TEAM 7 FINAL DESIGN REPORT

Junior Design Project

ME 387

Team 7
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Executive Summary:

Georgia-Pacific currently relies on manual handling to move large polymer rolls used in toilet paper packaging. The polymer rolls are heavy and difficult to maneuver, creating many safety risks for operators. To create a safer and more efficient design, this report focuses on designing and fabricating a scaled prototype capable of safely transporting and positioning a heavy cylinder without requiring excessive force from the operator.

The goal of the project was to design a low-cost, single-operator device that safely lifts a heavy steel cylinder from a pallet, easily maneuvers it through narrow factory settings, and lowers it into a machine receptacle. The prototype serves as a proof of concept for a full-scale solution and was designed while holding safety, ease of use, and reliability as paramount priorities.

The final design utilizes the mechanical advantage of a worm gear to lift and lower the cylinder with minimal force from the operator. The cylinder is supported throughout transport to prevent damage and mitigate safety risks. Caster wheels allow the device to easily navigate tight spaces, while counterweights prevent tipping during loading and unloading. The device is compact, stable, and easily operated by a single person.

Testing the prototype showed that all major requirements were met. The device completed the simulated course in under four minutes, required minimal input force from the operator, and transferred the cylinder without damaging it. During testing, no failure or yielding was observed.

The total final cost of the prototype was $56.23, which was below the $65 limit. Most components were manufactured using standard shop equipment and commonly available materials, supporting this design's scalability and ease of fabrication. Some components were ordered and required lead time, which must be considered for future iterations.

Potential risks include wear of moving components, and due to the sensitivity of the gear alignment, the operating force can be greatly affected. These issues can easily be addressed by design improvements like optimized gears or stronger structural components to ensure correct alignment under load.

Overall, this prototype demonstrates a practical and effective solution for safely handling the heavy cylinder. The device reduces the physical effort required to move the cylinder, increases efficiency, and improves operator safety, making it a good candidate for further development and full-scale implementation.

Problem Statement:

Georgia-Pacific, a major manufacturer of paper products, must safely and efficiently handle large polymer rolls used in packaging toilet paper. Each roll is approximately 39 inches long, 20 inches in diameter, and weighs about 200 pounds, which makes manual handling unsafe and inefficient. The current overhead hoist system is not ergonomic, exposes workers to injury risk, and slows production. To explore potential solutions, Georgia-Pacific has asked for a scale model project where engineers design a mechanism to transport, maneuver, and position a steel cylinder measuring 4 inches in diameter, 10 inches in length, and weighing about 35 pounds. This scaled prototype allows engineers to demonstrate safe handling principles, test ergonomic and functional features, and deliver a reproducible design supported by a digital model. A successful solution must improve safety, minimize human exertion, reduce downtime, and be adaptable, reliable, and cost-effective, with the scale model serving as a proof of concept for eventual full-scale implementation.

Stakeholder Analysis:

- Operators
    - Aspects that help
        - The solution will be safe for the operators to use.
        - The solution requires minimal human effort.
        - Save time so they can focus more on other aspects of their jobs.
    - Aspects that hinder
        - Solution takes a long time to load and unload
        - They must learn how to operate a new system.
        - The solution cannot easily turn around.
- Maintenance Technicians
    - Aspects that help
        - The solution has little to no parts that need maintenance.
        - The solution consists of easily replaceable parts.
        - The solution includes in-depth documentation.
    - Aspects that hinder
        - They must learn how to maintain the solution since it is new.
        - The solution is complex with many easily breakable parts.
- Managers
    - Aspects that help
        - The solution improves process efficiency.
        - The solution can be operated by one person.
    - Aspects that hinder
        - The solution damages the product.
        - The solution is too bulky to store or remove from the floor when not in use.
        - Integrating the solution requires additional team training.
- Georgia Pacific/EIS
    - Aspects that help
        - The solution will make the process faster, allowing for more products to be created throughout the year.
        - Reduces the probability of workplace injury.
        - The solution is easy to manufacture or has a contract to be manufactured.
    - Aspects that hinder
        - The solution may be expensive.
        - The solution will require maintenance and may need to be replaced if it fails prematurely.
        - The solution requires hiring more people for the operation.

Benchmarking:

Pallet Jack Image [1] CASLAD 5_ton	Pros ● Can lift heavy weights ● Maneuverable ● Simple	Cons ● Too high for this application ● Roll could fall off. ● May not fit into the corridor
		
The pallet jack would work in this situation by picking up the pallet on which the polymer roll is placed. This solution is simple and provides a significant mechanical advantage that allows heavy objects to be picked up with minimal force. Since the cylinder must be transferred from a higher pallet to a lower one, this design will not work. While the cylinder may be able to be placed directly onto the design, it may cause damage to the cylinder and be difficult to transfer directly onto the pallet jack.		

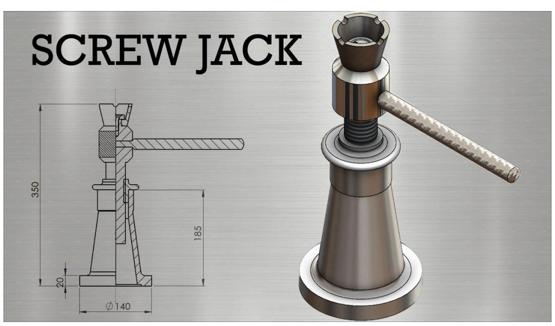
Ice hooks Image	Pros ● Simple ● Can lock ● Grips from above	Cons ● Needs high force to clamp ● May damage paint ● Needs another mechanism to

<table>
<tr><td></td><td></td><td>assist in moving the cylinder.</td></tr>
<tr><td colspan="3">Ice hooks would grip the outside of the cylinder, which could be connected to another system that lifts the cylinder. This is one idea of how to lift and remove it from the upper pallet. An issue with this solution is that the ice hooks may dig into the cylinder and cause damage to the roll. While this solution gives a good idea of how to lift the cylinder, it does not give a complete solution to the design since it does not provide any mechanical advantage.</td></tr>
</table>

<table>
<tr><td>Screw Jack

Image [2]</td><td rowspan="2">Pros

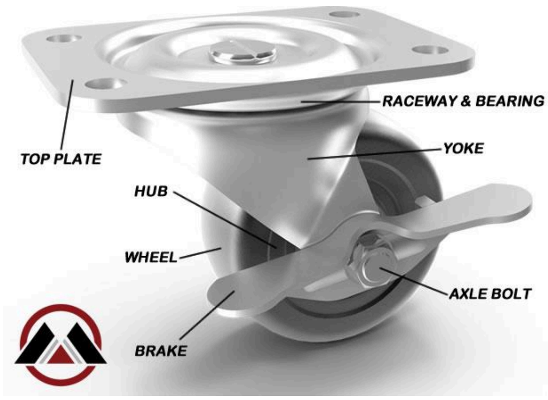
● Can lift heavy weights
● Compact
● Easy to use</td><td rowspan="2">Cons

● Slow
● Cantilevered
● Doesn’t have wheels</td></tr>
<tr><td>
</td></tr>
<tr><td colspan="3">The screw jack is a viable solution because it is able to lift heavy weights with minimal human input. This would be a component of the final solution, as it would need to be combined with other components that would allow it to be maneuvered. It will also have to have some sort of plate or platform that goes under the cylinder and is subsequently lifted by the screw jack. However, the inherent flaw here is that the cylinder will be cantilevered, and it's quite heavy.</td></tr>
</table>

<table>
<tr><td>Casters

Image [3]</td><td rowspan="2">Pros

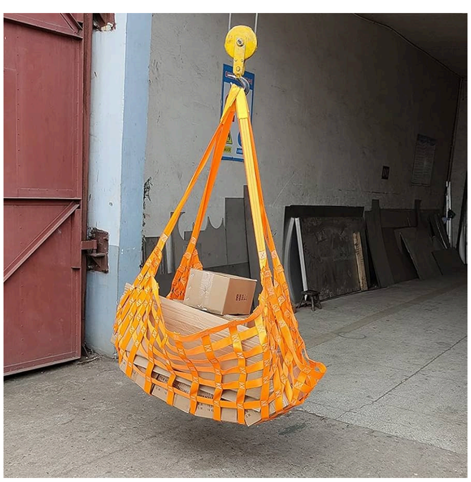
● Commonly used and cheap
● Can carry a lot of weight
● Can lock</td><td rowspan="2">Cons

● Tall
● Flat surface, so the roll may fall off easily</td></tr>
<tr><td>
</td></tr>
<tr><td colspan="3">Casters would act as a component of the project that would help maneuver the cylinder. This would help the object be moved from one location to the next, but is missing the lifting component. This will need to be combined with another system that will lift the cylinder off of the pallet and the casters can be placed on the bottom to get the solution to move.</td></tr>
</table>

<table>
<tr><td>Sling Hoist

Product [4]</td><td rowspan="2">Pros

● Can lift 1500 pounds
● Uses a pulley, so has a mechanical advantage
● Can be easily replaced as it wears</td><td rowspan="2">Cons

● Would be hard to get the roll onto
● The roll could potentially slip through the net
● Doesn't have wheels so it would be hard to maneuver once it is lifted</td></tr>
<tr><td></td></tr>
</table>

This design is useful because it lifts a lot of weight very easily and quickly, which would be helpful in the case of the cylinder. This is not a complete solution to the problem as it would need to be combined with a component that would maneuver the cylinder. The downside is there would have to be a track for the hoist to move along or it would have to be on wheels, because it would be too heavy to drag. Another consideration is that the cylinder could fall through the netting.

