Capstone Design Report #2
Drywall Cart Redesign

ME 4182 – A

3/16/2022

Instructor: Dr. Jariwala

Faculty Advisor: Dr. Green

Team "The Good Guys"

Ryan Grajewski, Will Hagler, Graham Brantley, Garrett Rodino

Nischal Bandi

Sponsor: Gypsum Management & Supply, Inc.

Contacts: andrew.horner@gms.com, jeff.johnson@gms.com

Table of Contents

Executive Summary

Drywall is a vital construction material, as it is used in the construction of interior walls and ceilings. However due to its heavy weight, most construction companies hire subcontractors such as GMS (Gypsum Management & Supply Inc) to deliver it to a specified location on the jobsite. These locations can be anywhere from the top floor of an apartment building or the basement of a residential home, so an efficient way of moving the drywall around must be utilized. For the past 40 years, GMS has used a drywall cart from Adapa as their primary tool for transporting drywall around jobsites. However, the unsafe design of this cart has cost GMS close to $1 million annually in work-related injuries, so they would like to explore alternative drywall cart solutions. By working closely with GMS, our team aims to design a new and safer cart design to reduce the number of work-related injuries in the drywall shipping industry.

In order to match the performance of the current drywall cart, our design must safely transport up to sixteen sheets of drywall, weighing up to 3000 lbs, in a variety of construction environments. The main safety concerns with the current design are twofold. First, the cart can easily tip over if it is pushed when its caster wheels are not pointing in the direction of the force applied to it. Second, the cart's caster wheels can often puncture the soft flooring of the jobsite under the immense weight of the drywall, which can lead to the wheels getting stuck and the cart tipping. By going through a standard ideation process, our team generated multiple potential designs to minimize these safety concerns. After using an evaluation matrix to rank our best concepts, a final design involving a two wheel steering mechanism and an automatic kickstand was chosen. The steering mechanism aims to allow the user to point the wheels of the cart in the direction of the applied force, preventing a key tipping mechanism. The kickstand is a last resort failsafe to prevent user injury in the event of the cart tipping. We strongly believe that this design will improve GMS jobsite safety and prevent unnecessary work-related injuries.

Nomenclature

- **A-Frame**: A support structure shaped like the letter A
- **Caster Wheels**: An undriven wheel that allows free rotation about the vertical axis
- **Tipping Moment**: The critical torque value above which the cart will tip over

Glossary

Term	Definition
DFM	Acronym for: Design for Manufacturability
GMS	Acronym for: Gypsum Management & Supply Inc.
MMH	Acronym for: Manual Materials Handling

1. Introduction and Project Background

The use of drywall panels for interior walls is ubiquitous throughout the U.S. construction industry. In 2020 alone, approximately 26 billion ft^2 of wallboard products were sold in the U.S [1], accounting for 95% of all U.S. interior wall construction schemes [2]. This massive demand for drywall has steadily encouraged growth for leaders in the commercial material supply sector such as GMS (Gypsum Management & Supply Inc.), who source and deliver drywall to job sites across the United States. The most labor-intensive aspect of the delivery process is the transportation of the drywall from the delivery truck to the jobsite through the use of a drywall cart. This process is also where the majority of worker-related injuries occur, with GMS reporting roughly 90 drywall cart related injuries per year that total to approximately $1 million in worker's compensation. It is likely that every major drywall supplier in the U.S. incur similar losses as they all use the same drywall cart design, and thus there exists a widespread need for improvements in drywall cart safety.

GMS has sponsored this Capstone Design Project in an effort to pursue a safer drywall cart design. GMS currently employs the industry-standard DC-2020-P drywall cart manufactured by Adapa, shown in Figure 1, which is a rudimentary A-frame cart with four caster wheels. Two operators are required to move this cart due to the heavy weight of the drywall, which they can only do by pushing on the drywall panels itself due to their size (up to 4 ft by 16 ft). The cart must also be narrow in order to navigate in tight indoor spaces, raising its center of mass and making it prone to tipping. Additionally, the weight on the cart can cause its caster wheels to puncture through jobsite flooring or even fail, causing weight imbalances and subsequent tipping. GMS suggests that this tipping is the most common cause of work-place accidents. Thus, our design aims to minimize both the chance and danger that cart tipping poses on jobsite while maintaining the current cart's maneuverability, durability, and efficiency.

The selected design concept, which comprises a modified adapa cart frame with a steering system and emergency kickstand mechanism, has been further developed throughout the ideation, engineering feasibility analysis, and CAD/prototyping phases. Because this project is intensely mechanical, the engineering feasibility testing phase required the most significant efforts to establish validity for the chosen design. Finite element analyses of the adapa cart frame, including the designed modifications, was first conducted in order to establish that those modifications could still support the full load of drywall. Using that FEA as a baseline, machine design calculations for the bevel gears, gearing system, and timing belt were performed in order to ensure that the required input torque could sufficiently translate to the calculated torque required to turn each wheel. Further, stress, buckling, and finite element analysis were conducted to validate the design of the emergency kickstand mechanism. These calculations helped inform the chosen material, geometry, and stress concentrations apparent in the initial design concept. The final result of these efforts is a fully validated prototype model which meets the customer and engineering requirements developed by GMS.

Figure 1: Model DC-2020-P Adapa Drywall Cart currently used by GMS [3]

2. Market Research

Market research was gathered through a variety of approaches. First, interviews were conducted with both GMS risk managers and worksite employees. The risk managers provided the current cart specifications, cart operation liabilities, the size of the drywall shipping market, and desired functions for the new cart design. The worksite employees helped by demonstrating how drywall carts are used onsite and their current functionality. Second, research on the general use and injury cases of carts throughout the construction and material/drywall supply industries helped to better understand the financial and human impacts of drywall cart related injuries. Finally, an anonymous survey was distributed to several drywall and construction worker groups on the internet to get clearer insight into what issues were identified by drywall cart users not affiliated with GMS. This research allowed us to pinpoint exactly what functions our drywall cart should perform.

The next step in our market research process was to fully understand the benefits and limitations of the current cart design. The current Adapa cart design used by GMS, seen in Figure 1, features a slanted A-frame base designed to lower the center of mass of the drywall payload [3] and sells for $650. While Adapa does not monopolize the drywall cart industry, it does currently produce the best drywall cart for construction material supplies on the market. Thus, our design must at minimum meet the Adapa cart's performance in terms of durability, stability, price, and safety. It is important to note that the drywall cart market includes not only construction material supplies like GMS, but also independent contractors throughout the U.S. Additionally, regions such as Western Europe and East Asia are rapidly adopting the use of drywall, so there is potential for our product to compete in future global markets [4].

As stated before, GMS reports roughly $1 million in workers' compensation per year, and we found that other companies in the space reported similar numbers [5]. It is therefore clear that there is a real and present need to produce a safer drywall cart design. Our research also found that roughly 60% of manual materials handling (MMH) claims involved the lower back, upper arms, wrists, hands, or fingers, all of which are key to pushing and tightly maneuvering carts that are heavily loaded with drywall [5]. Additionally, more than 57% of all MMH claims involved strains or sprains caused by workers overexerting their muscles to stabilize the cart [5]. Thus, these injuries are directly correlated with the instability of the current drywall cart design.

3. Relevant Prior Art

There are many different material transport carts currently available on the market, including pallet trucks for transporting heavy loads, platform carts for moving large materials, and A-frame carts for moving material boards. Due to the large number of material transport cart approaches, the scope of this prior art search was not limited to drywall transportation devices. Instead, we focused on identifying any prior art that could be relevant to our goal of improving drywall transportation safety.

3.1 The Classic Drywall Cart

The current market standard for drywall carts, and the drywall cart that defines our baseline performance, is the model manufactured by Adapa shown in Box A of Table 1. This design consists of an angle A-frame wheeled cart with caster wheels. The cart frame is made of welded steel tubing and the platform is covered with Teflon to reduce friction during loading and unloading. This cart benefits from being low cost and easy to manufacture, but it suffers from not having a brake to keep the cart stationary during loading and unloading, from not having a guide rail to ensure product is correctly loading, and from the aforementioned tipping risks.

3.2 Flatbed Carts, Troll Carts, Centered A-Frame Carts

Flatbed, troll, and centered A-frame carts are all iterations on the classic drywall cart that have different base designs. Flatbed carts have a platform parallel to the ground that is located between the caster wheels to lower the center of mass of the cart and improve stability. Troll carts have an H-frame base so that the drywall can rest on two support beams rather than a platform, providing two contact points and allowing the transportation of irregularly shaped items. Centered A-frame carts have a second angle platform, allowing them to centralize their center of mass at the expense of raising it. Examples of these carts can be seen in Boxes B, C, and D of Table 1.

3.3 Swiveling Carts

Swiveling carts allow the user to manipulate the orientation of the drywall on the cart by rotating the payload about multiple axes. This can allow the cart to fit into elevators or other tight spaces. These carts typically have square (Box E Table 1) or X-shaped (Box F Table 1) bases that are attached to a central beam. It is notable that this is the only cart design that allows for customisation of the drywall positioning depending on the situation, resulting in greatly increased maneuverability.

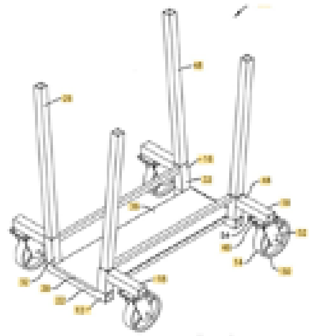
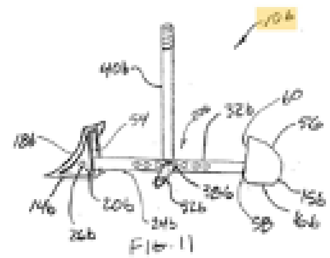
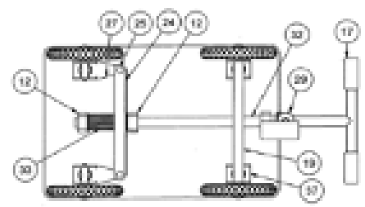
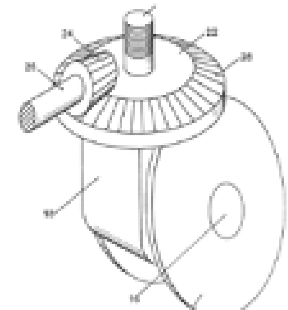
3.4 Wheel Brakes

Keeping a drywall cart stationary makes the loading and unloading processes significantly easier and safer. Some carts use clamp breaks, as shown in Box G of Table 1, to stop the rotational motion of the caster wheels, but this does not stop the casters from swiveling. Another alternative is to use a chock to secure the cart, such as the one integrated into the design shown in Box H of Table 1.

3.5 Steering and Wheels

While the classic drywall cart uses caster wheels to effectively maneuver itself in tight situations, there are some designs that instead use mechanical linkages to control the direction of the wheels. This can be done through a full steering system, such as the one in Box I of Table 1, or through a mechanism that allows for manual control of the caster wheels, such as shown in Box J of Table 1.

Table 1: Prior art designs referenced in the report

 **A** – Adapa Drywall Cart	 **B** – Flatbed Cart	 **C** – Telpro Troll Cart	 **D** – Center A-frame Cart	 **E** – Swivel Elevator Cart
 **F** – Haley Material Handling Cart	 **G** – Caster Wheel Clamp Brake	 **H** – Integrated Wheel Chock	 Fig 5 **I** – Gear Train Steering System	 **J** – Wheel Steering Mechanical Linkage

4. Applicable Codes and Standards

Although there are no explicit codes and/or standards that apply to the production and operation of drywall carts, there are company guidelines concerning the operation of drywall carts that should be considered. The safety and handling guidelines for drywall carts at a typical construction company like Island Acoustics calls for the following [6]:

1. Operating carts with at least 2 workers
2. Not modifying carts
3. Not overloading carts
4. Not using carts in a manner inconsistent with the manufacturer's suggestion
5. Not using damaged carts
6. Not storing carts in non-designated areas or with materials loaded
7. Keeping drywall carts stable and stationary during loading
8. Keeping the center of gravity of the cart low by stacking with lighter ones at the top
9. Securing particularly bulky payloads with straps

These company guidelines do not officially limit the design of drywall carts, but they do clarify best practices for operating such carts, which can help determine features that we may want to consider incorporating, such as foot brakes and side walls.

5. Customer Requirements and Engineering Specifications

5.1 Stakeholder Analysis

In order to determine our customer requirements, it was vital to first determine our primary stakeholders, or the people that are invested in our project. The three main stakeholders for this project were identified: GMS risk managers, GMS worksite employees, and Georgia Tech. Each stakeholder was ranked by their relative importance and influence on the project. Importance was defined as the relative value this project will provide to a given stakeholder, while influence was defined as a stakeholder's native ability to influence the project.

