct. 29
FEA for revised 3 concepts	Aditya	Nov. 1
Make presentation during next Monday for meeting with Dr. Duke on Friday	All	Nov. 4

1.5 hour — Concept Calculations Review — DEVON, JORRELL, ALEX, LYU, ADRIAN, ADITYA

Discussion Reviewed the solid models for each concept and the changes we need to make for each one. Reviewed calculations performed for each concept and what changes might need to be made.

Conclusions	All 3 solid models for our designs require some revising. Some more calculations also need to be done to conclude that each design is viable (based on feedback for extent of calculations from Dr. Duke and Mark).

Action items	Person responsible	Deadline
Update concept solid models	Alex, Aditya, Lyu	Nov. 1
Calculation flow chart	All	Nov. 3
Include meeting minutes in Phase 2 report project management section	All	Nov. 3

Group 2 Team Meeting

Minutes

NOV 1, 2019 2:00PM-7:30PM DICE 1-250

Meeting Called By	Apis Consulting
Type of meeting	Report Work Session
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Adrian Phiri, Aditya Jain

Agenda topics

1 HOUR Updates on Progress for Phase 2 Report DEVON, JORRELL, ALEX, LYU, ADRIAN

Discussion	Everyone provided updates on where they are with their work for the Phase 2 report. We discussed what everyone needs to work on for the rest of the meeting today and on Saturday.
Conclusions	Tasks for the Phase 2 report were distributed to team members to be worked on. Task assignment is noted using comments in the Phase 2 report on the google drive.

2.5 HOURS Work Session For Phase 2 Report DEVON, JORRELL, ALEX, LYU, ADRIAN, ADITYA

Discussion	We worked on writing the Phase 2 report together.
Conclusions	We got some good work done on the report and everyone will do some more work on their assigned sections before the meeting on Sunday.

2 HOURS Decision Matrix for Phase 2 DEVON, ADRIAN

Discussion	Devon and Adrian worked on the decision matrix to compare the three concepts for the phase 2 report.
Conclusions	The decision matrix was completed and it was concluded that the Tilter design was the best option to move forward with in Phase 3

Group 2 Team Meeting

Minutes NOV 3, 2019 9:30AM-8:30PM DICE 8-226

Meeting Called By	Apis Consulting
Type of meeting	Report Work Session
Note taker	Devon
Timekeeper	Devon
Attendees	Devon Saroya, Jorrell Serrano, Alexander Rodd, Zhe Lyu, Adrian Phiri

Agenda topics

11 HOURS Work Session For Phase 2 Report DEVON, JORRELL, ALEX, LYU, ADRIAN

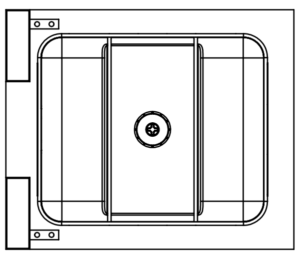
Discussion	We worked on writing the Phase 2 report and rendered SolidWorks models to include in the report. Calculations were finalized and put into the report.
Conclusions	We finished the report and everyone will review it for the meeting tomorrow before submission.

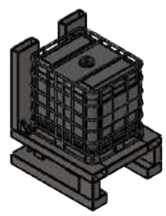
Action items	Person responsible	Deadline
Review course details during meeting tomorrow before submitting report to make sure we're not missing anything.	All	Nov. 4
Update Gantt Chart	All	Nov. 4
Insert Aditya FEA results and a few sentences regarding the results	All	Nov. 4