<table>
<tr><td>Floor Jack

Product [5]</td><td rowspan="2">Pros

● Can easily lift heavy weight
● Very fast lifting and setting down
● Requires little human force</td><td rowspan="2">Cons

● Not easy to maneuver under load
● Unstable for something that rolls
● Expensive and intricate</td></tr>
<tr><td></td></tr>
<tr><td colspan="3">The floor jack provides a solution to how to lift the cylinder with minimal force. The floor jack moves quickly, which helps with time constraints. One of the downfalls of this design is that it is stationary and not easily moveable. This component would need to be combined with wheels or another device that would allow it to be moved. It also may be too large to fit between the pallet and cylinder.</td></tr>
</table>

Dolly Product [6]	Pros ● Easy to maneuver when loaded ● Will be fairly light during transportation ● Once on there, the load is fairly secure	Cons ● May be difficult/unsafe to load/unload ● Could damage the payload during loading/unloading

<table>
<tr>
<td></td>
<td></td>
<td>
Cylinder may roll off while being transported
</td>
</tr>
<tr>
<td colspan="3">A dolly provides good mechanical advantage, and would help the cylinder be picked up with minimal force. This solution would be a very good idea for just the transportation aspect. With a cylinder, the loading would be difficult, but safe and precise unloading is almost impossible, especially with a 200 pound object.</td>
</tr>
</table>

<table>
<tr>
<td>Machinery Mover Dolly

Product [7]</td>
<td rowspan="2">Pros
Very easy to maneuver
Can withstand very heavy loads
Would fit in small corridors
</td>
<td rowspan="2">Cons
May not fix the problem, no easy way to load
Unstable for a cylinder, no walls
No mechanical advantage
</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td colspan="3">This product looks really good, it's thin, moveable, and can withstand weight. It being thin would allow it to fit between the cylinder and pallet, making it easier to be picked up. The downside is that it would be difficult to get the cylinder onto the solution and off of it. The cylinder also may roll off of it during transport.</td>
</tr>
</table>

Needs:

Need ID	Description	Justification	Required
1	Solution does not require manual labor to lift the steel cylinder.	Operator safety - Minimizes heavy lifting	Y
2	Solution is safe to operate.	General operator safety.	Y
3	Solution must safely handle the weight of the steel cylinder.	Operator safety - Design and materials can withstand the loads.	Y
4	Solution collects the steel cylinder from the incoming pallet.	Primary function of the solution.	Y
5	Solution capable of transporting the steel cylinder through the factory corridors.	Primary function of the solution.	Y
6	Solution deposits the steel cylinder into the machine receptacle.	Primary function of the solution.	Y
7	Solution can be maneuvered and controlled by a single operator.	Ergonomics and safe handling.	Y
8	The steel cylinder should not touch the floor at any time.	Protect the product from dirt and damage.	Y
9	The solution cannot damage the cylinder (no paint damage).	Protect the product from damage.	Y
10	Solution must be within budget.	Project requirement.	Y
11	Solution must comply with course guidelines and expectations.	Academic requirement.	Y
12	Solution must be within EIS capabilities and guidelines.	Academic requirement.	Y

Background Information and Research

Georgia Pacific is a manufacturer of paper products such as Brawny, Dixie Cups, Kirkland Signature, and other household brands that sell paper products. The production of these products involves plastic wrap for final packaging, which is delivered to the manufacturing facilities on a pallet in a large, heavy roll (approximately 200 pounds) that must be transported to the wrapping machine. Currently, the shop has no mechanism except manual lifting to transport the 200lbs roll to the machine, creating ergonomic, efficiency, and safety issues.

Figure [1]: Actual roll in Georgia Pacific Warehouse

That process usually includes two workers lifting, traversing, and loading the roll without any mechanical help. Georgia Pacific manufacturing prioritizes efficient operation, meaning downtime or difficult processes slow production and increase the risk of injury. Research on handling methods of plastic film shows that difficult loading processes can lead to contamination, damage, and worker strain. The plastic is sensitive to damage, which emphasizes the necessity of proper handling.

The workspace available for handling is tight and constrained. The production floor has a narrow track that is to be navigated while handling the plastic. Our project's 16-inch wide track and tight 90-degree turns emulate this feature, restricting the vehicle's overall area and turning radius.

The loading of the polymer roll is an integral part of the toilet paper packaging process. Before the packaging process, the toilet paper is created. To create toilet paper, wood chips are used to make pulp. This pulp is washed, dried, bleached, wrapped, and then moved into the paper machine to make toilet paper. The parent rolls are embossed, perforated, and rewound into smaller toilet paper rolls. Once the rolls have been cut, they are then wrapped in the polymer that is loaded onto the machine via the solution [9]. The solution must deliver the polymer to the machine promptly to ensure continuity in the toilet paper manufacturing process.

Figure [2]: A flow wrapping machine where the roll is loaded

The machine that the polymer roll is loaded onto is likely a flow wrapper. This machine works by unrolling the polymer roll using guided rollers, which keep it in consistent tension. The conveyor belt moves the toilet paper into the wrapping section, where the polymer is formed around the rolls. Once they are wrapped, the machine uses heat to seal the package and a blade to cut the package away from the other ones [10].

Problem Analysis

The problem of moving a steel cylinder is rather simple in the grand scheme of things, yet there are still a few calculations that can be made regarding the solution while remaining solution-neutral. One of the specifications is that the solution may take no more than four minutes to complete the course. Using SolidWorks, an example path is laid out through the course whose length is 244in. (See Figure 3)

To move along the 244” course at a walking pace of 3.0 mph [8] it would take:

$$\frac{3mi}{hr} * \frac{1\,hr}{60\,min} * \frac{1\,min}{60\,s} * \frac{5280\,ft}{1\,mi} * \frac{12\,in}{1\,ft} * \frac{1}{244\,in} = .216\,s^{-1} = 4.62\,s$$

However, this figure assumes the operator is simply walking unencumbered, which will not be the case. It is impossible to know how much operating the solution will slow a human, but to make a very conservative estimate, we could assume that the operator will be slowed down by a factor of 12.9x, bringing the course traversal time to one minute. This leaves 1.5 mins to load the cylinder and as much time to unload it.

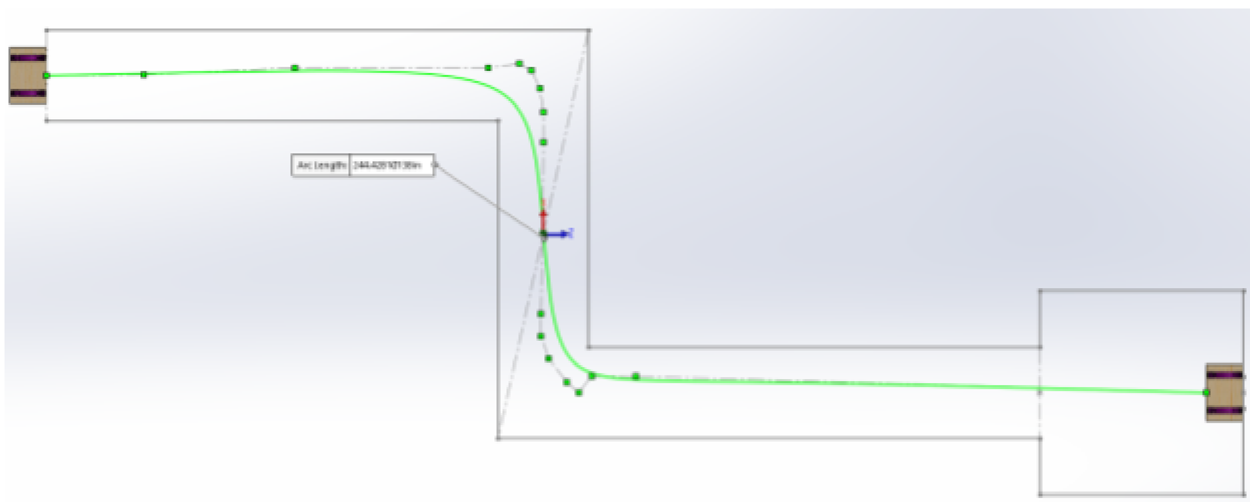


Figure [3]: The course in solidworks with a path of travel drawn and length calculated at 244.428”

Another specification is that the operator can exert no more than 15 lbs of force at any time to move the cylinder. However, the cylinder weighs 35 lbs, so at least some mechanical advantage is needed. Below, we calculate the minimum amount of mechanical advantage needed to lift the cylinder directly up and with a moment arm of length L.

To determine the lifting mechanical advantage η_L, we start by summing the vertical forces, as shown to

Fig. 2: $\sum F_y = F - mg = 0$

$$F = mg = 35\,lbf$$

With an allowed lifting force $F_{allow} = 15\,lbf$

$$\eta_L = \frac{F}{F_{allow}} * (FOS) = \frac{35\,lbf}{15lbf} * 1.5 = 3.5$$

For the moment, case summing moments without the page being positive, we have

$$\sum M_z = -mgL = 0$$

$$M = mgL$$

With L_1 being the moment arm with which M is applied, the minimum mechanical advantage(η_M) is:

$$F_{tot}L_1 = M = mgL$$

$$F_{tot} = \frac{35lbf * L}{L_1}$$

$$\eta_M = \frac{F_{tot}}{F_{allow}} * (FOS) = \frac{\frac{35lbf * L}{L_1}}{F_{allow}} (FOS) = \frac{35lbf * L}{15\,lbf * L_1} * 1.5 = 3.5\frac{L}{L_1}$$

In conclusion, pushing the solution across the course should take a negligible amount of time compared to unloading and loading the cylinder.

As for mechanical advantage, including the factor of safety to lift the cylinder directly, you need 3.5:1. This can be achieved in a variety of ways, gears, pulleys, springs, etc. Another common case could be utilizing an arm to lift the cylinder. When using an arm, you need a mechanical advantage of $3.5\frac{L_1}{L_2}$ which $\frac{L_1}{L_2}$ is the ratio of moment arms utilized. Below is a collection of equations that, while not used in this analysis, are generally useful for determining deflections and stresses in materials subject to the geometry of the solution.

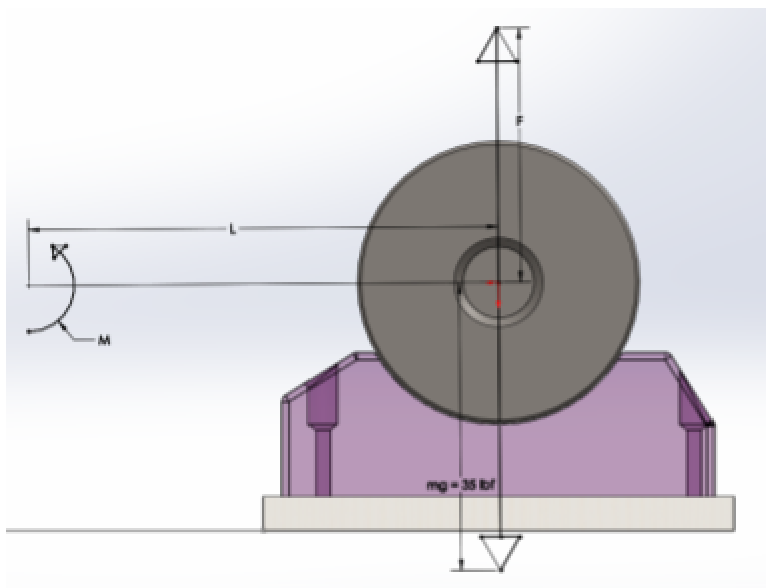


Figure [4]: Free body diagram of forces acting on steel cylinder when lifted directly up or via a moment arm.