GMS risk managers were found to be a high importance and high influence stakeholder, as not only does this project have the potential to greatly influence their business, but their help will be key in making this project a success. On the other hand, GMS worksite employees were found to have high importance but low influence since our project has the potential to directly impact their work, but they cannot directly impact the direction of the project. Finally, Georgia Tech was found to have low importance but high influence, as our grades directly depend upon Georgia Tech's grading scheme but Georgia Tech is not affected by our project in any way.

5.2 Customer Requirements

After key stakeholders were identified, we settled upon the customer requirements shown in Table 2. This table contains both requirements from the currently used model, such as being able to support the product load and be lifted by one person, and desired improvements, such as not tipping over during use and remaining stationary during loading and unloading.

Table 2: Table of customer requirements sorted by category

Category	Customer Requirement
Function	• Hold and support product load • Move easily over multiple surfaces • Remain stationary during loading and unloading • Navigate through narrow thresholds • Easily lifted by one person
Geometry	• Similar shape to previous design

Cost	● Use manufacturing processes similar to previous design ● Cost similar to previous design
Reliability	● Weatherproof ● Does not tip during use

5.3 Engineering Requirements

Once we had gathered our customer requirements, we tried to tie each of them into a corresponding engineering requirement. This allowed us to quantify how well we met each requirement through a numerical target value. Some key engineering requirements were to have a width less than 30” to fit through door frames, the ability to support a weight of up to 3000 lbs, and for the frame to not plastically deform during use. For more information on the engineering requirements, please see the full chart located in the Appendix as Table A1.

5.4 Engineering Constraints

In addition to all of our engineering requirements, two additional engineering constraints were identified. First, due to the narrow thresholds in many job sites, it was decided that our design must be similar in size to the current cart. Second, in order for our cart to deliver drywall effectively, it must be able to traverse uneven terrain and strange substrates. This also meant that our cart would be prone to wear and must therefore be easily repairable. These constraints were also considered as engineering requirements during the concept evaluation phase of our project.

5.5 House of Quality

The last step in our quest to comprehensively describe our cart’s requirements was to organize our customer and engineering requirements into a House of Quality. Our House of Quality determined that our most important customer requirements were safety, maneuverability, and stability, while our most important engineering requirements were the tipping moment and height of the center of mass of our cart. For more information on our House of Quality, please see the full document located in the Appendix as Figures A1a and A1b.

6. Design Concept Ideation

6.1 Function Tree

The first step in the design concept ideation phase was to define the essential functions that each concept must be capable of fulfilling. In general, it was evident that the final design would need to fulfill 4 main functions: be able to maneuver through tight spaces, be able to easily load and unload drywall, be able to support and secure drywall sheets during transportation, and be able to roll over uneven surfaces. These 4 functions were then broken down in further detail as sub-functions, which can be seen in the function tree shown below in Figure 2.

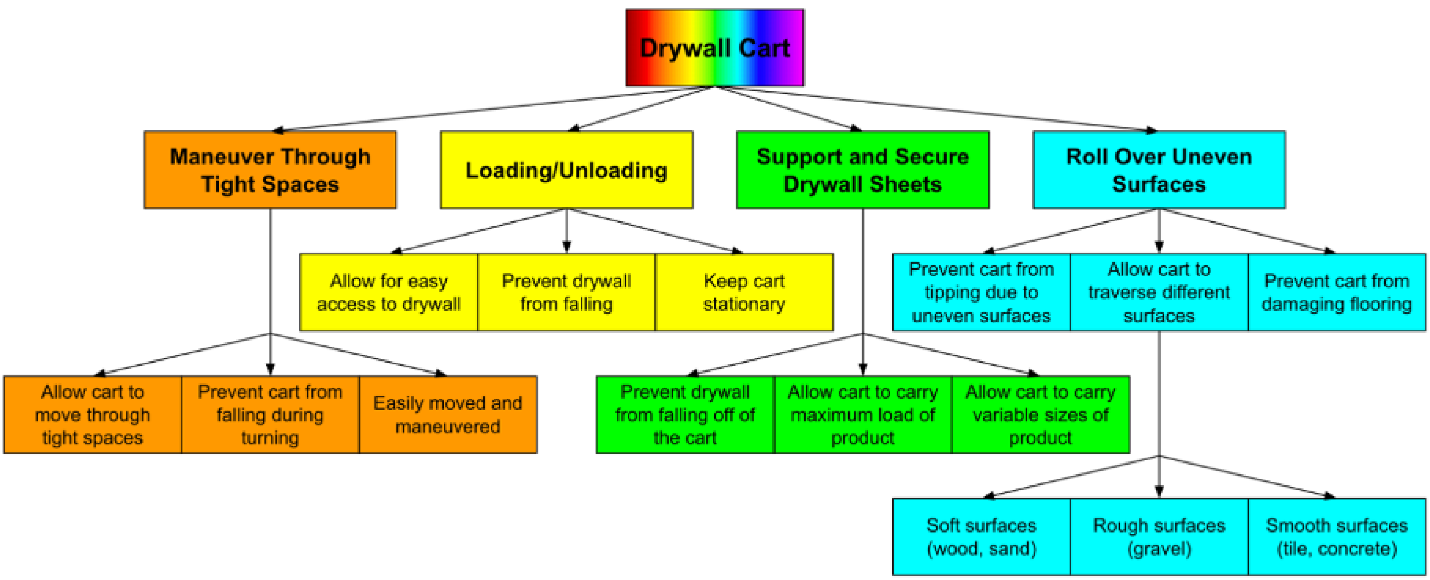


Figure 2: Drywall cart function tree showing 4 main functions and their sub-functions

6.2 Morphological Chart

Once the functionality of the drywall cart had been thoroughly defined, individual concepts or mechanism ideas were generated as solutions to each of the specific sub-functions defined in the function tree. Due to the modular nature of the drywall cart, many of the concepts could be used to satisfy multiple functionalities at once. A morphological chart was then used to compile each individual component idea into sub-function categories for the purpose of facilitating alternative design generation. For more information on our morphological chart, please see the full document listed in the Appendix as Figure A2.

6.3 Full Concept Designs and Evaluation Matrix

Each alternative design was selected in a piecewise fashion, incorporating one or multiple concepts from each sub-function category to result in a full cart design concept. The following five alternative designs were put forward for evaluation:

1. A 2-wheel drive steering cart with an angled base A-frame and an external kickstand
2. A 4-wheel drive steering cart with an angled A-frame base and an internal kickstand
3. A caster-wheel directed cart with an angled A-frame base, suspension, and a footbrake
4. A caster-wheel directed cart with a flatbed frame and a foot pedal brake
5. A caster-wheel cart with an angled A-frame base, internal kickstand, side guards, and footbrake

An evaluation matrix was then used to quantitatively compare each concept against the customer requirements. Each concept was given a score from 1 to 5 for how well it met each user need. To account for the varying importance levels of each user need, scores were weighted accordingly so that a high score in a highly important customer requirement category would have a higher impact. The sums of the scores for each alternative design were calculated and Design #1 was selected. For more information on our evaluation matrix, please see the full chart listed in the Appendix as Table A2.

7. Selected Design Concept and Justification

Our selected design concept, “2-Wheel Steering + External Kickstand”, is shown in Figure 3.

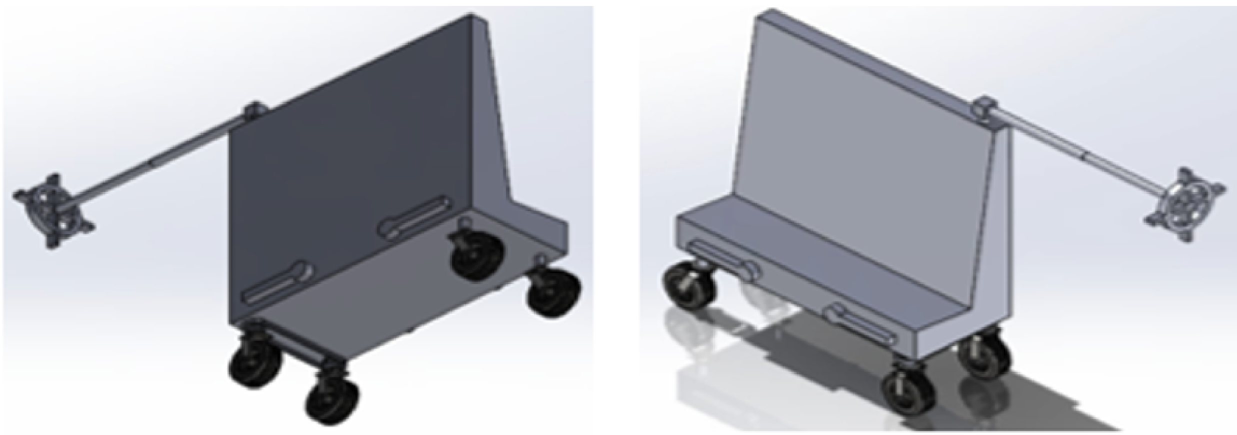


Figure 3: Selected design concept art

This design utilizes a steering system for the front set of wheels to allow the operator to steer the cart without having to apply a horizontal force to the cart, which could possibly tip the cart. Operators will also be able to use the steering to have much greater control over where the cart will go when it is pushed forward. The extended steering column allows the operator to steer the cart while holding the drywall. This design also utilizes an external safety mechanism that will automatically deploy when tipping begins. Once balance is restored, the “kickstand” arms can be quickly moved back into place so the cart can operate normally. The angled base also allows for quick loading and unloading of the material.

This design was selected because it scored highly in many of the important customer requirements, particularly in “Safe to Use”, “Highly Maneuverable”, and “Stable”. However, it did very poorly in the “Easy to Use” requirement due to the higher amount of training and coordination that the steering system requires. For more information on how our selected design performed in the evaluation matrix, please see the full chart listed in the Appendix as Table A2.

8. Engineering Analyses and Experiments

The final CAD model of our design is shown in Figure 4. Each component of the model underwent rigorous theoretical calculations to ensure its structural integrity and validity, as is described below.

Figure 4: Final CAD model of our drywall cart design

Figure 4a: Final CAD model of our drywall cart design

8.1 Frame

The frame of the drywall cart was primarily based on the Adapa cart design, as we chose to use the same tilted A-frame base. However, three key modifications were made in order to fit in the steering system. First, a box was placed on the top right corner of the back support bar in order to house the bevel gear system. Second, the top bar of the back support bar was extended to the right in order to house the steering column. Third, a bar was added to help support the steering wheel and column. Fourth, the angle of the front support bar was decreased to ensure that the drywall did not hit the steering wheel or bevel gear box. Finally, a plate was added to the bottom of the frame in order to allow the gearing system to be directly attached. The material selected for all of these additions was AISI 1010 low carbon steel due to its low cost, good machinability, and good weldability, making it an easy steel to work with.

Although we knew that the original Adapa cart was fully able to support the weight of the maximum drywall payload, all of these design changes could have decreased the strength of our frame. We decided to run two different FEA analyses in order to determine how our frame would react to different kinds of common loads.

The first FEA analysis run was to determine whether or not our frame design could still withstand the weight of 3000 lbs of drywall sheets. This was done by putting our frame into an assembly and placing a 96” x 192” x 8” block on top of it (i.e. where the drywall would ordinarily rest). Plain carbon steel, the closest material that SolidWorks had to AISI 1010 steel, was selected as the frame material, while a custom material had to be made for the block since SolidWorks did not natively have drywall as a material. The density of this custom material was tuned to ensure that the block weighed exactly 3000 lbs. The drywall block was also made a rigid body to prevent its deformation from affecting the frame’s results. The frame was meshed with a blended curvature-based mesh with a maximum element size of 3” and a minimum element size of 0.15”. The only applied load was gravity, with an acceleration of 9.81 m/s^2, and the only constraint was that the bottom of the frame was fixed, simulating its contact with the wheels and therefore the ground. The results of this FEA analysis are shown in Figure 5.