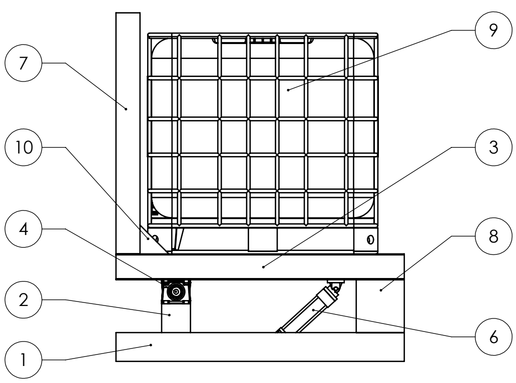
Appendix F: Drawing Packages

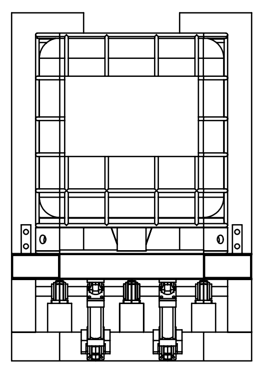
See attached drawing packages for each concept.

ITEM NO.	PART NUMBER	DESCRIPTION	Flat/QTY.
1	Base	This serves as the support for the whole machine	1
2	Base_Bearing_Mouunt	This bearing supports the hinge shaft	4
3	Top_Plate	This plate holds the the tote	1
4	Shaft	The tote rotates about this shaft	1
5	Base Mount	This serves as the support for the base bearing	4
6	Hydralic_Total	This provides the force to tilt the tote	2
7	Front_Plate	This prevents the tote from sliding	2
8	STEEL_Bar_end_Supp	The end support provides extra supprt for the tote at rest	2
9	Tote	This this chosen tote	1
10	Potential_Supprt	This is the potential suppport for the front plate	2









Mec E 460
Instructors: Dr. Kajsa Duke Fall 2019
Comments: This drawing shows each parts of the machine

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH μm 0.6
DO NOT SCALE DRAWING

DRAWN BY: Zhe Lyu
Group name: APIS
Group number: 2
SM By: Zhe Lyu
Reviewed by: Set in this file
Sunday, November 3, 2019 7:01:29 PM
Sunday, November 3, 2019 12:47:45 PM

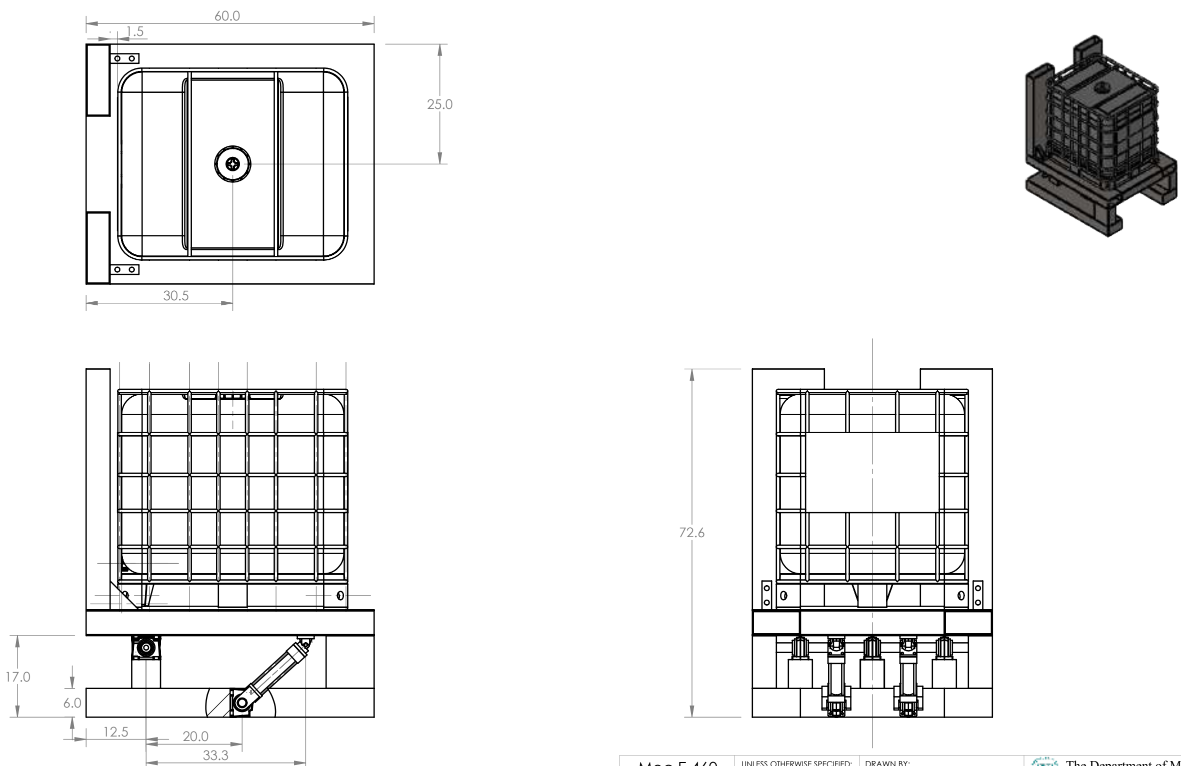
MATERIAL: Mixed
FILE NAME: Total_Assembly_Tote