Useful equations

Moment of Inertia of a rectangle of height *h* and width *b*: $I_x = \frac{bh^3}{12}$ $I_y = \frac{b^3h}{12}$ where the y axis is along *h,* and the x axis is along *b.*

Moment of Inertia of a circle: $I_x = I_y = \frac{\pi r^4}{4}$ $J = I_z = \frac{\pi r^4}{2}$ Axial Stress: $\sigma = \frac{F}{A}$

Principal stresses: $\sigma_1, \sigma_2 = \frac{\sigma_x+\sigma_y}{2} \pm \sqrt{(\frac{\sigma_x-\sigma_y}{2})^2 + \tau_{xy}^2}$ Max Torsion: $\tau_{max} = \frac{Tr}{J}$

Direction of max shear: $tan(2\phi_s) = -\frac{\sigma_x-\sigma_y}{2\tau_{xy}}$ Max Bending: $\sigma_{max} = \frac{Mc}{I}$

Shear stress transformation: $\tau = -\frac{\sigma_x - \sigma_y}{2} sin(2\phi_s) + \tau_{xy} cos(2\phi_s)$

Deflection along beam axial: $\delta_i = \int \frac{1}{AE} (F\frac{\partial T}{\partial F_i})dx$

Deflection along beam bending: $\delta_i = \int \frac{1}{EI} (M\frac{\partial M}{\partial F_i})dx$

Deflection along curved beam bending: $\delta_i = \int \frac{1}{EI} (M\frac{\partial M}{\partial F_i})Rd\theta$

Standards & Regulations:

These are standards and regulations must be complied with when creating our solution.

ISO 12100 [11]

International Organization for Standardization

- A foundational international standard for the safe design of machinery that provides a methodology for identifying hazards, assessing risks, and implementing risk reduction measures.
    - https://webstore.ansi.org/standards/iso/ISO121002010
    - This standard can impact the design and manufacturing of the actual project via machining, welding, and taking into account all the possible risks associated with the manufacturing process.
- EN ISO 12100 specifies:
    - The methodology for the design of safe machines
    - The application of risk assessment
    - The hierarchical use of risk reduction measures [12]
- This standard uses a three-step process for risk reduction
    - Inherently safe design
    - Safeguarding
    - Information (about use) about residual risks

ISO 11228-1: Lifting and carrying [13]

International Organization for Standardization

- Provides recommended limits for lifting, lowering, and carrying objects.
- Applies to manual handling of objects with a mass of 3 kg (6.6 lbs) or more, moderate walking speeds, and is based on an 8-hour working day, though it covers up to 12-hour shifts.
- Does not cover holding objects without walking, pushing or pulling, manual handling while seated, or handling by more than one person.
- This standard impacts the design by limiting the weight of the object that humans would have to manually lift.
- https://www.iso.org/obp/ui/#iso:std:iso:11228:-1:ed-2:v1:en

ISO 11228-2: Pushing and pulling [14]

International Organization for Standardization

- Specifies recommended limits for whole-body pushing and pulling actions.
- Applies to one-person, whole-body force exertions using two hands, such as moving or restraining an object in an upright position.
- This standard impacts the solution by limiting the amount of force the operator would use to push or pull it.
- https://www.iso.org/standard/89268.html#:~:text=Abstract,tasks%2C%20products%20and%20work%20organization.

AWS D1.1 [15]

American Welding Society

- Covers the welding requirements for welded structures that are constructed of carbon and low-alloy constructional steels.
- It provides criteria for inspectors who are completing non-destructive testing and visual checks of the welds.
- This standard would apply to a solution that needs to be welded together. This would ensure the safety of the welds so that they do not fail under load.
- https://pubs.aws.org/p/2264/d11d11m2025-structural-welding-code-steel?srsltid=AfmBOooHnh4G_W0LR_5WcjHxB9Yr8QWZitD-1HQ4mx90TlNx-djs6sWj

ISO 9001 [16]

International Organization for Standardization

- Outlines requirements for quality management systems, including quality planning, control, assurance, and methods for monitoring and measuring performance.
- Has guidelines for documentation, internal auditing, and corrective action. [17]
- This standard would impact the design process by outlining the documentation process and what needs to be maintained.
- https://www.iso.org/standard/62085.html

Specifications:

Spec ID	Related Need ID	Specification	Justification	Required
A	1, 7	No more than 15 lbs of force is required to maneuver, control, or operate the solution.	Safety - Minimize operator work	Y
B	1, 2	Solution causes zero injuries to the operator or others during use.	Safety requirement	Y
C	1,10	The operator cannot run while using the solution to transport the steel cylinder.	Operator safety.	Y
D	3	All aspects of the design must have a minimum Factor of Safety of 1.5.	Safety requirement	Y
E	4	The solution must remove one steel cylinder from the loading pallet without damaging or tipping the pallet.	Primary function - Loading of the steel cylinder	Y
F	5	The design and its operation navigate the 16-inch-wide factory corridor without "collision."	Primary function - Transport of a steel cylinder	Y
G	5	The design must not remain in the factory corridors after steel cylinder transport.	Primary function - Clear corridors for future use.	Y
H	6	The solution places one steel cylinder onto the machine receptacle.	Primary function - Unload the steel cylinder	Y
I	6	The steel cylinder must be in full contact with the entire surface of both cradles on the unloading platform	Primary function - Safety and alignment in the machine receptacle	Y
J	7	The solution is operated by a single person.	Improved efficiency for the company	Y
K	8	The steel cylinder at no time touches the floor during operation.	The product cannot be damaged during transport	Y
L	8	The paint on the steel cylinder is not damaged during operation (where the metal beneath can be seen).	The product cannot be damaged during transport	Y
M	9	The overall cost of the solution must not exceed $65.	Budget constraint	Y

N	10	Solution must complete transfer within four minutes (Application of force to the solution to exit from the area of use)	Ease of use, Time constraints within class	Y
O	10, 11	Solution must be manufactured in the EIS	Class requirement.	Y
P	10, 11	Proper usage of machines to manufacture or construct a device must be followed	EIS and safety requirements	Y
Q	10,11	The solution must fit within a 14" x 14" x 14" storage container.	Space constraints within the EIS	Y

Design Description:

To form a general understanding of what aspects are necessary to create a successful solution, functional block diagrams were created. These diagrams graphically show the sub-functions the design needs, how they interact, and how they are connected to each other. The top-level diagram shown in Figure 5 outlines the inputs to the design; energy supplied by the operator is used to move the cylinder.

Level Zero Block Diagram:

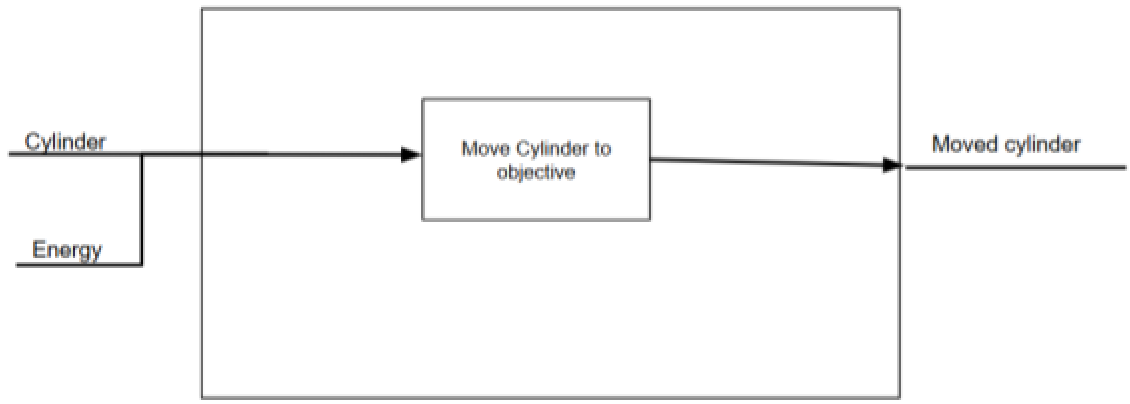


Figure [5]: Level zero block diagram showing necessary inputs and outputs to move the cylinder

The level one diagram, Figure 6, shows the subsystems of the device. This problem is a relatively simple one that has no complex interlacing systems, rendering these diagrams trivial.

Level One Block Diagram

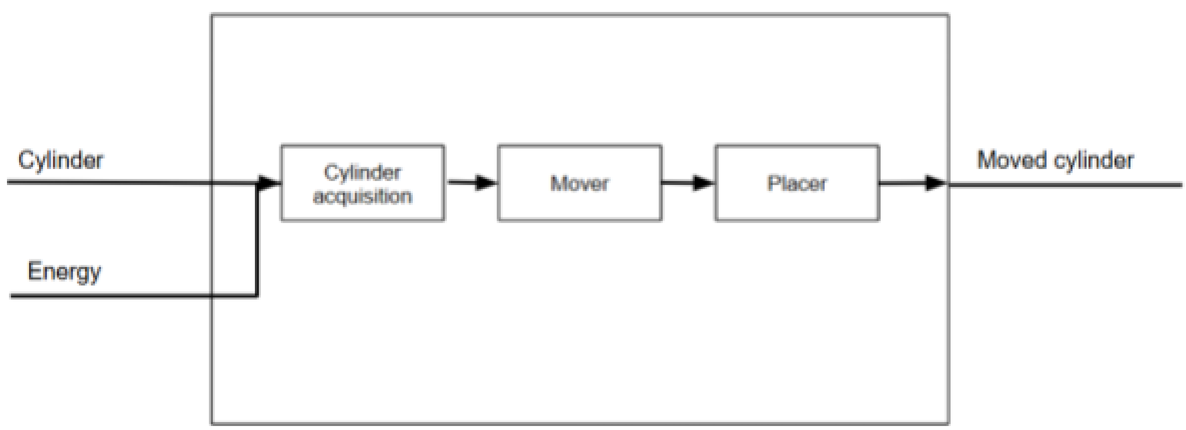


Figure [6]: Level one block diagram showing showing more specific inputs, outputs, and functions of the solution

Design Overview

The premise of the design is that the cylinder is lifted from its ends with Arms that rotate to the upper and lower cradle positions. This necessitates that a source of high torque be supplied. Our solution utilizes a worm Gearbox with a 20:1 gear reduction to achieve this. The operator supplies input via the Main Handle, which directly rotates the Worm Gear. Due to the cantilevered nature of the Arms, without counterweights, the solution would tip over; thus, they are included in the Rear Housing.