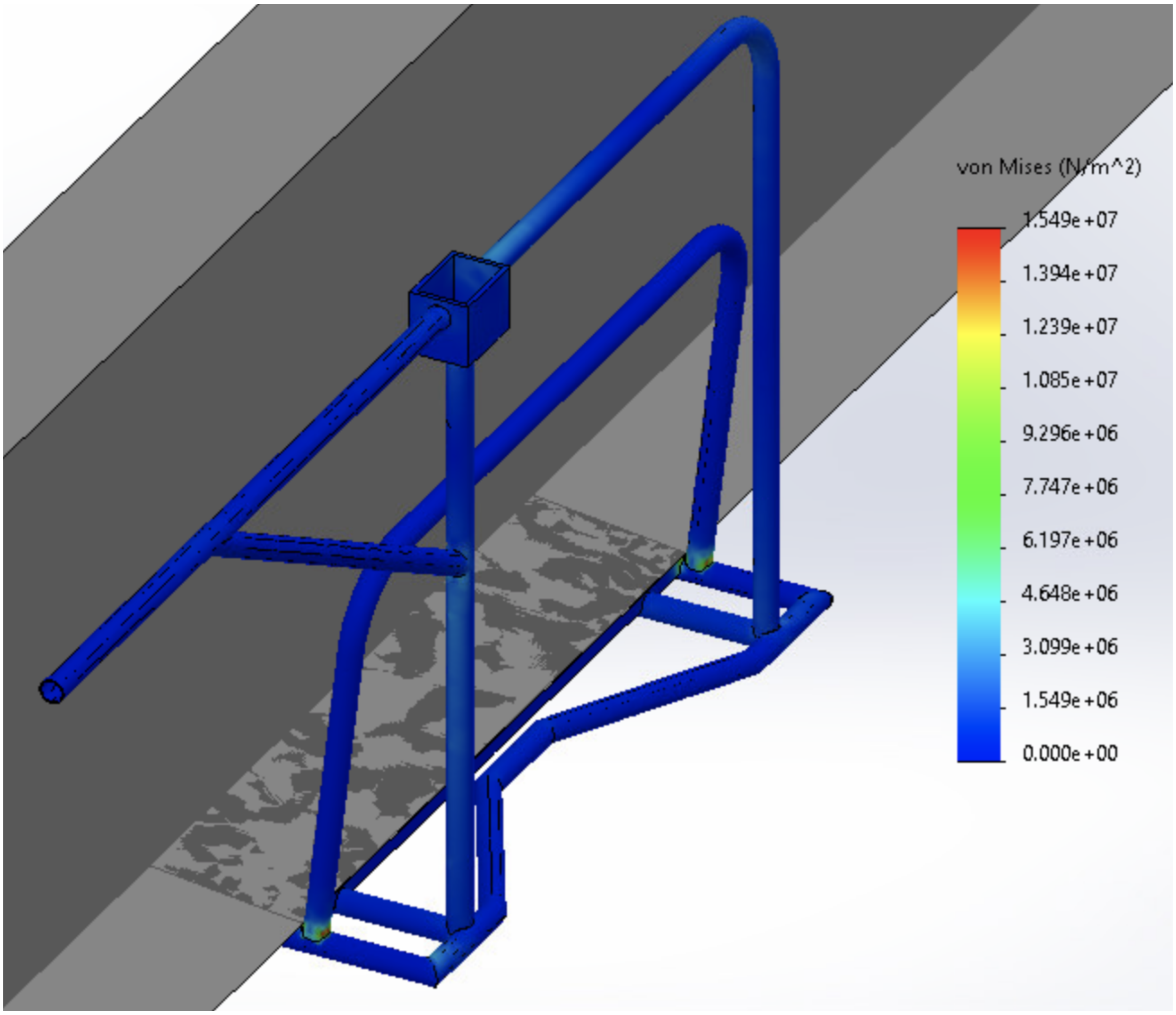


Figure 5: FEA analysis of the frame under the maximum load of 3000 lbs of drywall

The yield strength of cold drawn AISI 1010 steel is 305 MPa [7], which gives us a safety factor of 19.7 considering the maximum Von Mises stress experienced by the frame according to the FEA analysis. Additionally, the maximum stress is located at the bottom joints of the frame, which were not touched by our modifications. This gives us confidence that our changes to the Adapa cart design will not reduce its ability to carry the required maximum drywall payload.

Our second FEA analysis aimed to determine whether the steering column was strong enough to withstand the kinds of large and unusual loads that the frame might experience during transportation or use. Since only the steering column needed to be tested, only the right hand section of the frame was used, as shown in Figure 6.

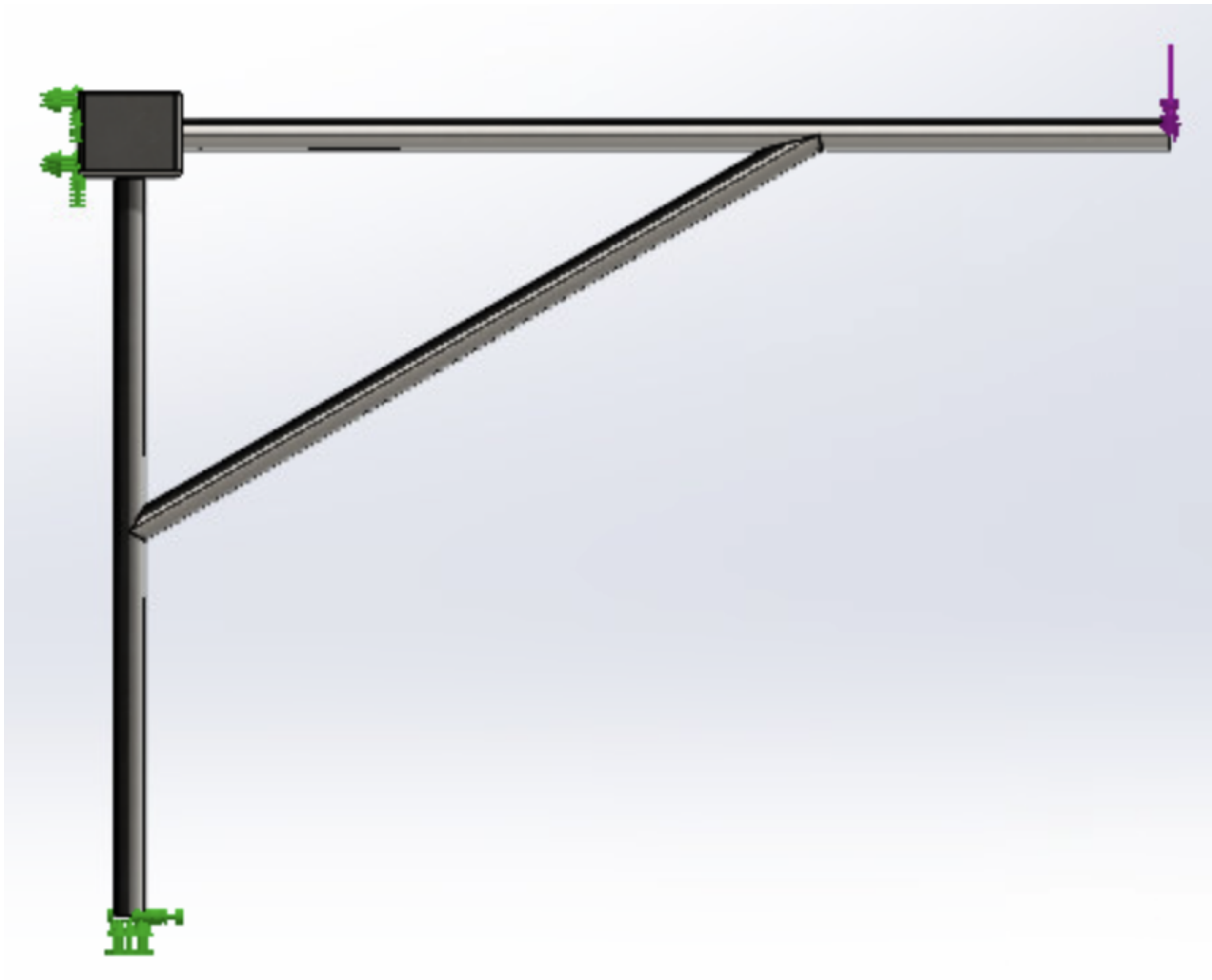


Figure 6: Portion of the frame used in the second FEA analysis

As shown in the above figure, fixed constraints were applied to the bottom of the vertical shaft and the left side of the bevel gear box. A vertical load of 100 lbf was applied to the end of the steering column to simulate someone pressing down on the steering wheel with a large amount of force. For reference, this is equivalent to the weight of half an average adult man. Again, the material chosen for the frame was plain carbon steel, though in our prototype it will be made of AISI 1010 steel. The results of this study are shown in Figure 7 below.

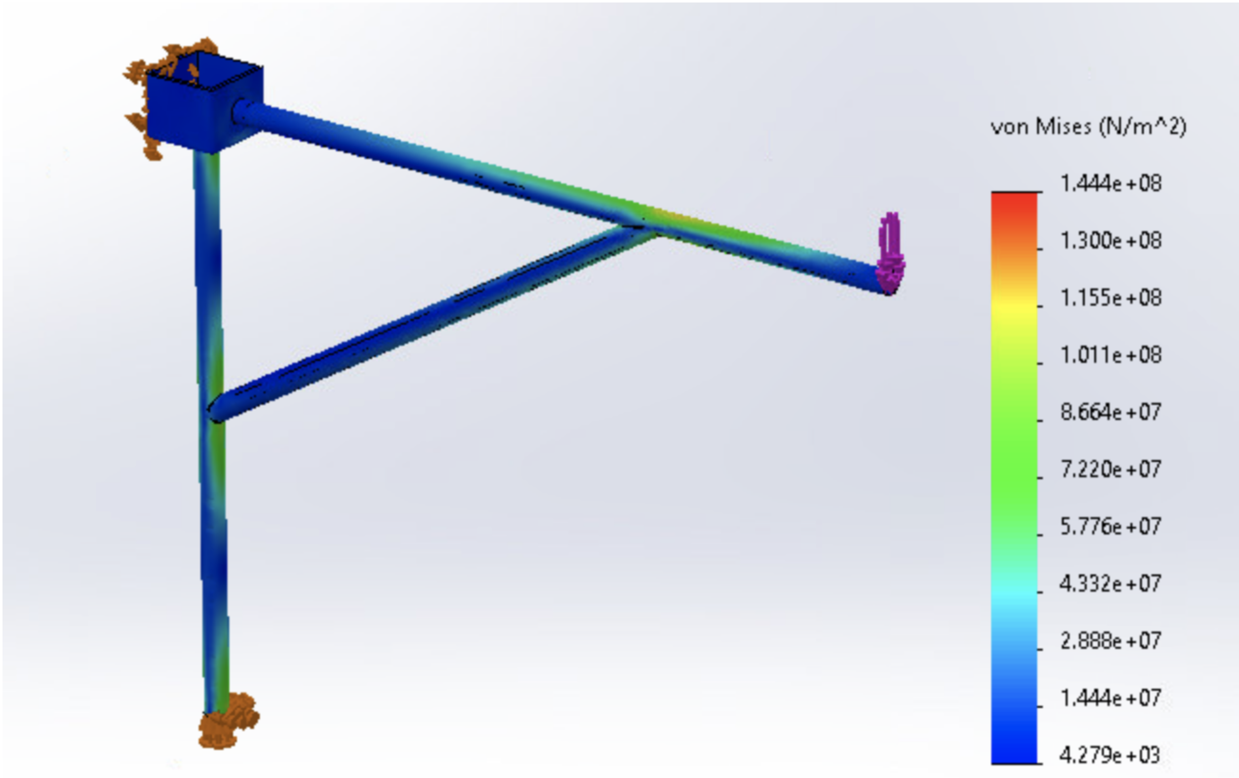


Figure 7: FEA analysis of the steering column under an unusually large load of 100 lbf at the steering wheel

The yield strength of cold drawn AISI 1010 steel is 305 MPa, giving us a safety factor of 2.11 in this case. However, this safety factor does not account for the fact that our frame should never actually undergo 100 lbf at the steering wheel. This means that the safety factor during normal use is much higher than this stated value. Thus through FEA analysis, we quantitatively determined that our frame is feasible for our final drywall cart design.

8.2 Shafts and Bevel Gears

The next components of our design to be systematically analyzed were the two shafts going into the bevel gears and the bevel gears themselves. Of these, the two shafts were the easiest to model through machine design principles, as their bending and torsion stresses could be calculated through Equations 1 and 2 found in a machine design textbook [8].

$$\sigma_{Torsion} = K_{fs}\frac{16T}{\pi d^3}, \sigma_{Bending} = K_f\frac{32M}{\pi d^3} \qquad \text{(Eq. 1 \& 2)}$$

The material selected for our shafts was AISI 1045 carbon steel, a reasonably priced medium carbon steel with a tensile strength of 310 MPa [9]. We assumed that there are no notches in the shafts, so K_{fs} and K_f were set to 1. The shaft diameter was 0.5” as per the McMaster Carr part that we ordered. For modeling purposes, we treated the shafts as beams that are pinned 2 inches from either end. In our CAD design, these supports come from ball bearings around the shafts and inside the frame tubing.

We assumed that our input torque from the steering wheel would be 10 Nm, generated by a force of 175 N applied to our steering wheel with a diameter of 4.5 inches. Since our top shaft was 44 inches long, this means that for the top shaft, T = 10 Nm and M = 8.89 Nm, so σ_t = 24.9 MPa and σ_b = 44.2 MPa. These values yield a minimum safety factor of around 7 when compared to the yield strength of the material, giving us confidence that the top shaft will not fail under its expected load.