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

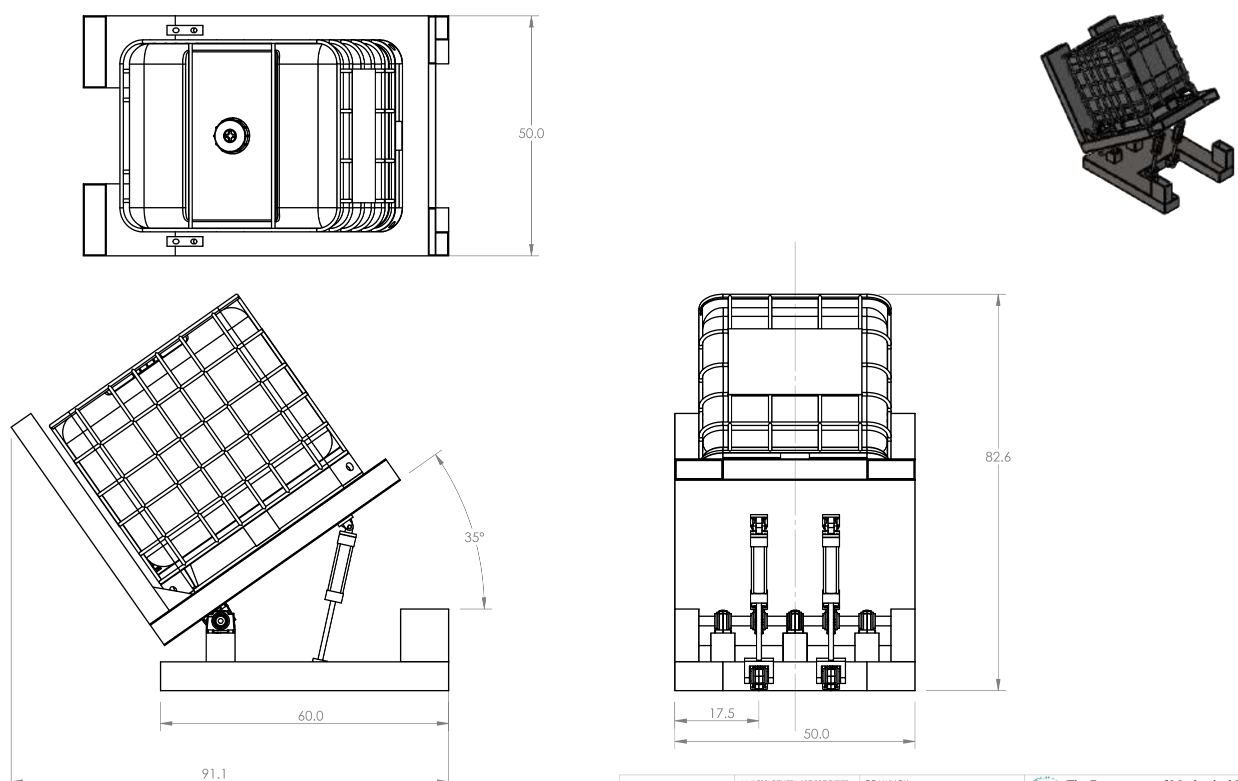
TITLE: The overall machine with ballons

SIZE: B | Part supplier/manufacturer | REV

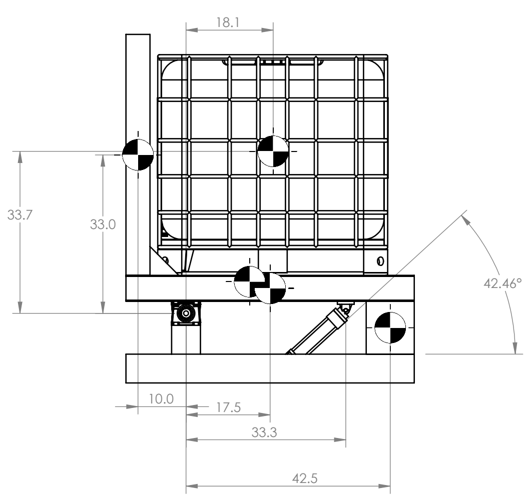
SCALE: 1:50 | Mass: 7338.48 | SHEET 1 OF 6

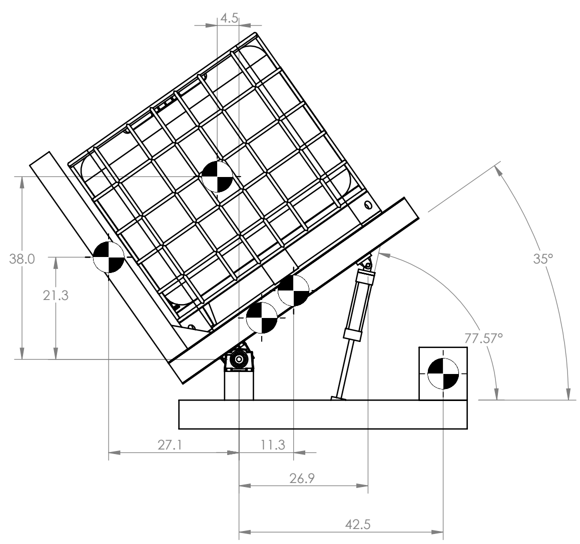


Mec E 460	UNLESS OTHERWISE SPECIFIED:	DRAWN BY: Zhe Lyu	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA
Instructors: Dr. Kajsa Duke Fall 2019	DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025	Group name: APIS	TITLE: Overall Dimension At flat
Comments: This Drawing shows the overall dimension of the machine at rest		Group number: 2	
	SURFACE FINISH 0.6 µm	SM By: Zhe Lyu	SIZE B — Part supplier/manufacturer — REV
	DO NOT SCALE DRAWING	Reviewed by: Set in this file	