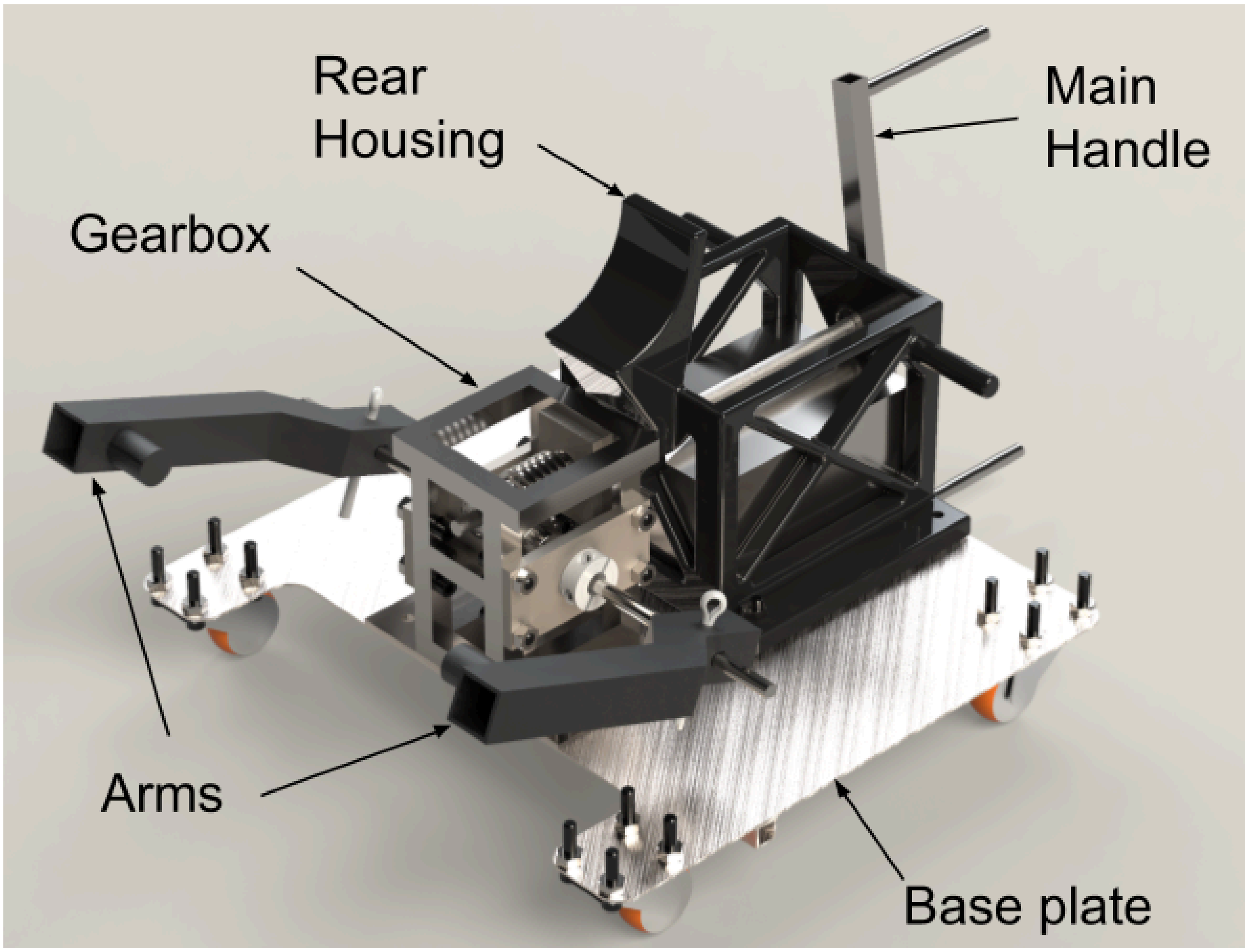


Figure [7]: Cad render of final design

Gearbox

The gearbox has several key components: the gears, welded frame, adjust plates, and bearings. The iterations of the gearbox, shown in Figure 8, represent the on-the-fly design changes we were forced to make. V1 was a simple 3D printed PLA cube with four holes for bearings. It held the gears, but they would slip on the shafts, and the teeth would skip. To fix the shaft slippage, the spur gear was impulsively welded to the main shaft in place, melting the 3D print.

V2 beefed things up and reduced the spur-to-worm distance by one mm to prevent gear slippage. The gearbox was split into three pieces to allow assembly with the welded gear.

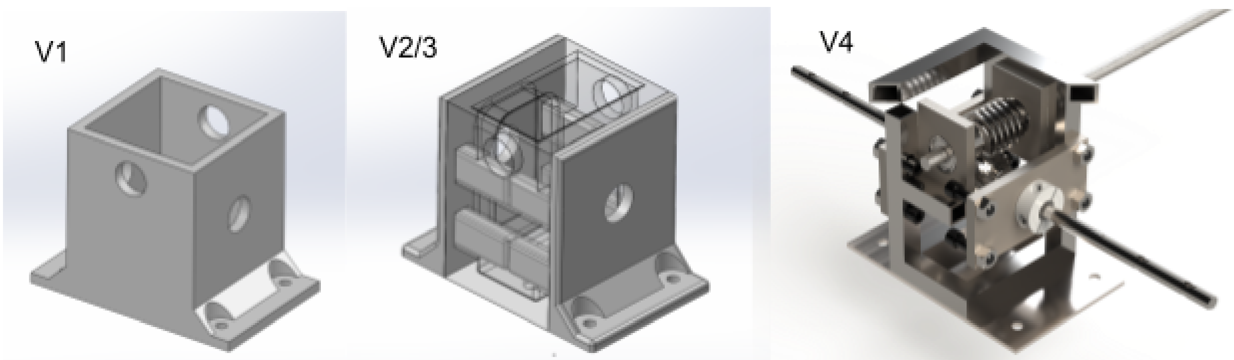


Figure [8]: Progression of gearboxes. V1 (melted) V2 (accommodates welded gear, stronger, but not strong enough) V3 (Same as V2 but PETG and even more beefed up) V4 (precision machined metal)

V2 failed in tension when we attempted to raise the cylinder, so for V3, the center piece was resliced stronger and printed in a new orientation out of PETG, drastically improving strength. We were unable to perform analysis on 3D printed parts due to their anisotropic nature, leaving experimental validation as our only option. Additionally, the one mm change from V1 to V2 was modified to 0.5mm because V2 had overly stiff gears. V3 did not fail, and many tests were performed. It was found that worm gears **must** be exactly aligned to work, the shafts must be perpendicular to each other, must not deflect, the bearings must be precisely aligned, etc. Because of this, it was decided to make the gearbox entirely out of metal.

The final version, V4 (Figure 9), was designed to be manufacturable in the short period of time available, with the resources at our

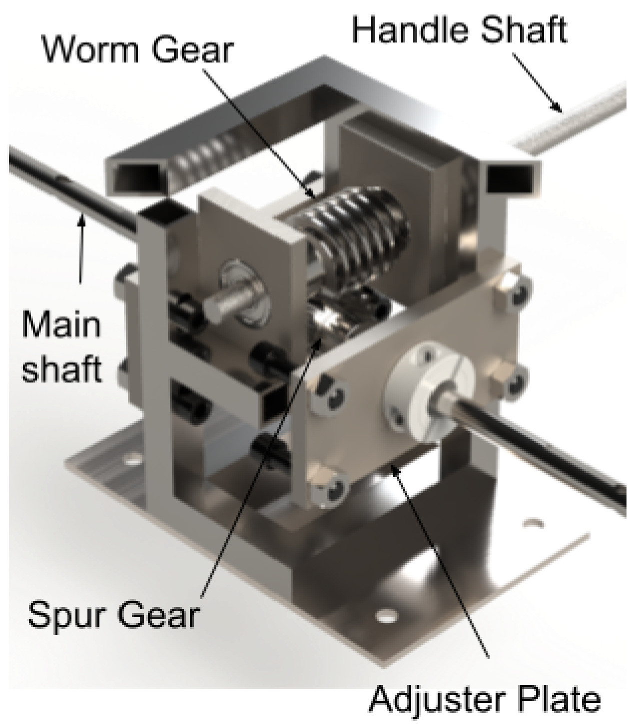


Figure [9]: Detail view of final gearbox. Parts of frame hidden to reveal internals.

disposal, while maintaining the highest level of precision as realistically achievable.

It consists of three main custom parts: The main frame and the two Adjuster Plates. The main frame is welded together from many 0.5” box tubes and two ¼” plates. The plates are precision bored after welding to enable press fitting the bearings. For details on how accuracy was maintained while creating the gearbox, see Appendix D.

The Adjuster Plates are simple, precision-machined ¼” steel plates that hold the bearings for the main shaft. Because the gear is welded on, these plates must be removable. The plates are bolted into slots, allowing the gear mesh distance to be adjusted.

The Worm Gear rides on the Handle Shaft, and because it is not welded on, flats are milled on the handle shaft so the setscrew can effectively prevent slippage. When in use, the Worm Gear is forced against the sides of the gearbox, causing immense friction and wear and tear. To mitigate this, an HDPE (High-Density Polyethylene) bushing was created and placed between the Worm Gear and the side of the gearbox. When a load is applied to the Main Shaft, the Spur Gear loads the Worm Gear, pushing it axially against the HDPE bushing, which is well greased. A thrust bearing would be the ideal solution here, but this was the best that could be produced with the materials on hand. (See Figure 10)

Figure [10] The HDPE bushing

To keep the Spur Gear in line with the Worm Gear, which is crucial for proper gear action, shaft collars are used on the outsides of the Adjuster Plates. The Handle Shaft retention is achieved by the setscrew in the worm gear and bushing.

The gearbox is welded to a 16-gauge plate, which is then bolted to the existing pattern already on the Base Plate.

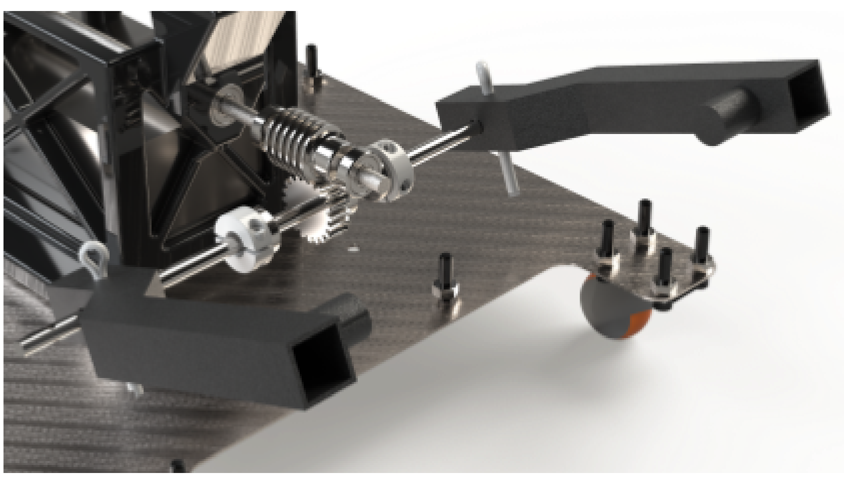


Figure [11]: The Arms and gears.