As the bevel gears have a ratio of 1:2, the torque experienced by the second shaft should be 20 Nm. Since little to no lateral force should be applied since both ends are attached to relatively fixed components (the bevel gears and the bottom gear), the bending stress should not need to be calculated. Using the torque value of 20 Nm yielded σ_t = 49.7 MPa, which is again below the yield strength of the material by a substantial margin. Accordingly, the vertical shaft should also not fail under its expected load.

The calculations for the bevel gears were substantially less straightforward. From a machine design textbook [8], the maximum allowable contact stresses and bending stresses could be found from Equations 3 and 4.

$$\sigma_{Contact} = \frac{\sigma_{c,lim} Z_{NT} Z_W}{K_\theta Z_Z}, \quad \sigma_{Bending} = \frac{\sigma_{b,lim} Y_{NT}}{K_\theta Y_Z} \qquad \text{(Eq. 3 \& 4)}$$

In these equations, the Z, K, and Y variables are all multipliers that can be estimated using AGMA equations found in the machine design textbook [8]. Using those equations and the material constants of our bevel gear material, which was hardened AISI 1045 carbon steel as per our selected McMaster Carr part, resulted in the multiplier values listed below:

$\sigma_{c,lim}$ = 1200 MPa

Z_{NT} = stress-cycle factor for pitting resistance = 1.52

Z_W = hardness ratio factor = 1

K_θ = temperature factor = 1

Z_Z = reliability factor = 1.22

$\sigma_{b,lim}$ = 150 MPa

Y_{NT} = stress-cycle factor for bending strength = 1.20

$Y_Z = Z_Z^2 = 1.50$

By inputting these multipliers, maximum contact and bending stresses of 1495 MPa and 120 MPa were found, which is well below the estimated 49.7 MPa and 44.2 MPa shear and bending stresses applied to the shafts. Thus our bevel gears, just like our shafts, should be able to easily withstand the applied stresses that our cart will undergo during normal use.

8.3 Gearing System and Wheels

Seen in Figure 8, the gearing system of the cart is one of the most important aspects to the steering design as it will act to increase the torque from the steering wheel to the wheel itself. The required torque to turn to the wheel was calculated to be approximately 211 N-m**,** and with an input torque of 10 N-m, our gearing system must 'torque up" the system by a ratio of at least 1:21.1. For unmodelled effects, our gearing system will provide a 1:36 ratio. In order to obtain a ratio of 1:36, our system will go through a series of 2 sets of gears and a timing belt pulley. The first bevel gear provides a ratio of 1:2 , the second set of gears located underneath the cart provides a 1:6 ratio , and the belt system provides a ratio of 1:3.

With a selected material and known input torque, a gear analysis calculation was performed to determine the right parameters to meet our goal of having a safety factor of at least 1.2 for both contact and bending stresses. Shown in {reference calculation}, the pinion and gear chosen will have 17 and 102 teeth respectively with a face width of 1 in. From these findings, it is evident that the threat of failure for both the pinion and gear is bending and it is expected that the pinion would fail before the gear.. As one can see in the CAD model, there is an idler gear in between the pinion and gear. Its only act is to change the direction of the gear so that the steering wheel and the wheel will move in the same direction.

For the bending and contact stress calculation for gear, we used the Lewis equation provided by the textbook. Also provided as S_F and S_H, are the safety factor formulas for both bending and contact stresses:

$$\sigma_{bending} = \frac{(W^t\ K_o K_v K_s P_d K_m K_b)}{FJ},\ \sigma_{contact} = C_p\left(\frac{W^t\ K_o K_v K_s P_d K_m C_f}{d_p FI}\right)^{1/2}$$

$$S_{F,bending} = \frac{S_t Y_N}{\frac{K_T K_R}{\sigma,bending}},S_{H,contact} = \frac{S_C Z_N C_H}{\frac{K_T K_R}{\sigma,contact}}$$

With material selection using carburized and hardened steel grade 1, the factors shown in Table 3 below were used to calculate the respective bending and contact stresses.

Table 3: Lewis Factors for Gear Train Calculations

Factors	Pinion	Gear
[W^t] - Transmitted Load	208.2353 (lbs)	208.2353 (lbs)
[K_o]- Overload Factor	1	1
[K_v]- Dynamic Factor	1.0041	1.0036
[K_S]-Size factor	1.1222	1.1340
[P_d]-Pitch Diameter	1.7 in	10.2 (in)
[K_m]-Load Distribution Factor	2.2834	2.2834
[K_b]- Rim thickness factor	1	1
[F] - Face Width	1 (in)	1 (in)
[J]-Geometry Factor for bending strength	0.20	0.20
[C_p]- Elastic Coefficient	2300 (psi)	2300 (psi)

[C_f]- Surface Condition factor	1	1
[C_H]- Hardness Ratio Factor	1	1
[I]-Geometry factor of pitting resistance	0.1071	0.1205
[K_T]- Temperature Factor	1	1
[K_R]- Reliability Factor	0.850	0.850
[S_T]- Bending Strength at 10^7 cycles and 0.99 reliability. For carburized and hardened steel grade 1	28,260 (psi)	28,260 (psi)
[S_C]- Contact strength at 10^7 cycles and 0.99 reliability. For carburized and hardened steel grade 1	93,500 (psi)	93,500 (psi)
[Y_n]-Stress Cycle factor for bending strength	1	1
[Z_n]- Stress cycle factor for pitting resistance	1	1

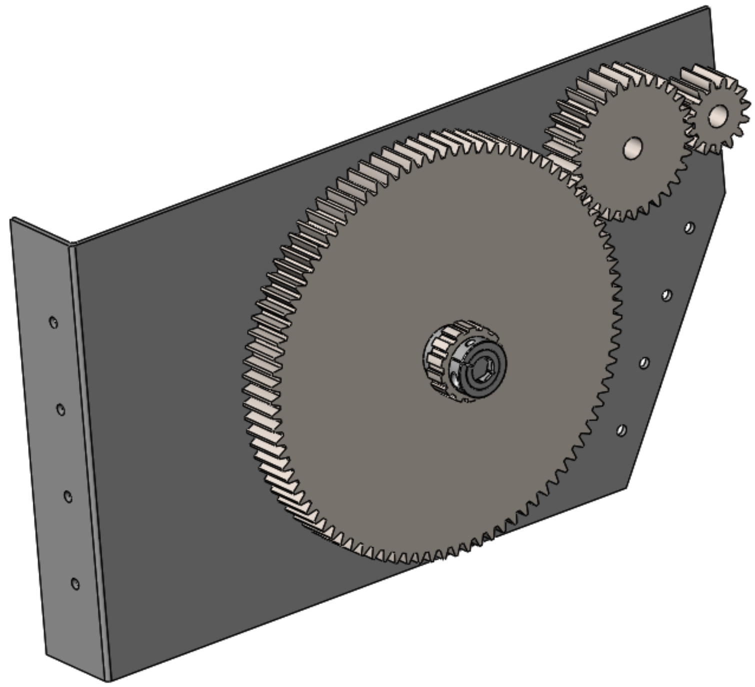


Figure 8: Gear System Assembly

8.4 Calculations for the timing belt

Following the gear system that has a ratio of 1:6, a shaft connected to the driven gear will drive a pulley that is directly below it. This pulley system is shown in Figure 9. It has a timing belt with trapezoidal teeth shape, made of urethane with kevlar reinforcement. Its goal is to increase the torque from the gear system and translate it to the shafts that turn the front two wheels. The given textbook is not very clear in providing failure analysis of timing belts; however simple tensile calculations were done to verify if the belt can withstand the tensile force due to the input and output torques. From sources [10], the max allowable belt tension for the material chosen is 1229 $\frac{N}{25\,mm}$. The belt width chosen is 1 in [25.4 mm], therefore the max allowable belt tension is 1147.1 N. From basic tension calculations, the tensile force on the tight side of the belt is approximately 146.25 N-m, which is well below the max allowable belt tension.

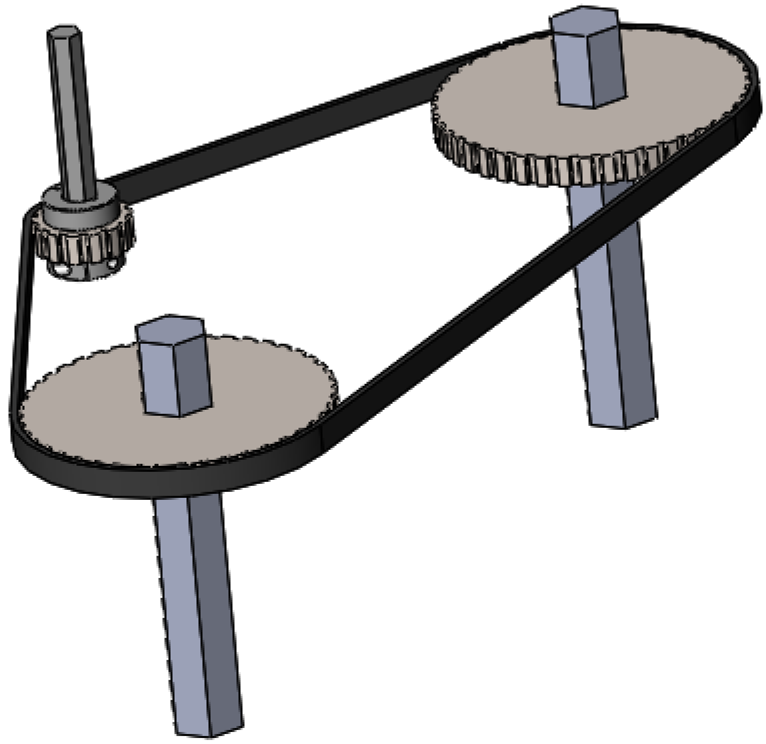


Figure 9: Pulley System

8.5 Kickstand Overview

The final kickstand design is showcased in operation in Figure 10 such that the swinging arm makes a 45° angle with the center mounting rod for the emergency kickstand mechanism as the swinging arm supports the load of the cart and its payload. Figure 11 and Table 4 below present an overview of the loads that act upon/between the main components of the emergency kickstand (i.e. swinging arm, center rod, top plate, etc.), while items of Appendix B highlight calculations/justifications/charts that indicate how the cart is resistant to various modes of failure. Major forces acting on the center rod include F_{load} = 2,000 lb., which is the portion of the drywall payload weight that acts perpendicular to the rod's axis, R_1 = 1,429.8 lb., which is the reaction force imparted by the bottom bracket upon the center rod, F_1 = 1,414.2 lb., which is the vertical force supplied by the swinging arm upon the center rod at their pin connection, and F_2 =

15.6 lb., which is the force imparted upon the top plate by the swinging arm at their contact point. The small relative magnitude of F_2 is notable; the top plate is designed to prevent the swinging rod from extending beyond the 45° angle, so under normal operating conditions the reaction from the top plate is minimal. For the swinging arm, two main loads exist: an equal in magnitude but opposite in direction force F_2 = 15.6 lb. acts upon the swinging arm from the top plate by pushing on it horizontally towards the cart body, and a normal force $F_N = F_1$ = 1,414.2 lb. pushes vertically upon the base of the swinging arm from the ground. Using these main loads, the various failure modes evaluated in Appendix B and discussed below were tested.