MATERIAL: Mixed		Sunday, November 3, 2019 7:01:29 PM Sunday, November 3, 2019 12:47:45 PM	
FILE NAME: Total_Assembly_Tote			SCALE: 1:50 — Mass: 7338.48 — SHEET 2 OF 6

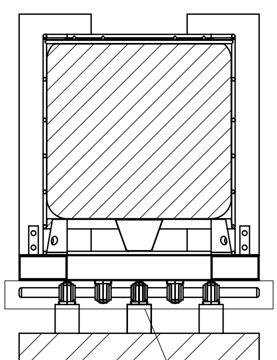


Mec E 460 Instructors: Dr. Kajsa Duke Fall 2019 Comments: This drawing shows the overall dimsension when the tote is tilted at max angle	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH 0.6 µm DO NOT SCALE DRAWING	DRAWN BY: Zhe Lyu Group name: APIS Group number: 2 SM By: Zhe Lyu Reviewed by: Set in this file	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Tote Tilted
MATERIAL: Mixed FILE NAME: Total_Assembly_Tote		Sunday, November 3, 2019 7:01:29 PM Sunday, November 3, 2019 12:47:45 PM	SIZE B · Part supplier/manufacturer · REV SCALE: 1:50 · Mass: 7338.48 · SHEET 3 OF 6