Arms

In Figure 11, we see the Arms that lift the cylinder. They are made from 1” steel box tube welded together with some barstock welded on the end as a pin for the cylinder to engage with. The Arms have slots machined into them, which interface with the Main Shaft and are the means of torque transfer. The Main Shaft has matching flats machined into it (Figure 12,) which allow for the arm to translate along the shaft, secured by cotter pins. At the beginning and end of the run, at least one arm has its pin removed and is slid out

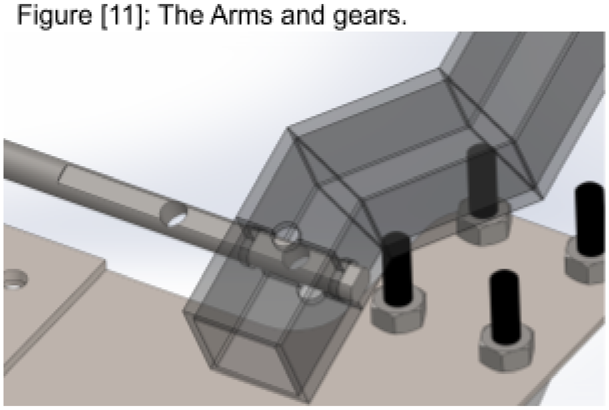


Figure [12]: Arm attachment

to accommodate the width of the cylinder. When the cylinder is lined up, the arm is slid back into the closed position and pinned. The slot attachments on the arms did undergo significant wear and tear. By test day, the left arm had degraded enough that it was decided to bypass the slot attachment and weld it in place just before the test, leaving the right arm to slide around the cylinder. For future iterations, it would be wise to create a force distribution block that fits inside the arm and engages with the shaft, distributing the load along the inner surfaces of the arm instead of just the walls.

Rear Support and Handle

The rear support is a large 3D printed part that serves four purposes: retains the handle shaft, contains the counterweights, supports the cylinder in its resting position, and provides handles for the operator to push the solution.

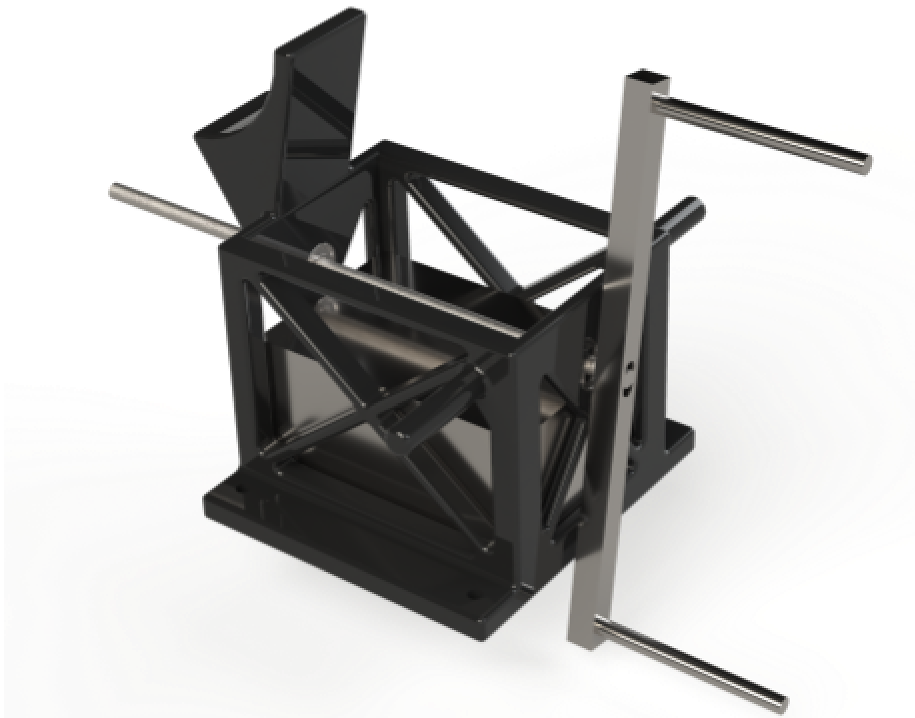


Figure [13]: Rear support, counter weights, and handle.

The bearings are pressed by hand into the part, as they don't need to be aligned. The counterweights sit in the basket, which is a skeleton to reduce the 3D print cost. The piece on the front supports the cylinder in its resting position. During testing, the center of the cylinder did not end up in the same position as the center of the 3D print’s arc, but there was no need to re-print it. The handles on the back give the operator somewhere to grip when pushing the solution along the course.

The Handle Shaft utilizes the same slot attachment pattern as the main arms and has the same issues, but exaggerated. Because the Handle Shaft is made from a 0.5” steel box, it is more susceptible to stripping than the arms; however, it still lasts quite a while. The handle is as long as possible to enable the operator to use as little force as possible. Additionally, through testing, having a double-sided handle instead of a single-sided handle significantly reduces the amount of resistance encountered by eliminating bending loads in the shaft while operating.

Base Plate

The base plate is a simple 16GA sheet steel plate with casters welded on it. It has bolt holes popped all over the place to mount everything that needs mounting. The 16-gauge sheet metal is insufficient to keep everything in place, so 0.5” box tubes are welded on the bottom to reinforce it.

Design Analysis:

There are several key specifications that analysis can validate. Firstly, Spec A requires that the design must require no more than 15lbs of force, Spec D that the safety factor be at least 1.5, and finally, Spec K, the cylinder cannot touch the floor during transportation. Four studies are presented:

- Operator input analysis
- Tipping moment analysis
- Handle shaft safety factor FEA analysis
- Plate deformation FEA analysis.

Operator input analysis

The goal of this analysis is to prove compliance with specification A, that the force applied by the operator may not exceed 15lbf. This analysis determines the force applied to the handle as a function of the weight of the steel cylinder, the gear ratio, the friction of the gears meshing with each other, and the friction of the worm gear pushing against the HDPE bushing in the gearbox.

This analysis assumes:

- Friction between the worm and spur gear is accurately modeled without utilizing pitch angle.
- Bearings are frictionless
- Shafts are straight

Considerable uncertainty is inevitably involved when utilizing friction coefficients. We mitigate this by sourcing friction data from reliable sources and evaluating our model with various values to characterize the magnitude of uncertainty.

This analysis will build up the calculation in three parts: frictionless, with gear friction, and with all friction. In Figure 14 below, we see the structure of the problem and various lengths are defined. Any steps that are skipped or specific values that are left out can be found in Appendix A.

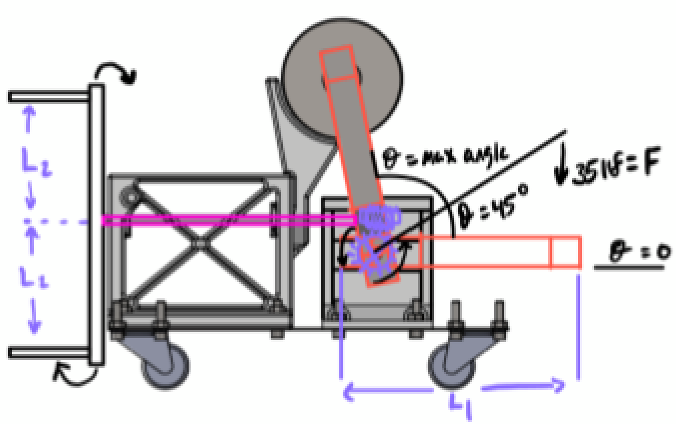


Figure [14]: Diagram of moment analysis parameters.
Gears shown in purple, arms in red, handle shaft in pink.

First, we will consider the frictionless case:

Torque at the spur gear: $\tau_1 = Fcos(\theta) \bullet L_1$

Torque at the handle: $\tau_2 = \frac{1}{n}\tau_1$

Force applied by operator: $F_{app,base} = \frac{\tau_2}{L_2} = \frac{Fcos(\theta)L_1}{nL_2}$

The maximum force is seen at $\theta = 0$, evaluating the above expression, we find our baseline value:

$$F_{app,base} = 2.19\ lbf$$

Now we will add in the friction between the worm and spur gears.

In Figure 15 we see what the worm and spur system looks like.

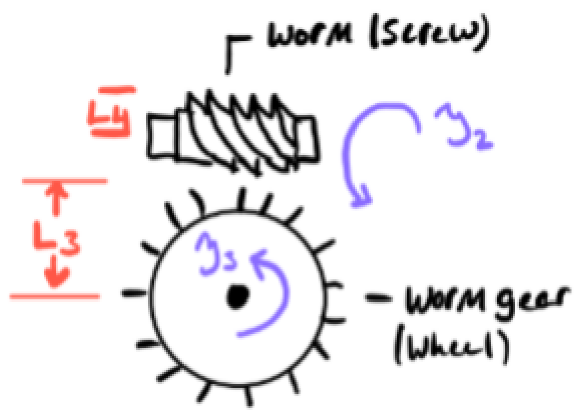


Figure [15]: Detailed view of worm and spur system. Lengths labeled in red and torques in purple.

Friction is related to the normal force between the gear teeth $F_{fr} \equiv \mu N$.

We can find the normal force from τ_1 and the radius as:

$$N = \frac{\tau_1}{L_3} = Fcos(\theta)\frac{L_1}{L_3}$$

This friction force acts in opposition to any motion, increasing the torques needed:

$$\tau_{fr} = F_{fr}L_4 \Rightarrow \tau_2 = \frac{1}{n}\tau_1 + \tau_{fr}$$

Resulting in a revised applied force equation:

$$F_{app,gearonly} = Fcos(\theta)\frac{L_1}{L_2}(\frac{L_3+\mu n L_4}{nL_3})$$

Now, evaluating the above equation at max force with both the greased and ungreased steel-on-steel friction coefficients, we find that:

Greased: $\theta = 0, \mu = .09,\ F_{app} = 5.07lbf$

Dry: $\theta = 0, \mu = .42,\ F_{app} = 15.67lbf$

Now, adding the friction between the bushing and worm gear:

Figure [16]: Friction between worm gear and HDPE bushing.

The effect of this friction is determined in much the same way; however, integration must be utilized because the torque applied to the worm gear varies over the radius of the surfaces in contact.

Integrating friction from ID to OD: $\tau_{bushing,\,fr} = \int_{r1}^{r2}(\mu N r)dr = \mu N\frac{r^2}{2}\Big|_{r1}^{r2}$

Augmenting our prior equation for torque due to friction:

$$\tau_{fr} = F\mu_{st-st}L_4 + \mu_{HDPE-st}N\frac{r^2}{2}\Big|_{r1}^{r2}$$

Substituting in all values, we find that:

$$F_{app} = \frac{\tau_2}{L_2} = Fcos(\theta)\frac{L_1}{L_2}(\frac{1}{n} + \frac{\mu_{st-st}L_4+\mu_{HDPE-st}(\frac{r_2^2}{2}-\frac{r_1^2}{2})}{L_3})$$

Where μ_{st-st} is the friction of the worm on the spu,r and $\mu_{HDPE-st}$ is the friction of the worm on the bushing.

The coefficients of friction are difficult to obtain; for a full discussion on how they were achieved, see Appendix A. In summary, there are three numbers to consider: the HDPE-steel coefficient and the steel-on-steel coefficient for lubricated and non-lubricated cases. Evaluating the equations, we find the following results:

Table [1]: Friction coefficients.