Figure 10: Model Rendering of deployed kickstand

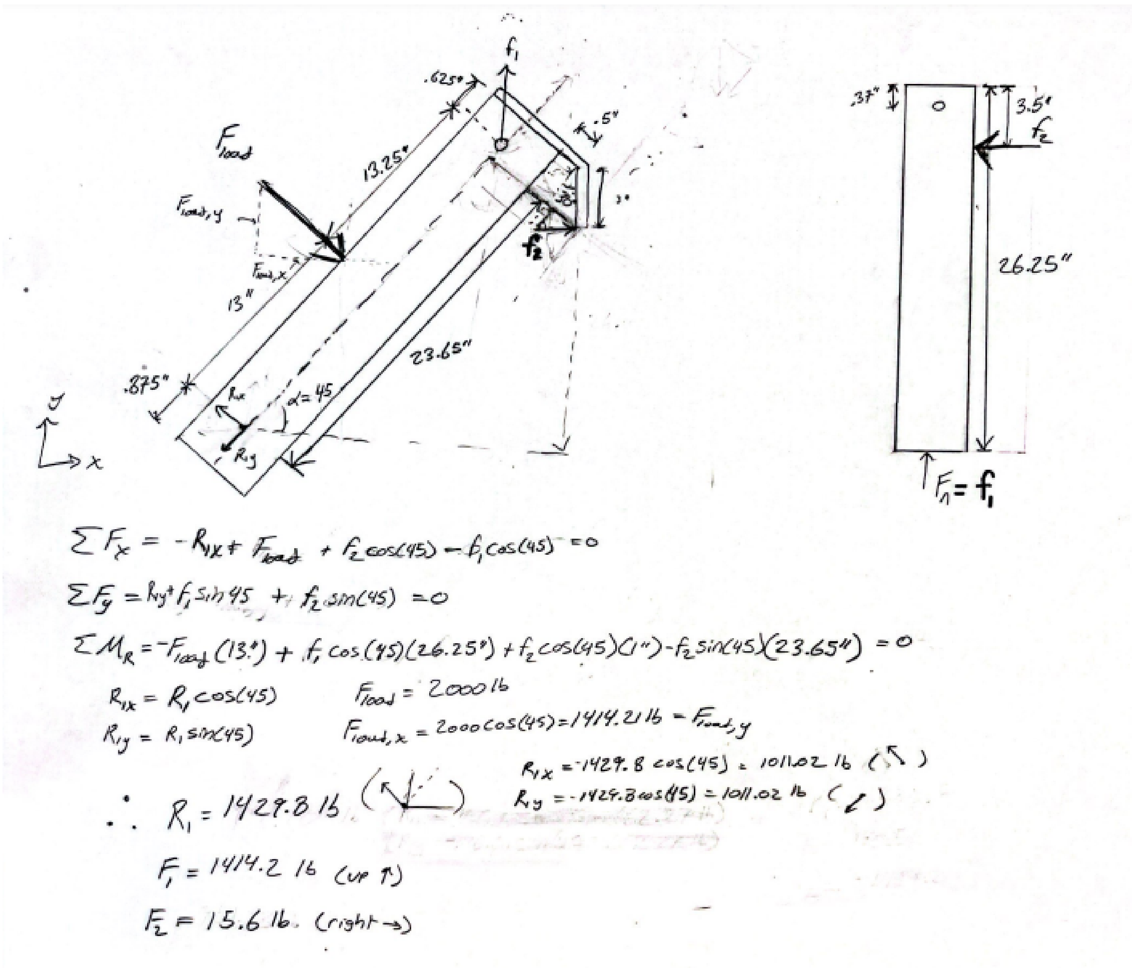


Figure 11: Free Body Diagram of Emergency Kickstand Assembly Depicting Main Forces

Table 4: List of Forces and Corresponding Magnitudes Apparent in Emergency Kickstand

F_{load}	*2000 lb*	*perpendicular to center support rod*
F_1	*1414.2 lb*	*reaction force from swinging rod*
F_2	*15.6 lb*	*reaction force from top plate on swinging rod*
R_{1x}	*1011.02 lb*	*x-component of connection hardware reaction force*
R_{1y}	*1011.02 lb*	*x-component of connection hardware reaction force*

8.6 Connecting Pin Shear

The first failure mode tested for the emergency kickstand was shear failure on the pin that connects the swinging arm and the center rod of the mechanism. As shown in Calculation B1, F_1 is split into two equal components that act vertically upon the pin from the swinging arm, while the center portion of the pin is pushed down on in the opposite direction by a force equivalent to F_1. Analyzing a half of the pin shows up to what shear force the pin may experience in operation, helping with the selection of a pin of appropriate dimensions and composition. From the analysis presented in Figure B1, the pin should be able to withstand a shear force of up to 707.1 lb. without failing, and so a 0.25" diameter dowel pin of length 4" made of 4140 Alloy Steel was selected for the design due to its rated shear breaking strength of 10,000 lb.

8.7 Buckling of Center Rod and Swinging Arm

The first step to determining buckling loads for the center rod and swinging arm is to determine whether the component being tested for failure behaves as a short or long column. To do this, the components' slenderness ratios were calculated as $\frac{L}{K}$ and compared to a critical slenderness ratio, $(\frac{L}{K})_1 = \sqrt{\frac{2C\pi^2 E}{S_y}}$, where C = 1, E is Young's Modulus, and S_y is yield strength. L is simply the component length, and K is the square root of the ratio of I, the 2nd area moment of inertia of the cross-sectional surface being applied an axial force, and A, the actual area of the cross-sectional surface being applied an axial force. After performing these calculations, detailed in Calculation B2, the center rod was found to behave like a short column, since its slenderness ratio is 94.1278 in. which is less than its $(\frac{L}{K})_1$ value of 110.627, whereas the swinging arm behaves as a long column, since its slenderness ratio is 125.22 in. which is greater than the $(\frac{L}{K})_1$ value of 110.627. And so, the critical load for the center rod to buckle is calculated as $\frac{C\pi^2 EI}{L^2}$ = 14,875 lb., which is significantly greater than its largest axial force of 1,011.02 lb., hence the rod won't buckle. Similarly, the critical load for the swinging arm to buckle is calculated as $\frac{C\pi^2 EI}{L^2}$ =

5,818.8 lb., which is significantly greater than its largest axial force of 707.1 lb., hence the swinging arm won't buckle.

8.8 Stress Performance of Rods and Mounting Bracket

Next, a stress analysis was performed on the center rod to determine whether the component could withstand the various bending and axial stresses imparted upon its features. Calculation B3 provides an overview of the various shear forces and bending moments acting upon the center rod at various points along its main axis. The point with the greatest bending moment occurs at the point where F_{load} applies upon the center rod, yet the stress analysis was conducted on the hole for the top bolt of the center rod's connection to the cart base. The reason this point was chosen for the stress analysis is because it has relatively large shear force and bending moment values as indicated by Calculation B3, it is the location of a stress concentration since there is a discontinuity in the center rod due to the hole, and it is also close to another feature and stress concentration — the other bolt hole — hence this point is an area of interest for the stress analysis as suggested by St. Venant's Principle. At this point, the axial stress was calculated using $\sigma_{axial} = K_t \cdot \sigma_0 = K_t(\frac{4F}{\pi(d_2^2 - d_1^2)}) = (3.533)(1056.2\, psi) = 3{,}731.63$ psi, where d_2 is the outer diameter of the rod, d_1 is the inner diameter of the rod, and F is the axial reactionary force exerted at this one hole. The bending stress was calculated using a similar approach to be $\sigma_{bending} = K_t \cdot \sigma_0 = K_t(\frac{32Md_2}{\pi(d_2^2 - d_1^2)}) = (3.41)(10,560.7\, psi) =$ 36,011.987 psi, where M is the bending moment at this hole. For the above calculations, stress concentration values, K_t, were calculated by interpolating on Figures 12 and 13 which showcase stress concentration value curves. Then, the shear due to bending was calculated as $\tau_{xy} = \frac{2V}{A} = \frac{2(505.1)}{0.4786\, in.^2} = 2{,}112.45$ psi. Using these calculated values, a von Mises stress value was calculated to be $\sigma' = \frac{1}{\sqrt{2}}(\sqrt{\sigma_{bending}{}^2 + 6(\tau_{xy})^2}) = 36{,}104$ psi. Note, only the bending stress and shear bending stress apply to this von Mises stress calculation, for the axial stress aligns with the center rod's main axis and are not acting at the same location. Finally, von Mises Failure Theory was applied to ensure that the structure can withstand the stresses applied upon it, which the feature passes since its von Mises stress of 36,104 psi is less than half the yield strength, S_y, of the material the center rod is made of; the S_y value is 75,000 psi, so $\frac{S_y}{2} =$ 37,500 psi. In-depth calculations for the entire stress analysis can be found in Calculation B4. Finally, since the center rod houses more discontinuities and stress concentrations than the swinging arm, and since the components are made from the same material, no stress analysis was necessary for the swinging arm, for the less stable center rod already proved viable in this aspect.

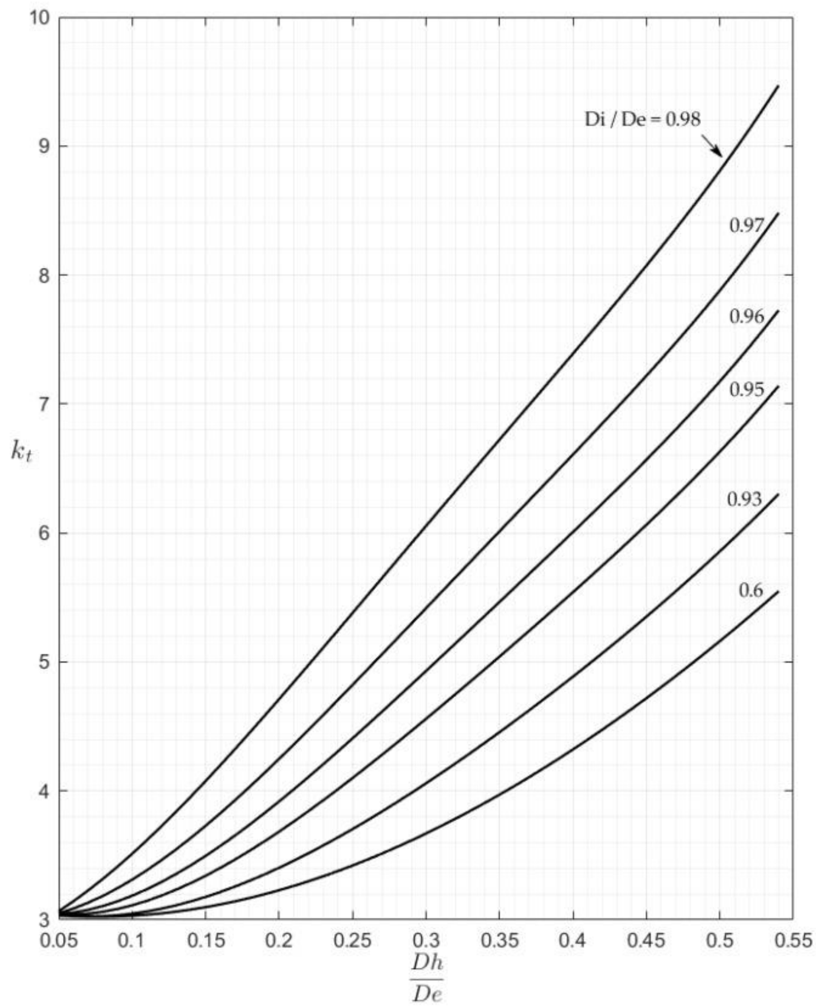


Figure 12: K_t (Stress Concentration) for Bending

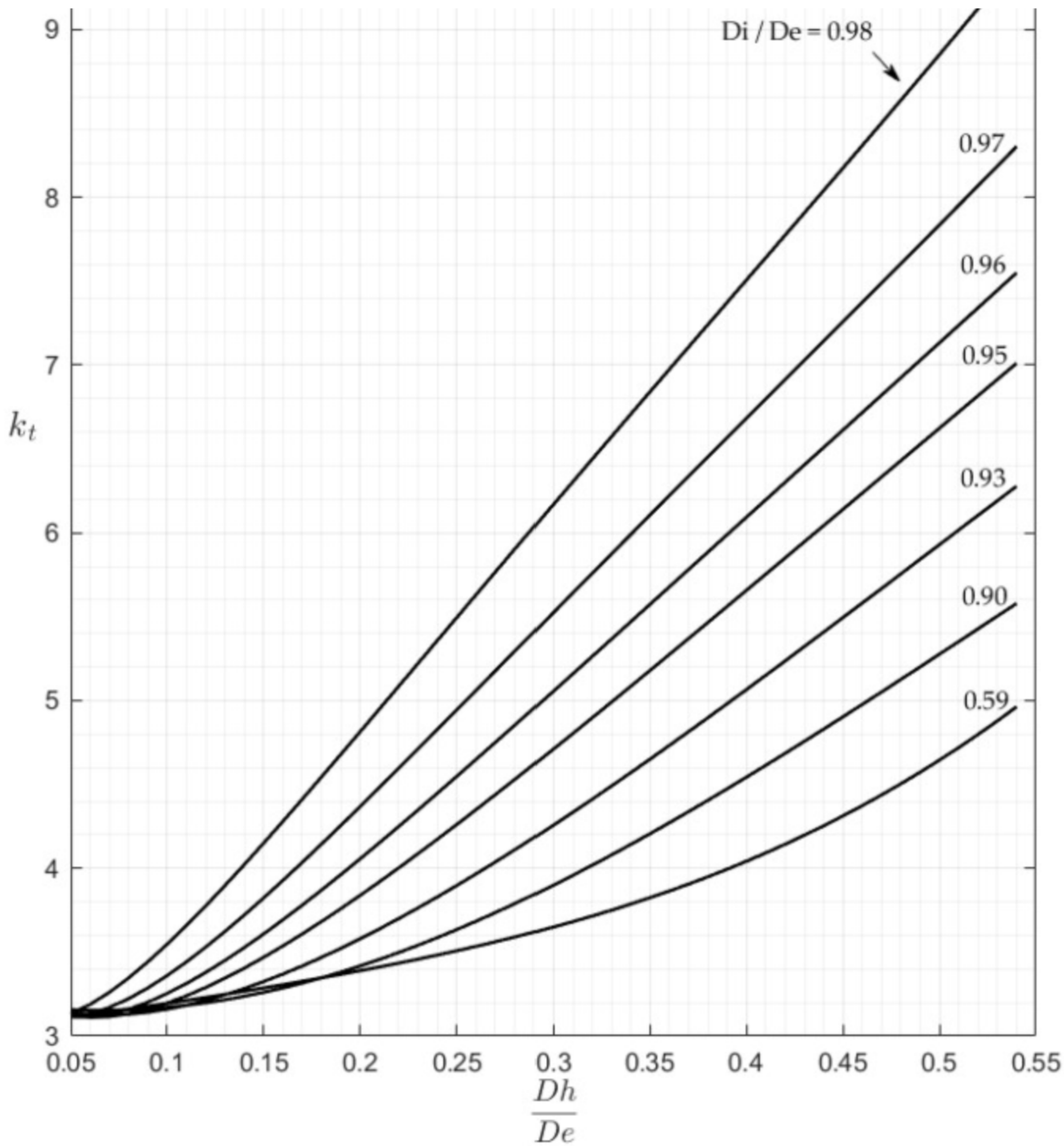


Figure 13: K_t (Stress Concentration) for Axial

Once the stresses experienced by the center support rod were defined and validated, a finite element analysis was able to be conducted on the mounting bracket component. With the reaction forces from the center support rod on the connection hardware fully defined and validated, the design for the mounting bracket could be evaluated rapidly using FEA. Figure 14 depicts the results of this study.