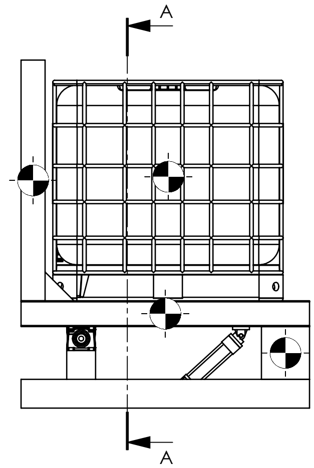


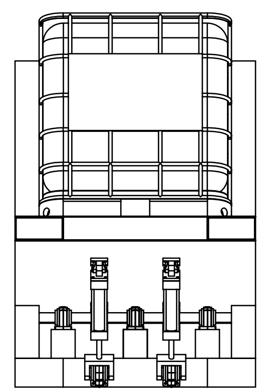


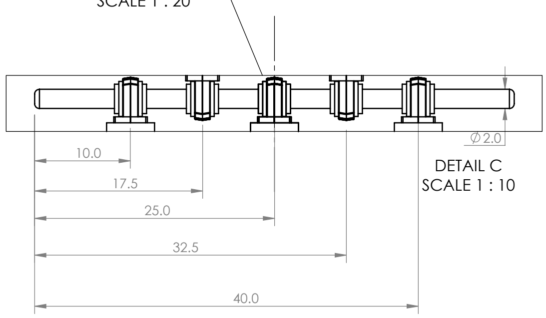
Mec E 460 Instructors: Dr. Kajsa Duke Fall 2013 Comments: This drawing shows the relative distance of COGs to the shaft hinge	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH 0.6 µm DO NOT SCALE DRAWING	DRAWN BY: Zhe Lyu Group name: APIS Group number: 2 SM By: Zhe Lyu Reviewed by: Set in this file	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: The flat and tilted position
MATERIAL: Mixed FILE NAME: Total_Assembly_Tote		Sunday, November 3, 2019 7:01:29 PM Sunday, November 3, 2019 12:47:45 PM	SIZE B · Part supplier/manufacturer · REV SCALE: 1:50 · Mass: 7338.48 · SHEET 4 OF 6