Steel on steel (dry) [18]	μ=0.42
Steel on steel (greased) [18]	μ=0.09
HDPE on steel [19]	μ=0.08

Table [2]: Operator input analysis results. All units lbf. Baseline force (Frictionless) 2.19

	Only Friction between gears	All Friction
Lubricated	5.07	5.191
Non-Lubricated	15.67	15.78

The results of this analysis show that even without a lubricant, the amount of force needed to lift the payload is just slightly over the limit. During testing, a silicone grease lubricant was used, ensuring the force required remains within the specified limits. The factor of safety, calculated using lubricated friction coefficients, is comfortable. However, there is considerable variability in friction coefficients. But the analysis shows that increasing the gear friction coefficient 4.6x from .09 to .42 (lube vs. no lube) brings the FOS to .95. Thus, even accounting for inaccurate friction coefficients, a safety factor >1.5 is highly probable (Spec D), and Spec A is satisfied.

Tipping moment analysis

The goal of this analysis is to determine the weight of needed counterweights in order to prevent tipping when loading the payload onto the loader, which would satisfy spec K.

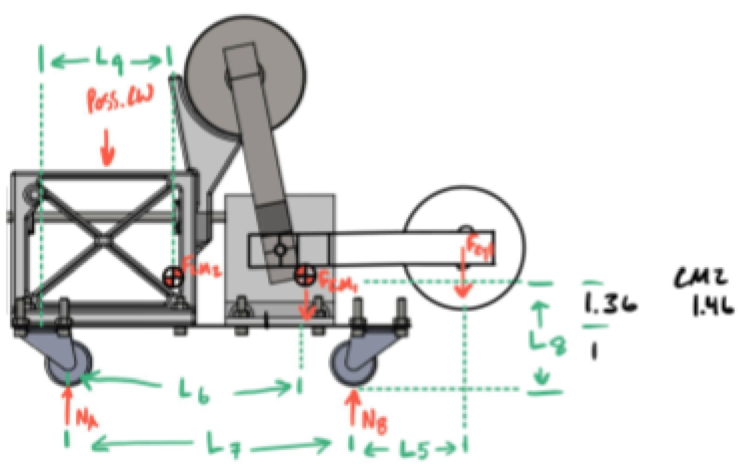


Figure [17]: Lengths in tipping analysis

Knowns:

$L_5 = 2.6in$, $L_6 = 5.9in$, $L_7 = 11in$, $L_8 = 2.36in$, $L_{8-2} = 2.46in$, $L_9 = 3.49in$, $F_{Cyl} = 35lbf$, $F_{Cm1} = 10.39lbf$, & $F_{Cm2} = 25.39lbf$, where F_{Cm} is the force at the center of mass, F_{Cyl} is the force of the cylinder.

Equations & Solutions:

This part is solving without the counterweights in the basket to first determine if it would tip or not.

Sum of moments about the rear wheel: $\sum M_a = - F_{Cm1}(L_6) + N_b(L_7) - F_{Cyl}(L_7 + L_5)$

Solving for the normal force at b, $N_b = 48.85lbf$

Now solving for the normal force at a with a sum of forces: $\sum F_y = - F_{Cm1} + N_b + N_a - F_{Cyl}$

We get $N_a = - 3.46lbf$, meaning that the loader is currently tipping without any counterweights.

Now, solving with 15lbf of counterweights loaded in the center of the rear holder, this shifts the center of mass.

So starting with a sum of the moments again: $\sum M_a = - F_{Cm2}(L_9) + N_b(L_7) - F_{Cyl}(L_7 + L_5)$

We get $N_b = 56.89lbf$

Now solving for the normal force at a: $\sum F_y = - F_{Cm2} + N_b + N_a - F_{Cyl}$

We get $N_a = 3.499lbf$, giving us a positive force, meaning that the loader is no longer tipping over, satisfying spec K.

Handle Shaft Safety Factor FEA Analysis

The goal of this analysis is to characterize the stresses developed in the handle shaft when a torsion is applied to the handle.

Assumptions:

- The only applied force is a force couple on the handle.
- The end of the shaft is fixed
- The shaft is bonded directly to the handle.

These assumptions make the FEA somewhat unreliable. The shaft is not actually fixed, and due to simulation methods, the mesh nodes between the two parts do not align, forcing SolidWorks to perform a hacky force interpolation calculation, which reduces accuracy. Finally, as has already been laid out, the force the user is expected to apply is around 5lbf, while this simulation considers the full 15lbf.

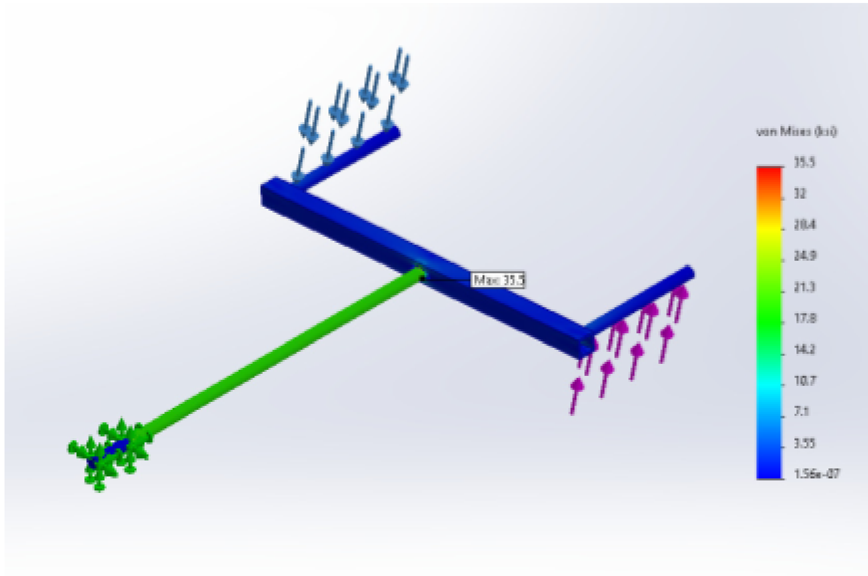


Figure [18]: Handle stress: 19.6 ksi. Min FOS: 1.44

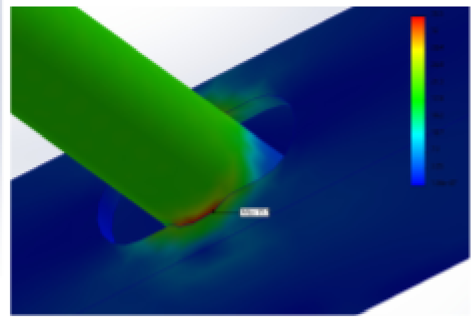


Figure [19]: Critical Stress Element. 35.5ksi

In this study, we see that the minimum factor of safety comes out to be 1.44, which is below the required 1.5. In order to check this result, the stress developed in the shaft itself was compared with a hand calculation. Assuming a 15lbf torsion applied directly to the shaft, we find:

$$\tau_{max} = \frac{Tr}{J} = 11.577\text{ksi}$$

For a 15lbf force, 4.625in moment arm, and .15625in shaft radius. Comparing this to FEA's result of 19.6, we see that FEA is significantly overestimating the stresses. Adjusting the safety factor by the difference between the hand calcs and FEA, we find $n_{adjust} = \frac{19.6}{11} * 1.44 = 2.56$ it is far more acceptable.

The FEA does accurately predict the critical stress element. Indeed, upon testing in real life, the handle works just fine at first, but over time starts to wear out in exactly the place indicated by FEA. The rate of decay is acceptable for a prototype, but for a full-scale product, the handle would need to be reinforced with a solid block in the middle. Although not analyzed, the arms wear out in the same way, albeit a lot slower. Thus, the analysis predicts an acceptable safety factor

Plate deformation FEA analysis

The final analysis performed determines plate deflection and highlights the need for reinforcement underneath the base plate. This study has the areas where the casters attach supported as well as the sides (to prevent numerical instability). A 50lbf force was applied in the area where the gearbox attaches to the plate to simulate the 35lbf weight of the cylinder, ~9lbf counterweights, and superstructure. While the applied force is admittedly a little arbitrary, the study results are consistent and useful.

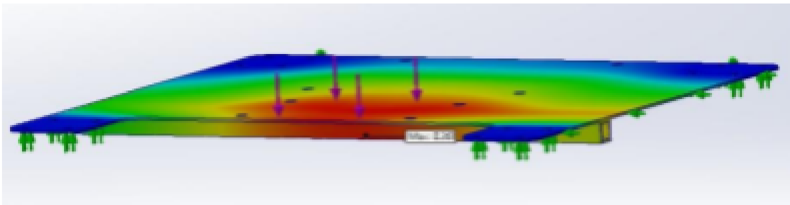


Figure [20]: Supported plate with loads and fixtures.

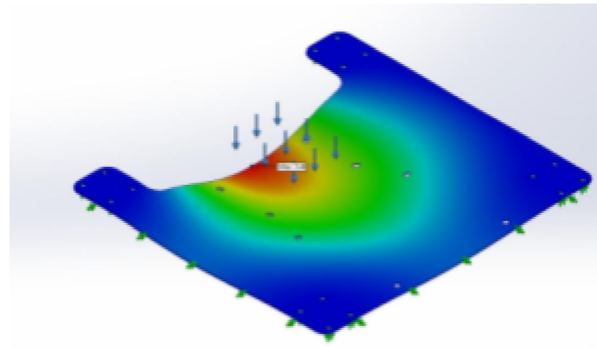


Figure [21]: Unsupported plate with loads and fixtures.

Two studies were performed with the exact same setup, the only difference being that one plate was just 16GA sheet metal, while the other was the same 16GA sheet metal with a .5” box tube welded to the bottom as reinforcement. Additionally, beam superposition equations were used to manually evaluate for both scenarios and compare with FEA. (Equations in Appendix B)

Table [3]: Plate deflection results

Plate Deflection	No Support	With Support
Analytical (Hand Calc)	10.13mm	0.251mm
FEA (SolidWorks)	3.48mm	0.261mm

Shown in Table 3 are the results of FEA compared with the hand calculations. For the non-supported case, SolidWorks warned that it was a “Large Displacement Study” whose results are not to be fully trusted. Additionally, the beam superposition equations are not the most reliable when applied to sheet metal. These two factors explain the large difference between the analytical and numerical results for the non-supported case; however, both deflection magnitudes are consenting, and the conclusion is the same: support is needed.

For the support study, SolidWorks does not complain, and the equations are valid, and they both yield extremely similar and plausible results. Additionally, 1/4mm is a very satisfactory amount of deflection and is consistent with the lack of deflection observed in real life. Overall, this study shows that the design needs support to prevent the plate from deflecting, and the .5” boxtube we used is sufficient.