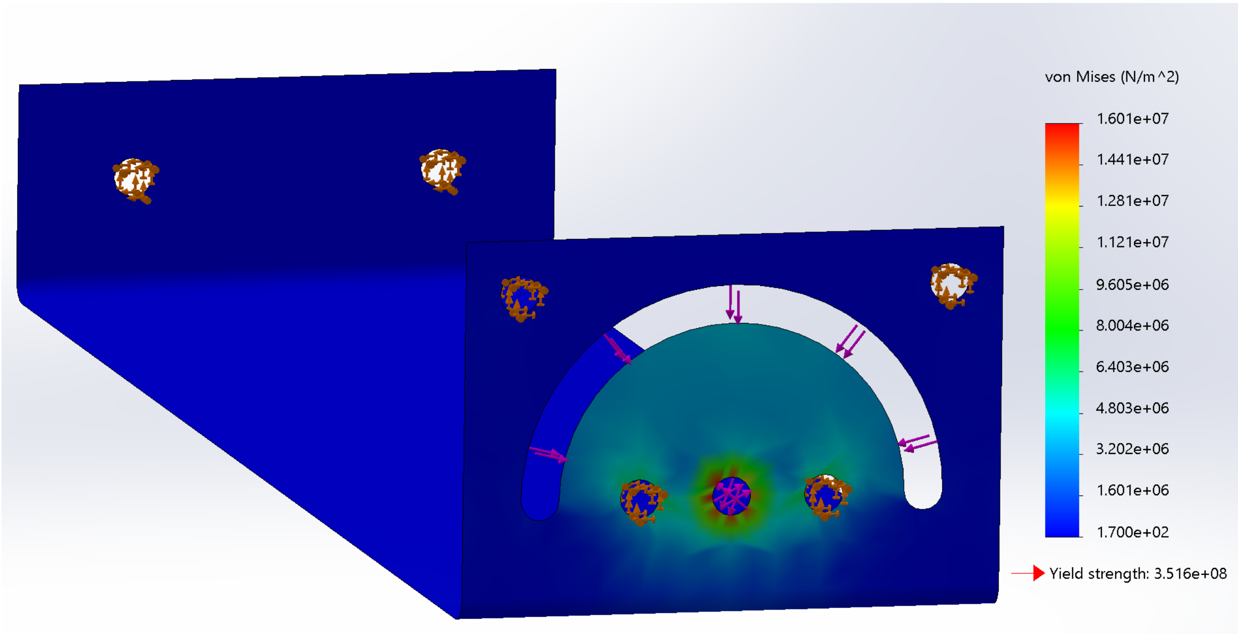


Figure 14: Results of FEA simulation on the Mounting Bracket for the Emergency Kickstand

To aid the upcoming prototyping efforts, the mounting bracket was modeled using sheet metal where only two bends and a few hole punctures are required to achieve the final geometry. The mounting bracket material was chosen to be $\frac{1}{4}in$ sheet metal made from 1020 Low Carbon Steel. The area of highest stress on the bracket occurs at the lower connection point between the bracket and the center support rod, reaching maximum stress values of $6.4 \times 10^{6} \frac{N}{m^{2}}$ (928.24 psi) due to the feature stress concentration. These stresses are orders of magnitude lower than the yield strength of the sheet metal, validating the geometry for the bracket and connection points. The largest takeaway from this FEA study was the identification of stress magnitudes for the purpose of selecting connection hardware ratings. The standardized washers, bolts, bushings, and lock nuts used to mount the center support rod to the mounting bracket were chosen to have a tensile strength of 150,000 psi, a rating that far exceeds even the worst case failure modes anticipated by the emergency kickstand failsafe.

9. Societal, Environmental and Sustainability Considerations, and Industrial Design

In terms of societal, environmental, and sustainability considerations, the team's current design has only briefly been vetted through the use of EduPack's Eco Audit Tool; the team's primary goal was to select materials so as to maximize cart stability and safety rather than using the most environmentally-friendly parts. After further review of the Eco Audit, the team will share insights on the sustainability of our final design in the final report and presentation. As for industrial design, the current iteration of the cart has taken into account several considerations. First, the addition of the steering system provides an ergonomic solution for worksite employees to turn the heavy cart — the advantageous gear ratios allow for the workers to turn the cart while supplying less torque than they currently do with their manual push carts while also having a dedicated and easy-to-grip interface to turn the cart. Also, since our design actually comprises relatively easy-to-install modular additions to the current Adapa cart, the cart is able to maintain both its general shape and color familiar to many leading drywall carts on the market, which should appeal to the target audience of this upgraded cart design: construction materials distributors and construction worksite employees.

10. Team Member Contributions

Ryan Grajewski (Team Leader):

- Participated in the redesign of the emergency kickstand for the cart
- Created CAD components of emergency kickstand and assembled all the components into subassembly of the kickstand
- Led communication and meetings with sponsors at GMS
- Ordering and maintaining team finances for reimbursement of parts
- Performed majority of engineering analysis for the emergency kickstand components
- Writer of portions of the engineering analysis for the emergency kickstand; writer of the conclusion

Graham Brantley (Writing Lead/Materials Lead):

- Editor of executive summary, introduction, market research, prior art, codes and standards, customer requirements, design concept ideation, and selected design concept sections (cut down significantly from Report #1)
- CAD of the shafts, and bevel gears
- Materials selection for the frame, shafts, and bevel gears
- Engineering feasibility calculations for the frame, shafts, and bevel gears
- Writer of engineering analysis of the frame, shafts, and bevel gears

Will Hagler (Prototyping Lead):

- CAD for gearing system and timing belt system
- Engineering feasibility calculations for the gear, and timing belt system
- Writer of engineering analysis for the gear and timing belt system
- Been getting certified for specific machine in the MMM

Garrett Rodino (CAD Lead):

- Combined Subassemblies in a full CAD assembly
- Created exploded and parts drawings for each individual CAD part

- Compiled drawings and views into a comprehensive fabrication package with subassemblies and grouping the parts associated with those subassemblies together

Nischal Bandi (Analysis Lead):

- Participated in the redesign of the emergency kickstand for the cart
- Responsible for CAD and geometric calculations related to the emergency kickstand's center rod, swinging arm, top plate, and pin
- Aided in failure calculations for components of the emergency kickstand
- Wrote majority of engineering analysis for emergency kickstand components

11. Conclusion and Future Project Deliverables

There is a well-established need for a redesign of the standard drywall cart; injuries occur primarily due to the cart tipping when a weight imbalance is applied or due to the cart's caster wheels straying from the direction of motion of the cart. The selected design concept works to address both of these failure modes through implementing a steering system that enhances the maneuverability and control of the cart's motion and an emergency kickstand that acts as a failsafe mechanism to minimize injury in the occurrence of the cart tipping. Through market research, consultations with the GMS sponsors, and financial considerations, the opted for approach to designing and prototyping the new cart was to modify, rather than completely recreate, the current Adapa drywall cart frame. This approach directly benefits GMS, for the chosen design will not require manufacturing of a whole new fleet of carts. Rather, both the steering and emergency kickstand mechanisms have been designed with the ultimate goal of being cost-effective additions GMS can implement to their current carts that significantly improve the maneuverability and stability/safety of carts they already own in large numbers

Since the concept/ideation phase, significant efforts were put into validating the mechanical design of each sub-component. First, the FEA of the modified cart frame was used to establish that the frame modifications would still be supported by the current load ratings, even when subject to larger stresses at connection points to the alterations. The design of the steering system required an initial designation of input and output torque requirements to turn the wheels. The general layout of the gear train was then developed to maximize torque increases while minimizing weight and volume. Machine design calculations were crucial in the design of the bevel gears and steering gear system; FEA and stress performance calculations for the gear systems verify that the components can interface with each other and convert torque delivered by significant factors to allow for easy steering of the cart without posing a serious risk of failure. Finally, FEA on the bracket that connects the center rod of the emergency kickstand to the cart's base, buckling load failure calculations on the center rod and swinging arm of the emergency kickstand, a shear force analysis on the pin that connects the center rod and swinging arm, and a von Mises stress failure analysis confirm that the emergency kickstand can resist many of the most impactful modes of failure that it might face in operation. With the updated cart design now validated from an engineering perspective, the team is ready to share the design with its GMS sponsors one more time at an in-person meeting on March 16, 2022 to receive and incorporate last-minute feedback. Afterwards, pending part orders will be placed so that the team can begin prototyping promptly so as to have a completed product by the Capstone Design Expo. Throughout the upcoming weeks of machining/prototyping, every member of the team will participate in some form of machining training and prototyping to ensure quick production of the necessary parts needed to complete our final design before the quickly approaching Capstone Design Expo on April 26, 2022. Finally, our team will be meeting regularly — likely weekly or at least bi-weekly — with our sponsors at GMS for the remainder of this project to update them on prototyping status of our final design and facilitate some field testing to evaluate our final product. The Gantt Chart below in Figure 15 provides a broad overview of the team's deadlines/goals for the remainder of the project.