SECTION A-A
SCALE 1 : 20

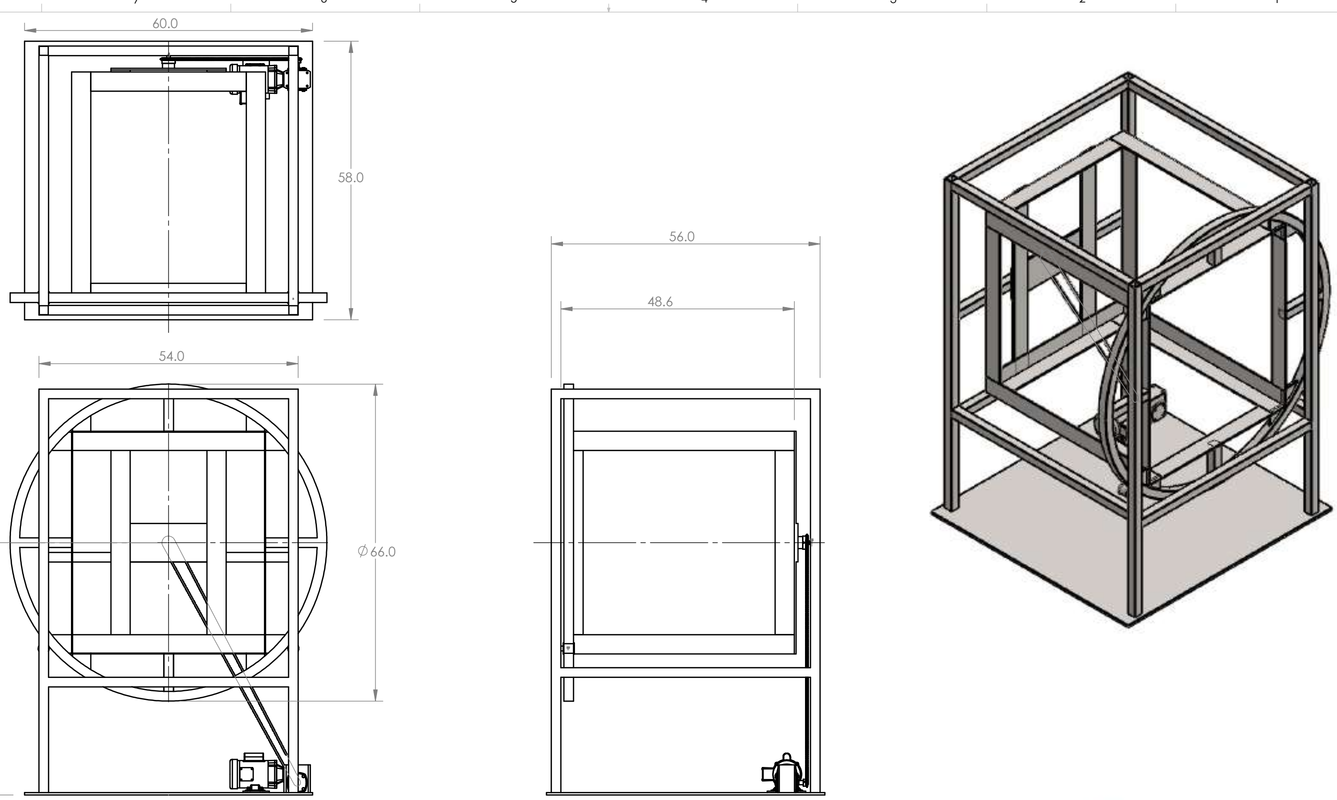






DETAIL C
SCALE 1 : 10

Mec E 460 Instructors: Dr. Kajsa Duke Fall 2013 Comments: This drawing shows the relative location of each bearing on the hinge shaft	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH 0.6 µm DO NOT SCALE DRAWING	DRAWN BY: Zhe Lyu Group name: APIS Group number: 2 SM By: Zhe Lyu Reviewed by: Set in this file	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Bearing location on the Hinge Shaft
MATERIAL: Mixed FILE NAME: Total_Assembly_Tote		Sunday, November 3, 2019 7:01:29 PM Sunday, November 3, 2019 12:47:45 PM	SIZE B | Part supplier/manufacturer | REV SCALE: 1:50 | Mass: 7338.48 | SHEET 5 OF 6