Bill of Materials

Table [4]: Bill of Materials

ID	Vendor	Item Description	Link	Quantity	Cost/Unit ($)	Subtotal	Business Purpose
1	Amazon	8x33x8mm Steel Worm Gear Set 20T 1.5 Module 20:1 Reduction Rate	8x33x8 Worm Gear	1	17.99	17.99	Worm gear that will be used to provide mechanical advantage and will drive the shaft that lifts the cylinder.
2	EIS	1.5” 360° Swivel Caster Wheels	1.5" 360° swivel caster wheels	4	0.94	3.76	Caster wheels will be mounted to the base plate to maneuver the solution.
3	EIS	16ga sheet metal		1.301	1.75	2.28	Sheet metal will be used as the base plate.
4	EIS	PLA 3D Printing Filament		0.27	21.99	5.94	The 3D print filament used to create a shaft support and cylinder rest
5	Amazon	8mm Bore 5/16" Double Split Aluminum Clamping Collar Shaft Collars	8mm Bore Shaft Collars	1	8.99	8.99	Shaft collars will hold the shafts in place.
6	EIS	1" x .083" square tubing A513 steel		1.67	1.20	2	The square tubing will be used to create arms
7	EIS	1/4"-20 x 1" bolt		16	0.17	2.72	Bolts attach the 3D print and shaft support box to the base plate.
8	EIS	1/4"-20 nut		16	0.10	1.60	Nuts assist in securing the bolts to the plate.
9	EIS	5/16" solid round A36 HR steel		2.6	0.32	0.83	Solid round creates the shaft and part of the handle.

10	EIS	1/2" x .065" square tubing A513 steel		6.96	0.46	3.20	Square tubing is used to create the handle and gearbox.
11	EIS	608-ZZ Ball Bearings	608-ZZ ball bearings	6	0.17	1.02	Holds the shaft in place and allows it to rotate.
12	EIS (Scrap Bin)	High Density Polyethylene	10 Foot Polyethylene Rod	0.0042	49.99	0.21	The high density polyethylene is used to create the bushing which prevents the gear from rubbing against the metal.
13	EIS (Scrap Bin)	¼" Steel Plate	¼" A36 Steel Plate 1 Square Foot	0.13	29.9	3.89	The metal plate is used to stabilize the shafts and are where the bearings are housed.
14	EIS (Scrap Bin)	¾" Solid Round	¾" A36 Solid Round 21 Inch Rod	0.08	21.41	1.80	The solid round is used on the end of the arms to grip the cylinder.

Bill of Materials Total: $56.23

The main components of this solution include the gearbox, cylinder rest, base plate, and arms. The most expensive component is the gear box which uses bearings, ½" square tubing, the worm gears, and the shaft. Since the shaft collars and worm gear are ordered parts, it is important to consider shipping time and costs. This project is within the budget specification since it is under $65.

Design Testing:

To test the solution in accordance with the specifications and test plan outlined in Appendix C, the scaled device was tested on a scaled course. The course was constructed using tape to define a 16-inch-wide path and included two 90-degree turns to assess maneuverability. The test cylinder initially rested on an elevated pallet and was required to be transferred to a lower pallet using less than 15 pounds of applied force.

During testing, the solution was measured to verify compliance with Specification Q, requiring it to fit with a 14”x14”x14” storage container. The solution met this requirement. After it was measured, the cylinder was loaded from the upper pallet onto the solution without scratching or damaging it. As it was maneuvered through the course, no part of the solution went over the line. The cylinder was successfully loaded into the lower pallet without damage, and the solution was returned to the start line. The operator took two minutes and twenty seconds to complete the test, which is significantly faster than the maximum four-minute specification. Due to the fish scale being broken, the amount of force used to operate the device was not tested.

Design analysis indicated that the solution had a factor of safety greater than 1, meaning it would not fail. This conclusion was upheld during testing as the design successfully passed the test with no failures being observed. Although the operating force was not measured, analysis showed that it was under 15 pounds with the gears greased. Operating the solution did not seem to require 15 pounds of force. Design analysis also showed that the base plate would not deflect under loading, which remained true during testing. Through the test run, results did not deviate from the calculations.

Engineers' Assessment:

This design should advance to the next phase of prototyping for several key reasons. First, the core mechanism - a worm-gear drivetrain - which provides significant torque multiplication. This characteristic allows for a notable reduction in operator input force, enabling a lighter, more efficient system between the load point and the user’s point of interaction. In our case, the custom gearbox developed for the final prototype further distinguishes this design from typical solutions.

The design also aligns well with a few established benchmarks. For example, the use of caster wheels ensures maneuverability on the shop floor, the ice-hook concept inspired the pinch-loading mechanism on the arms, and the incorporation of a screw-jack principle inspired the use of the worm gear. After testing, the results validated the system’s performance: the prototype completed the full course with zero failures and met the required time constraints.

The design analysis indicated that the design meets all specifications outlined by the client. These calculations remained accurate during testing, showing that the design meets all specifications.

One identified shortcoming involved the stripping of the loading arm shaft attachment slots during pre-testing. This issue arose from excessive use and insufficient wall thickness in the tubing, which limited the contact area with the load-bearing bar and caused uneven stress distribution. Using thicker-wall tubing would greatly improve durability and prevent similar failures.

Looking ahead, several refinements would further enhance functionality and reliability. Friction in the worm-gear assembly can be drastically reduced by transitioning to a larger 40-tooth gear to decrease tooth loading. The grease used to reduce friction between the gears could be changed to Compounded Mineral Oils (with fatty agents for metal wetting) or Synthetics (PAOs & PAGs for efficiency/temp range). Additionally, increasing the wall thickness of the loading arms or adding a solid reinforcement in the tubes would significantly slow slot degradation and eliminate the risk of stripping.

Overall, with these targeted improvements, the design demonstrates strong promise and is well-positioned for continued development in the next phase.

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[20] "ASTM A513 Material Properties | Steel Properties, Yield Strength, Steel Specifications," *Beam Dimensions | Section Properties and Dimensions*, 2015. https://www.beamdimensions.com/materials/Steel/ASTM/ASTM_A513/ (accessed Dec. 08, 2025).

Appendices:

Appendix A: Operator Input Analysis Details

Known Values:

See Figures 14, 15, and 16 for a visual of these constants.

Lengths: $L_1 \equiv 5.78\ in$ & $L_2 \equiv 4.62\ in$

Let the radii of the gears be:

$L_3 = 0.671in$, $L_4 = 0.494in$

Gear Ratio: $n = 20$

From a research paper, the coefficient of kinetic friction of HDPE on steel is: $\mu \approx 0.08$

Here, $\mu_{st-kinetic,\, greased} = .09$ for a greased gear &, $\mu_{st-kinetic,\, dry} = .42$ for a dry gear. See the engineering toolbox [18]

Derivation:

First, we will consider the frictionless case:

Torque at the spur gear: $\tau_1 = Fcos(\theta) \bullet L_1$

Torque at the handle: $\tau_2 = \frac{1}{n}\tau_1$

Force applied by operator: $\frac{\tau_2}{L_2} = \frac{Fcos(\theta)L_1}{nL_2}$

The maximum force is seen at $\theta = 0$, evaluating the above expression, we find our baseline value:

$$F_{app,\, base} = 2.19\ lbf$$

Now we will add in the friction between the worm and spur gears.

Friction is related to the normal force between the gear teeth: $F_{fr} \equiv \mu N$.

This is where a somewhat critical assumption is made, we are modeling friction between these gears as an additional torque on the worm gear. This torque is calculated by assuming there is no pitch angle and calculating the normal force between the gears.

$$N = \frac{\tau_1}{L_3} = Fcos(\theta)\frac{L_1}{L_3} \Rightarrow F_{fr} = F\mu cos(\theta)\frac{L_1}{L_3}$$

This friction force acts in opposition to any motion, increasing the torques needed:

Augmenting our equation for τ_2: $\tau_{fr} = F_{fr}L_4 \Rightarrow \tau_2 = \frac{1}{n}\tau_1 + \tau_{fr}$

$$\tau_2 = \frac{1}{n}(Fcos(\theta)L_1) + F\mu cos(\theta)\frac{L_1 L_4}{L_3}$$

$$= Fcos(\theta)L_1(\frac{1}{n} + \frac{NL_4}{L_3}) = Fcos(\theta)L_1(\frac{L_3 + \mu n L_4}{nL_3})$$

$$F_{app,\, gearonly} = \frac{\tau_2}{L_2} = Fcos(\theta)\frac{L_1}{L_2}(\frac{L_3 + \mu n L_4}{nL_3})$$

Finally considering the friction between the HDPE and worm gear

The friction force is related to the normal force, and the torque created is related to the radius. Thus, we must integrate the friction force over the cross-section to find the total torque added by the bushing friction.

Let the radii be:

$$r_1 \equiv 0.156 \;\&\; r_2 \equiv 0.25$$

Integrating friction from ID to OD: $\tau_{bushing,\, fr} = \int_{r1}^{r2} (\mu N r) dr = \mu N \frac{r^2}{2}\Big|_{r1}^{r2} = \mu F cos(\theta) \frac{L_1}{L_3} (\frac{r_2^2}{2} - \frac{r_1^2}{2})$

Augmenting our prior equation for torque due to friction:

$$\tau_{fr} = F\mu_{st-st} cos(\theta) \frac{L_1 L_4}{L_3} + \mu_{HDPE-st} F cos(\theta) \frac{L_1}{L_3} (\frac{r_2^2}{2} - \frac{r_1^2}{2})$$

$$= F cos(\theta) \frac{L_1}{L_3} (\mu_{st-st} L_4 + \mu_{HDPE-st} (\frac{r_2^2}{2} - \frac{r_1^2}{2}))$$

$$\tau_2 = \frac{1}{n} (F cos(\theta) L_1) + F cos(\theta) \frac{L_1}{L_3} (\mu_{st-st} L_4 + \mu_{HDPE-st} (\frac{r_2^2}{2} - \frac{r_1^2}{2}))$$

$$F_{app} = \frac{\tau_2}{L_2} = F cos(\theta) \frac{L_1}{L_2} (\frac{1}{n} + \frac{\mu_{st-st} L_4 + \mu_{HDPE-st} (\frac{r_2^2}{2} - \frac{r_1^2}{2})}{L_3})$$

Friction Coefficients

Friction coefficients for steel on itself and various materials are not hard to come by. We elected to make use of the EngineeringToolbox.com because of its broad scope. In this analysis, because we are concerned with moving parts, we use kinetic friction coefficients. There are different coefficients listed for different surface conditions. The dry coefficient is easy; it's simply listed directly, giving 0.42. The lubricant we used was silicon grease, but the entry for grease only has a static coefficient of 0.16, which is significantly less than the dry number. (As would be expected) Kinetic coefficients are provided for Castor oil (0.084), Lard (.084), graphite (0.058), and Stearic Acid (0.15). Thu,s it was decided to assume our kinetic friction coefficient to be 0.09 for silicone grease. This is less than the static coefficient and comparable but greater than the coefficients for castor oil and lard.

As for HDPE on steel, it was not on the engineering toolbox or other typical websites; however, a research paper was found. It is a study about how friction changes as you slide 304L steel on HDPE for many cycles. In the paper's [19] 4th figure, we find what we need. As we are starting with a fresh HDPE, we chose the lower end. Additionally, the surface between the steel and HDPE is lubricated, further justifying our choice of 0.08 for the HDPE-Steel coefficient.