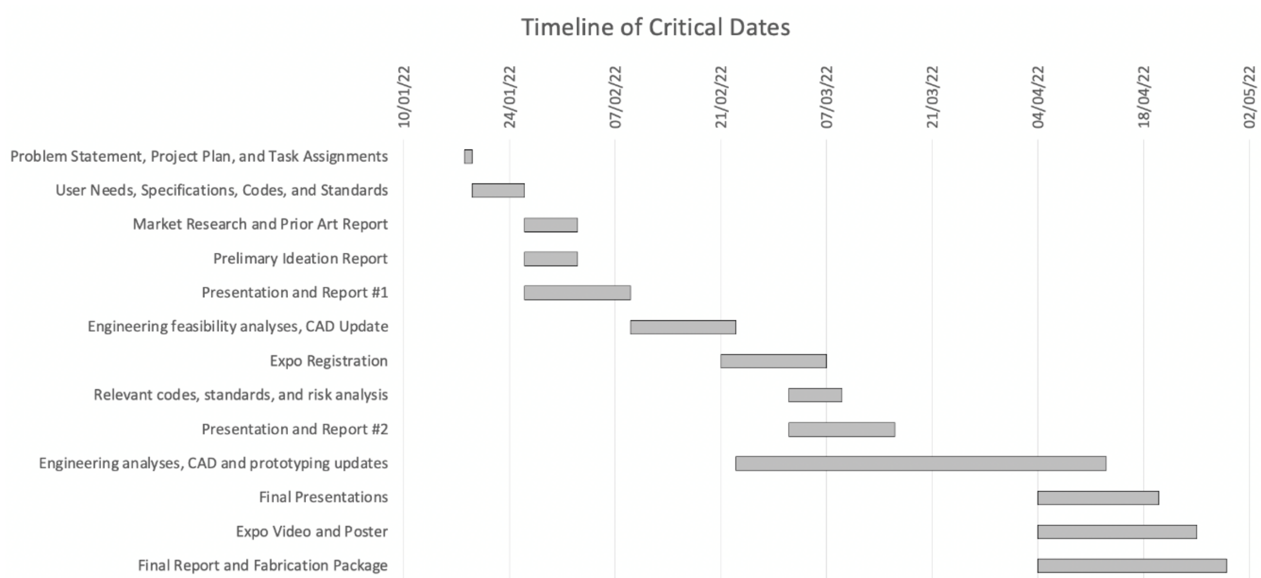


Figure 15: Gantt Chart of General Deadlines

Appendix A

Table A1: Specifications sheet

`				Issued:	1/31/2022	
			For:	Page:	1	
			Specification	Drywall Cart		
No.	**Date**	**D/W**	**Requirements**	**Responsible**	**Source**	**How Validated**
General						
Cost	1/31/2022	W	Total Manufacturing Cost between 350-450$	Everyone	Sponsor	Cost Analysis
Schedules	1/31/2022	D	Finished by Expo	Everyone	GT	Is it done?
Physical Characteristics						
Size	1/31/2022	D	Fits within Doorframe and small hallways (30-32 in.) Length < 16'	Everyone	Standard	Testing
Maneuverability	1/31/2022	D	0 turn radius	Everyone	Sponsor	Testing
Material	1/31/2022	D	Lightweight, strong, cheap material	Everyone	Sponsor	Material Optimization
Change of Direction	1/31/2022	D	Smooth turning, no wheel hangups	Everyone	Sponsor	Testing
Mechanical						
Stiffness	1/31/2022	D	No flex during use	Everyone	Standard	Testing
Weight	1/31/2022	D	Max weight of 90 lbs.	Everyone	Sponsor	Scale
Strength	1/31/2022	D	Supports a minimum of 2500-3000 lbs	Everyone	Sponsor	Modeling/Hand Calcs
Performance						
Manufacturable	1/31/2022	W	Must be manufacturable using simple machining equipment	Everyone	Sponsor	Prototyping
Repairable	1/31/2022	D	Must be modular in design for ease of repair	Everyone	Sponsor	DFMA Analysis
Durable	1/31/2022	D	Infinite Life Assumption (10^6)	Everyone	Sponsor	Modeling/Hand Calcs
Safe	1/31/2022	D	Design must be safe within operating ranges	Everyone	Sponsor	Testing/Modeling
Operation	1/31/2022	D	Must be within OSHA guidelines	Everyone	GT	Testing
Assembly	1/31/2022	D	Modules assembled using standard fasteners	Everyone	Sponsor	Prototyping
Ergonomics	1/31/2022	D	Must be easy to load and unload	Everyone	Sponsor	Sponsor feedback

Table A2: Concept evaluation matrix. Concept #1 (2 wheel drive + angled A-frame + external kickstand) was selected as the best design concept moving forward.

		Drywall Cart Design Concepts									
		Concept #1		Concept #2		Concept #3		Concept #4		Concept #5	
		2WD + Angled A-Frame + External Kickstand		4WD + Angled A-Frame + Internal Kickstand		Caster Wheels + Angled A-Frame + Suspension + Footbrake		Caster Wheels + Flatbed + Footbrake		Caster Wheels + Angled A-Frame + Internal Kickstand + Side Guards + Footbrake	
											
User Needs	Importance	Score (1-5)	Product	Score (1-5)	Product	Score (1-5)	Product	Score (1-5)	Product	Score (1-5)	Product
Low cost	5	2	10	1	5	2	10	5	25	3	15
Safe to use	10	4	40	5	50	3	30	1	10	3	30
Can use on multiple surfaces	9	4	36	3	27	4	36	1	9	1	9
Durable	7	4	28	1	7	3	21	5	35	2	14
Lightweight	6	3	18	3	18	3	18	5	30	2	12
Not complex	4	3	12	1	4	4	16	5	20	3	12
Highly maneuverable	10	5	50	4	40	5	50	5	50	5	50
Same capacity as current cart	7	3	21	3	21	3	21	5	35	3	21
Stable	10	5	50	5	50	3	30	1	10	3	30
Easy to use	3	2	6	1	3	5	15	5	15	4	12
Easy to repair	5	3	15	1	5	4	20	5	25	2	10
Looks nice	1	3	3	3	3	3	3	5	5	3	3
Portable by 1 person	5	3	15	3	15	3	15	3	15	3	15
Can fit in truck	10	5	50	5	50	5	50	4	40	5	50
Total Scores		354		298		335		324		283	
Rank		1		4		2		3		5	

Engineering Requirements	Total width (in)	Total length (in)	Total height (in)	Total weight (lbs)	Total cost ($)	Number of cycles until failure (unitless)	Load capacity (lbs)	Tipping moment (lb*ft)	Failure system response time (ms)	Impact yield stress (ksi)	Turning radius (ft)	Number of parts (unitless)	Minimum safety factor (unitless)	Longest time to replace part (min)	Maximum corrosion rate (mpy)	Minimum surface roughness (milli in)	Time to learn to operate (hours)	Number of controls (unitless)	Height of center of mass (ft)	Maximum speed (mph)	Packing density (units/ft^3)
Total width (in)																					
Total length (in)																					
Total height (in)																					
Total weight (lbs)	⊕	⊕	⊕																		
Total cost ($)				+																	
Number of cycles until failure (unitless)																					
Load capacity (lbs)				+																	
Tipping moment (lb*ft)	⊕		⊖	+			+														
Failure system response time (ms)								+													
Impact yield stress (ksi)						+															
Turning radius (ft)	-	⊖																			
Number of parts (unitless)				+	⊕	-				-											
Minimum safety factor (unitless)					⊕	⊕	+	+	-	⊕											
Longest time to replace part (min)				+								⊕									
Maximum corrosion rate (mpy)					⊖	-				-		+									
Minimum surface roughness (milli in)					⊕	+							+		⊖						
Time to learn to operate (hours)												+									
Number of controls (unitless)					+						-	⊕		+			⊕				
Height of center of mass (ft)			⊕					⊖													
Maximum speed (mph)				-													+		-		
Packing density (units/ft^3)	⊖	⊖	⊖																		
Direction of Improvement	∘	∘	∘	↓	↓	↑	↑	↓	↓	↑	∘	↓	↑	↓	↓	∘	↓	↓	↓	∘	↑
Importance (1-10)																					

Figure A1a: House of Quality (correlation matrix)

		Engineering Requirements																				
Customer Requirements	Importance (1-10)	Total width (in)	Total length (in)	Total height (in)	Total weight (lbs)	Total cost ($)	Number of cycles until failure (unitless)	Load capacity (lbs)	Tipping moment (lb*ft)	Failure system response time (ms)	Impact yield stress (ksi)	Turning radius (ft)	Number of parts (unitless)	Minimum safety factor (unitless)	Longest time to replace part (min)	Maximum corrosion rate (mpy)	Minimum surface roughness (milli in)	Time to learn to operate (hours)	Number of controls (unitless)	Height of center of mass (ft)	Maximum speed (mph)	Packing density (units/ft^3)
Low cost	5	○	○	○	○	◎							○				Δ		Δ			
Safe to use	10								◎	◎	Δ			◎		○	○			◎	○	
Can use on multiple surfaces	9																					
Durable	7						◎				◎					◎			○			
Lightweight	6	○	○	○	◎																	
Not complex	4												◎		○			○	◎			
Highly maneuverable	10	○	○	○	○				Δ			◎								Δ	Δ	
Same capacity as current cart	7							◎														
Stable	10	Δ	Δ	Δ	Δ				◎					Δ						◎	○	
Easy to use	3				○				○									◎	○			
Easy to repair	5												◎		◎							
Looks nice	1																					
Portable by 1 person	5	○	○	○	◎																	
Can fit in truck	10	◎	◎	◎																		◎
Targets		12	50	48	90	450	10^6	3000	300	10	72.5	0	20	2	3	1	1	1	2	2	3	0.1
Absolute Importance		178	178	178	163	45	63	63	199	90	73	90	96	100	57	93	35	39	71	190	70	90
Relative Importance (%)		8.24	8.24	8.24	7.54	2.08	2.92	2.92	9.21	4.16	3.38	4.16	4.44	4.63	2.64	4.30	1.62	1.80	3.29	8.79	3.24	4.16
Rank		4	4	4	6	19	16.5	16.5	1	11	13	11	8	7	18	9	21	20	14	2	15	11

Figure A1b: House of Quality (relationship matrix and engineering requirement rankings)

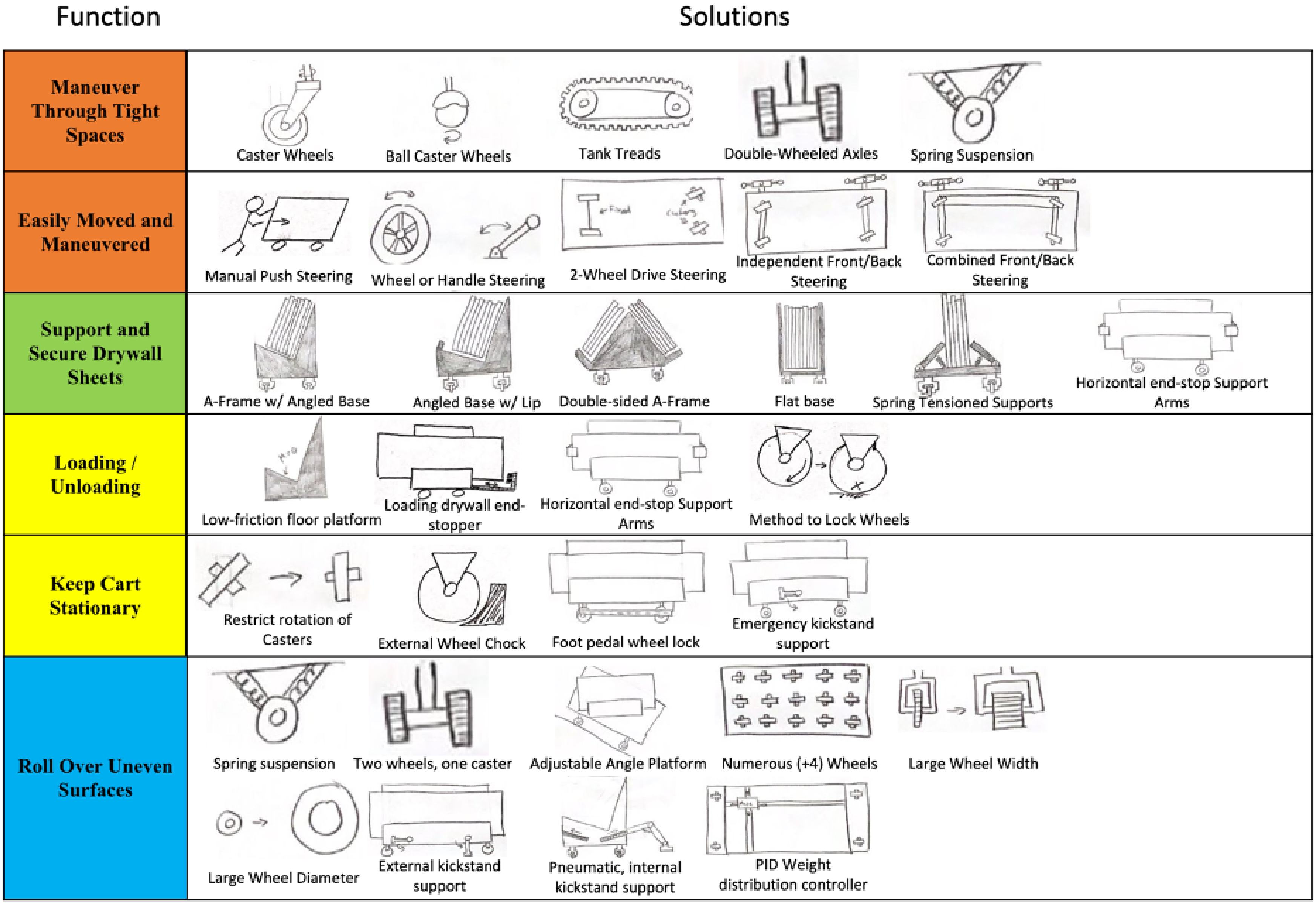
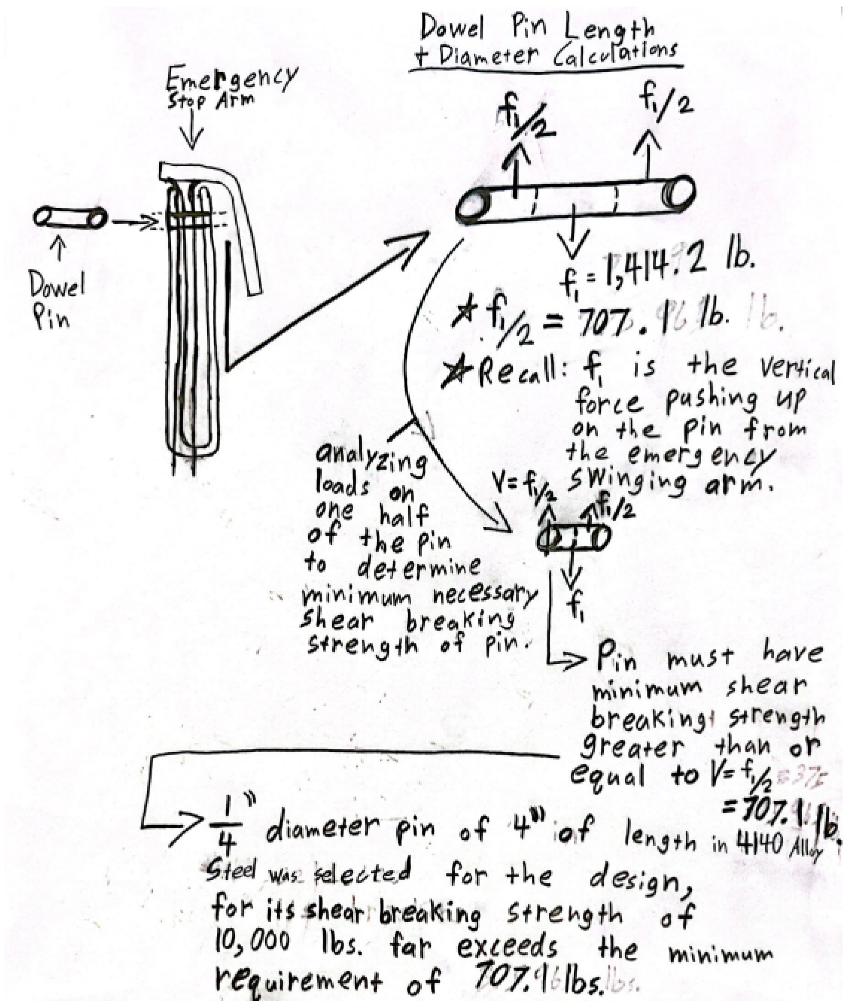