MEC E 460

Instructor:
Dr Kajsa Duke

Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH 0.6 µm

DO NOT SCALE DRAWING

MATERIAL:
Plain Carbon Steel

FILE NAME:
drum assembly 2a

DRAWN BY:
Alex Rodd

Group Number: 2

SM By: **Alex Rodd**

Group Name: Apis Consulting

rodd
Monday, November 4, 2019 3:43:54 PM
Saturday, November 2, 2019 3:12:20 PM

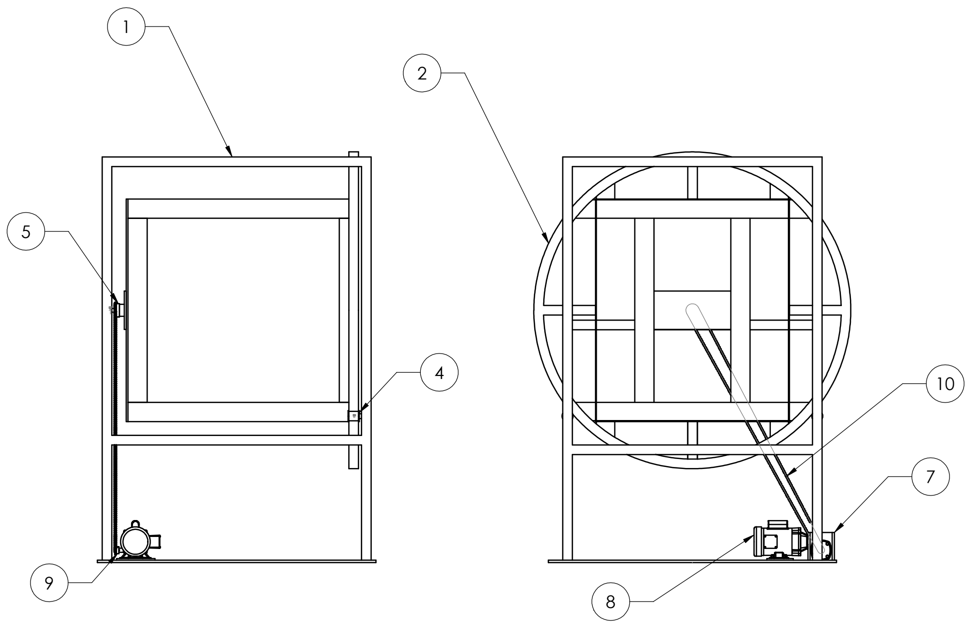
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Rotating Frame Tote Inverter

SIZE: **B**
Assignment Number
REV: -

SCALE: 1:20 Mass: 448.1 kg SHEET 1 OF 2

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Base	Main frame	1
2	Subframe	Rotating frame	1
N/S	6383K120	Ball bearings (guide rollers)	2
4	Guide	Guide roller	2
5	6280K582	Driven sprocket	1
N/S	6383K520	Ball bearing (driveshaft)	1
7	5887K221	Reduction gearbox	1
8	59845K110	Motor	1
9	6280K447	Motor sprocket	1
10	6261K174	Drive chain	-



MEC E 460

Instructor: Dr Kajsa Duke

Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

DRAWN BY: **Alex Rodd**

Group Number	2
SM By	**Alex Rodd**
Group Name	Apis Consulting

rodd
Monday, November 4, 2019 3:43:54 PM
Saturday, November 2, 2019 3:12:20 PM

MATERIAL: Plain Carbon Steel

FILE NAME: drum assembly 2a

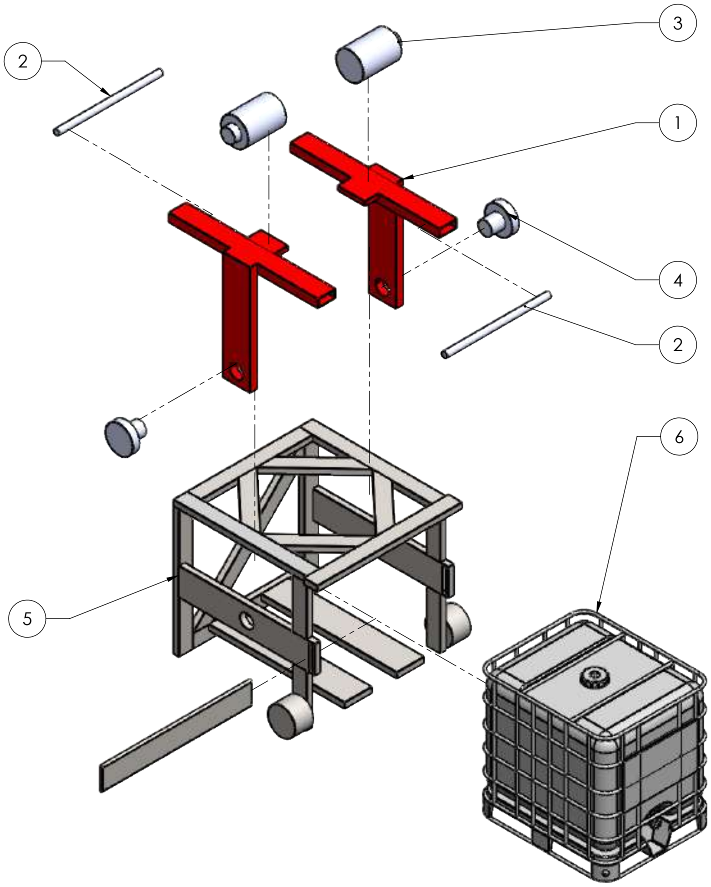
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Rotating Frame Tote Inverter