Figure A2: Morphological chart with sketches of each component concept design

Appendix B

Calculation B1: Connecting Pin Shear and Dowel Pin Specifications



Calculation B2: Center Rod & Swinging Arm Buckling Load

Long or Short Column?

long: $\frac{L}{K} > \left(\frac{L}{K}\right)_1 = \sqrt{\frac{2C\pi^2 E}{S_y}}$ | $K = \sqrt{\frac{I}{A}}$ $I = \frac{\pi}{64}(d_2^4 - d_1^4)$

$A = \frac{\pi}{4}(d_2^2 - d_1^2)$

Center Support Rod

$d_2 = 1''$ $d_1 = 5/8''$ $L = 27.75''$ $I = \frac{\pi}{64}((1in)^4 - (5/8 in)^4) = .0416\ in^4$

$A = \frac{\pi}{4}((1)^2 - (5/8)^2) = .4786\ in^2$ $K = \sqrt{\frac{.0416\ in^4}{.4786\ in^2}} = .2948\ in$ $S_y = 75{,}000\ psi$

$\frac{L}{K} = 94.1278\ in < \left(\frac{L}{K}\right)_1 = \sqrt{\frac{2(1)\pi^2(27{,}900\ ksi)}{(75{,}000\ psi)}} = 85.69$

∴ Long Column:

$P_{crit} = \frac{C\pi^2 EI}{L^2} = \frac{1(\pi^2)(27{,}900{,}000)(.0416\ in^4)}{(27.75)^2} =$

$\boxed{P_{crit} = 14875\ lb}$

→ greatest axial force is $R_1 \sin 45 = 1{,}011.02$ lb.

→ will not buckle ✓

Swinging Support Rod

$d_2 = 3/4''$ $d_1 = 3/8''$ $L = 26.25''$ $I = \frac{\pi}{64}((3/4)^4 - (3/8)^4) = .0146\ in^4$

$E = 27{,}900{,}000\ psi$ $S_y = 75{,}000\ psi$ $A = \frac{\pi}{4}((3/4)^2 - (3/8)^2) = .3313\ in^2$

$K = \sqrt{\frac{.0146\ in^4}{.3313\ in^2}} = .2096\ in$

$\frac{L}{K} = 125.22\ in > \left(\frac{L}{K}\right)_1 = \sqrt{\frac{2(1)\pi^2(27{,}900\ ksi)}{4\ 75000\ psi}} = 85.69$

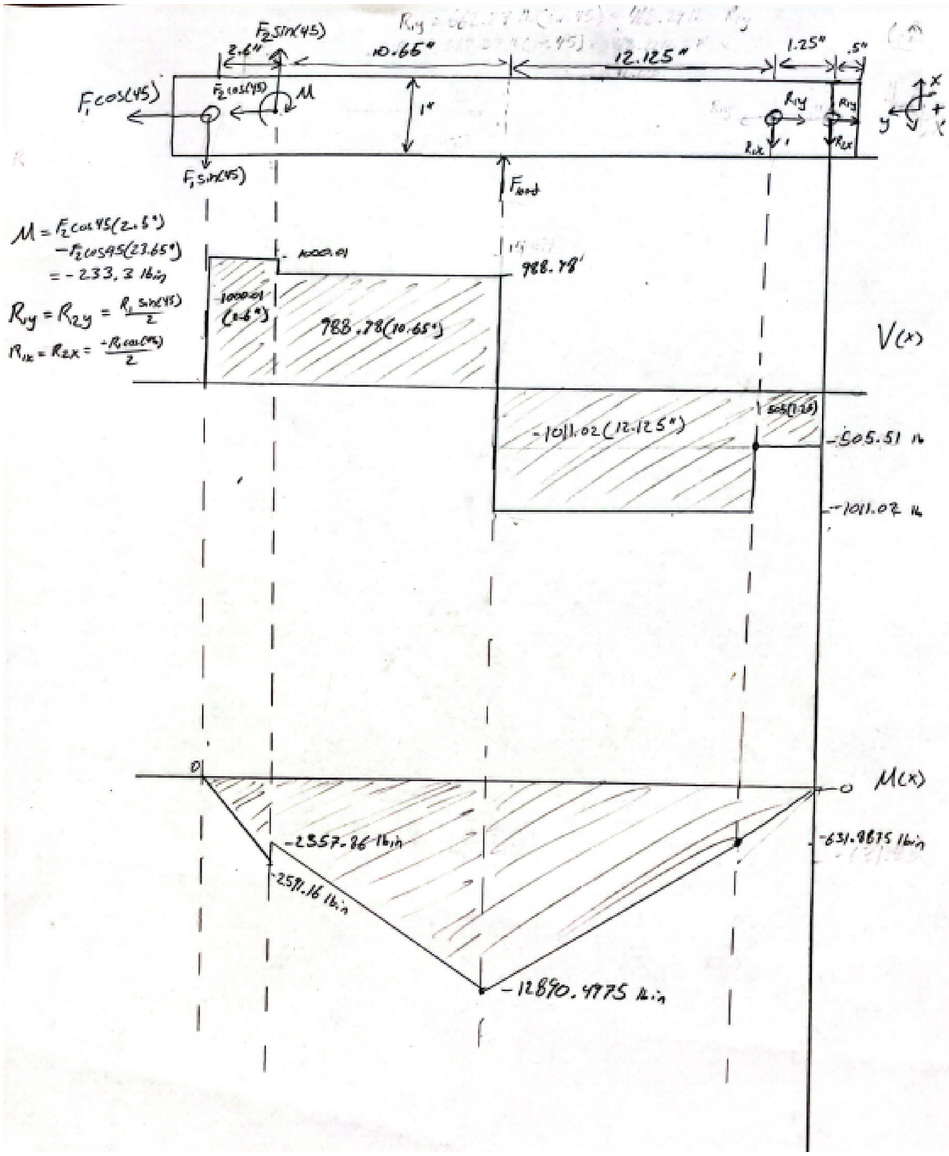
∴ Long column:

$P_{crit} = \frac{C\pi^2 EI}{L^2} = \frac{1(\pi^2)(27{,}900{,}000\ psi)(.0146\ in^4)}{(26.25)^2}$

$\boxed{P_{crit} = 5{,}818.8\ lb}$ → greatest axial force is $f_1 = \frac{1414.2\ lb}{2} = 707.1$ lb. (2 rods for swinging arm)

→ will not buckle ✓

Calculation B3: Shear Force & Bending Moment Diagrams of Emergency Kickstand Center Rod



Center Rod Support

axial

$$\sigma_{axial} = K_t \sigma_o = K_t\left(\frac{4F}{\pi(d_2^2 - d_1^2)}\right) \qquad \sigma_o = \frac{4(505.51\ lb)}{\pi(1^2 - (5/8)^2)} = 1056.2\ psi$$

interpolate $\frac{.9 - .625}{3.8 - K_t} = \frac{.9 - .59}{3.8 - 3.5}$

$\rightarrow K_t = 3.533$

$\sigma_{axial} = 3.533(1056.2\ psi) = 3731.63\ psi$

bending

$$\sigma_{bending} = K_t \sigma_o = K_t\left(\frac{32 M d_2}{\pi(d_2^2 - d_1^2)}\right) \qquad \sigma_o = \frac{32(631.8)}{\pi(1^2 - .625^2)} = 10560.7\ psi$$

interpolate: $\frac{.93 - .625}{3.6 - K_t} = \frac{.93 - .6}{3.6 - 3.4}$

$\rightarrow K_t = 3.41$

$\sigma_{bending} = 3.41(10560.7) = 36011.987$

Shear due to bending

$\tau_{xy} = \frac{2V}{A} = \frac{2(505.1)}{.4786\ in^2} = 2112.45\ psi$

Von Mises Failure Theory

$\sigma_x = \sigma_{bending,x} = 36011.987$

$\sigma_y = 0$

$\tau_{xy} = 2112.45\ psi$

$$\sigma' = \frac{1}{\sqrt{2}}\sqrt{(\sigma_x - \cancel{\sigma_y})^2 + (\cancel{\sigma_y} - \cancel{\sigma_z})^2 + (\cancel{\sigma_z - \sigma_x})^2 + 6(\tau_{xy}^2 + \cancel{\tau_{yz}^2} + \cancel{\tau_{zx}^2})}$$

$$= \frac{1}{\sqrt{2}}\sqrt{(36011.987)^2 + 6(2112.45)^2}$$

$\sigma' = 36104\ psi$

$\sigma' = 36104\ psi \leq \frac{S_y}{2} = \frac{75,000\ psi}{2}$

✓ will not fail

References

1. Garside, M. “Wallboard Products Sold U.S. 2020.” Statista, 25 Nov. 2021, https://www.statista.com/statistics/219348/us-sales-of-wallboard-products/.
2. “Drywall Makes Its Mark on American Construction.” Sioux City Journal, 31 July 2003, https://siouxcityjournal.com/special-section/local/drywall-makes-its-mark-on-american-construction/article_613518f1-e1bd-58f9-a8f7-ffe43ce3d48d.html#:~:text=Today%2C%20drywall%20(also%20known%20as,all%20U.S.%20interior%20wall%20construction.
3. “DC-2020-P.” Adapa USA, https://adapausa.com/product/dc-2020-p/.
4. “Gypsum board industry growth, size, share market: 2022 - 27: Industry share, size, growth - mordor intelligence,” Gypsum Board Industry Growth, Size, Share Market | 2022 - 27 | Industry Share, Size, Growth - Mordor Intelligence, 2022. [Online]. Available: https://www.mordorintelligence.com/industry-reports/gypsum-board-market. [Accessed: 01-Feb-2022].
5. J. Sie, “AN EVALUATION OF MANUAL MATERIALS HANDLING OF DRYWALL MATERIALS USING DRYWALL CARTS AT TAMARACK MATERIALS, INC.,” An evaluation of manual materials handling of drywall materials using drywall carts at Tamarack Materials, Inc., Dec-2006. [Online]. Available: https://minds.wisconsin.edu/handle/1793/42163. [Accessed: 01-Feb-2022].
6. Toolbox Talk: A-Frame Rolling Carts, aka “Drywall Cart,” 21-Nov-2018. [Online]. Available: http://www.island-acoustics.com/Files/f2b08f41-abe5-4f81-aaa3-47ee1630a25d/A%20Frame%20Rolling%20Cart%20Safety%2020181121.pdf. [Accessed: 09-Feb-2022].
7. “AISI 1010 Carbon Steel (UNS G10100),” AZO Materials, 21-Sept-2012. [Online]. Available: https://www.azom.com/article.aspx?ArticleID=6539
8. R. Budynas and J. Nisbett, *Shigley's Mechanical Engineering Design*, 9th ed., McGraw-Hill Education, 2011
9. “AISI 1045 Medium Carbon Steel, ” AZO Materials, 5-July-2012. [Online]. Available: https://www.azom.com/article.aspx?ArticleID=6130
10. https://assets.gates.com/content/dam/gates/home/knowledge-center/mectrol/catalogs/urethane-belt-program-catalog.pdf