SIZE **B** | Assignment Number | REV -

SCALE: 1:20 | Mass: 448.1 kg | SHEET 2 OF 2

ITEM NO.	PART NUMBER	Material	QTY.
1	arm	AISI 1020	2
2	Supporting Beam	AISI 1020	2
3	motor	-	2
4	rotator bottom	AISI 1020	2
5	cage	AISI 1020	1
6	Tote	-	1



MecE 460
Instructor: Dr. Duke
Fall 2019
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

MATERIAL: Various
FILE NAME: Forklift_Attachment

DRAWN BY: Aditya Jain
Student # 1425827
SM By Aditya Jain
Group Memebers Alex Rodd, Devon Saroya, Jorrell Serrano, Adrian Phiri, Zhe Lyu

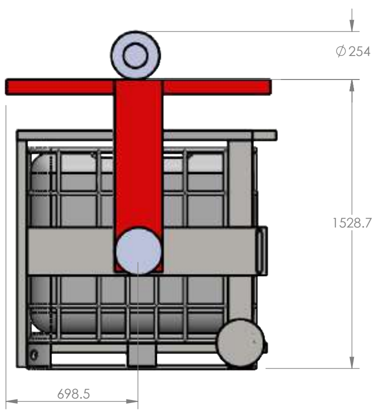
Saturday, November 2, 2019 1:39:14 AM
Friday, October 11, 2019 4:43:04 PM

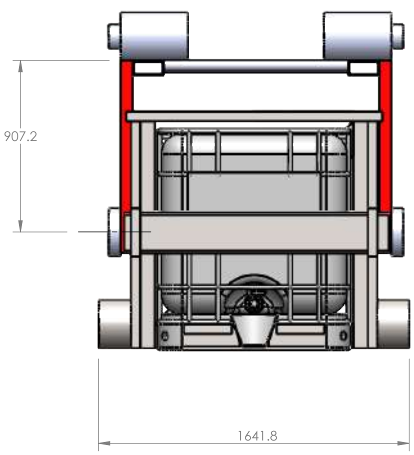
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

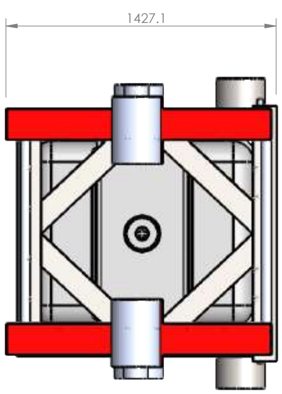
TITLE:
Forklift Attachment Assembly

Group 2
APIS Consulting
REV A

SCALE: 1:25
SHEET 1 OF 2







MecE 460 Instructor: Dr. Duke, Fall 2019 Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Aditya Jain Student # 1425827 SM By Aditya Jain Group Memebers: Alex Rodd, Devon Saroya, Jorrell Serrano, Adrian Phiri, Zhe Lyu	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Forklift Attachment Assembly
MATERIAL: Various		Saturday, November 2, 2019 1:39:14 AM Friday, October 11, 2019 4:43:04 PM	Group 2 — APIS Consulting — REV A
FILE NAME: Forklift_Attachment			SCALE: 1:25 — SHEET 2 OF 2