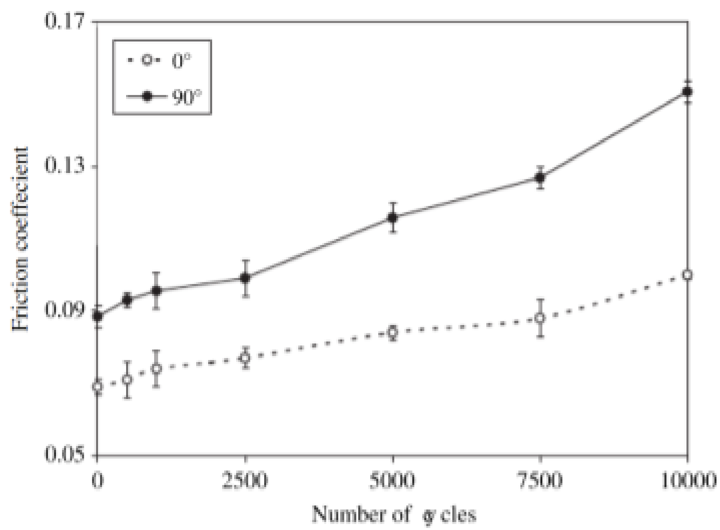

Figure [22] from source 19, Friction coefficient versus number of cycles for the 0° direction and the 90° direction; fresh HDPE surface, Fn=500 N, 1 Hz.

For results of this analysis, see Table 2.

Appendix B: Bending analysis hand calculations

Here are the equations used in performing the beam superposition calculations that support the beam deflection FEA study.

$$I_t = \frac{b^4 - b_{\text{in}}^4}{12} \qquad I_p = \frac{b_p t_p^3}{12} \qquad \bar{y} = \frac{A_t y_t + A_p y_p}{A_t + A_p} \qquad y_{max} = \frac{-F\ell}{48EI}$$

Appendix C: Testing Plan

Test 1

Objective	This test will aim to find the force required to maneuver, control or operate the design. - Spec A: No more than 15 lbs of force is required to maneuver, control, or operate the solution
Assumptions	- The fish scale used will be accurately reading the force exerted on the gear and cart. - The force on the gear will always be normal to the position of the handle. - The friction between the floor and the wheels is negligible.

Hazards & PPE	Hazards - 35lb steel cylinder falling - Pinch point in gear - The cylinder causes the design to tip over - Loaded casters rolling on hands or feet PPE - Safety Glasses
Tools required	- Fish Scale
Test Setup, Required Environment	- The cylinder will begin on the lower cradle because when lifting from that position, max force is encountered when the arms are horizontal. - The solution will be aligned with the cylinder on the loading pallet. The design will not operate as intended unless the arms are aligned properly with the cylinder. - There must be space in front of the design so it can be rolled with the cylinder attached. This is needed to test the amount of force it will take to roll the design.
Procedure (Execution and Data Collection)	- Remove pins from grabber arms. - Slide the arms apart and align them with the cylinder. - Push the arms together and insert the pins. - Rotate the lever arm with a fish scale attached. - Record the force reading of the fish scale. - Attach the fish scale to a handle and observe the required pulling force to cause the solution to roll.
Data Analysis & Conclusion	The fish scale should produce a measurable amount of force that can be compared to the limit of 15lbs. If the fish scale shows a force that is greater than this limit, the test will fail. If it shows less than 15lbs, the test was passed successfully. This applies for both applications of the fish scale, it is all or nothing. The test can only pass if the fish scale does not exceed 15lbs when lifting the cylinder and when maneuvering it.

Test 2

Objective	Gather objective evidence that one operator can load, transport through a 16-inch corridor, unload to the machine receptacle, then clear the corridor, without pallet or product damage and while meeting safety/handling requirements. - Spec E - L
Assumptions	- The friction between the floor and the wheels is negligible.

	- Testing course and cylinder matches the course and cylinder the design will be evaluated on - All track widths are 16in
Hazards & PPE	Hazards - Wheels of the carrier rolling over someone's foot - Pinch points between cradles, cylinder, and design PPE - Safety Glasses
Tools Required	- Timer to ensure a time of less than 4 minutes
Test Setup, required Environment	To run this test, the factory course, both loading cradles, and the 35lb cylinder are required. - Course is required to evaluate the width constraints - Cradles required to evaluate tipping, damage, loading, and unloading specs - Cylinder is required to evaluate damage, loading, unloading, and transportation specs
Procedure (Execution and Data Collection)	Factory track and cradles, with the taller one loaded with the cylinder, will be laid out with design nearby. One single operator will load, navigate, and unload under the allotted time without breaking any constraints. - Stopwatch will be used to track duration of operation - Damage, tipping, loading, and unloading spec data will be collected by observation
Data Analysis & Conclusion	- After the device has loaded, transported, unloaded, and cleared of the area, the time will stop and be recorded. If the operator can clear the area in under 4 minutes, it will pass spec G. - If no damage or tipping was observed during operation, the device passes specs E and L. - If a single operator, operating the design, successfully transports the cylinder from one cradle to another without any “collisions” with the factory walls or the floor, the design will pass specs F, H, I, J, and K.

Appendix D: Gearbox Precision Machining Methods

As explained in the design description, it is critical that the gearbox be machined precisely, maintaining alignment between shafts and keeping bearing holes aligned. To achieve this, quite a lot of time was spent in the machine shop creating it. The main frame was cut to size on the bandsaw and welded together. We created some custom tooling to keep things the right distance apart and securely bolted it all together before welding.

Despite these efforts, as anticipated, it was still too imprecise to be used. To achieve high precision, the bearing holes in the plates that were welded onto the gearbox were not bored prior to welding.

The unfinished gearbox was placed on the mill and carefully aligned on a bolt plate to get the top surface as close to flat as possible. Then it was decked with a shell mill around .025” to get it all on the same plane. Then the DRO was used to cut out the bearing hole, located off the Adjuster plates, which were already bolted on. In order to maintain alignment between the two bearing holes, a special bit of tooling was made. A cylinder of precisely the same OD and ID of the bearing hole was made on the lathe and bolted down onto the bearing plate. First, the center of the tooling was located with a dial indicator, and then the gearbox was flipped, and the previously milled face was laid on the bolt plate, the gearbox being slid onto

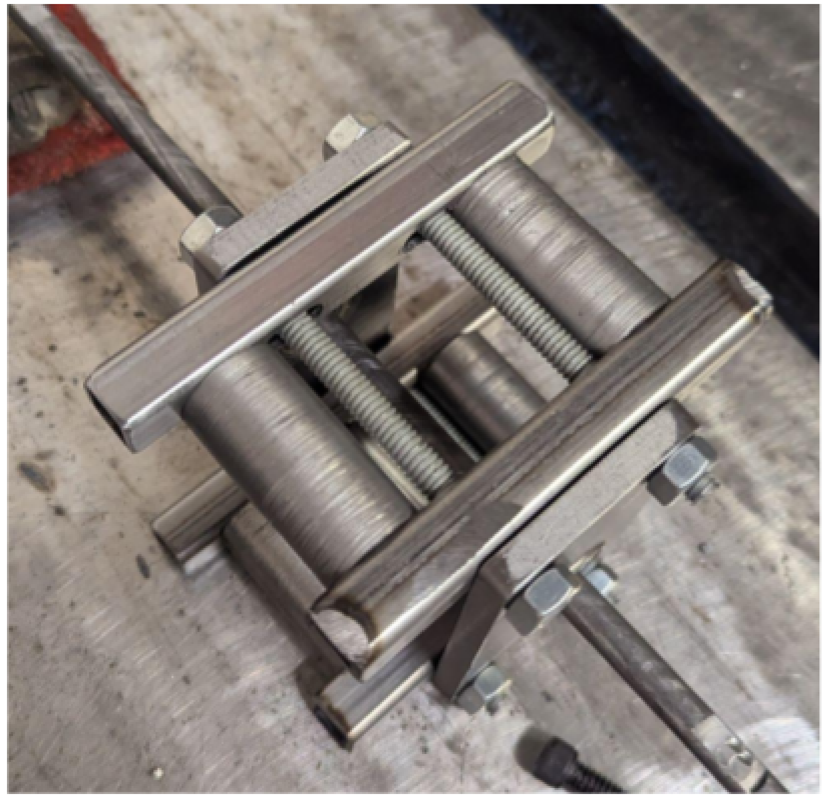

Figure [23]: The gearbox frame bolted into its frame prior to welding

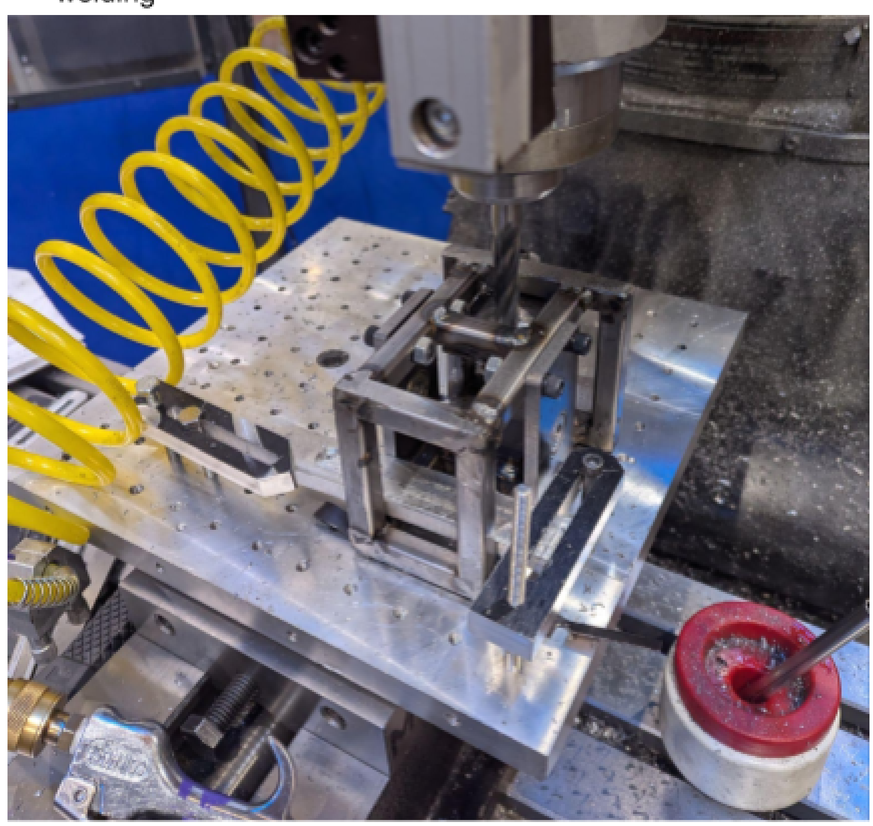

Figure [24]: Milling the first bearing hole the tooling. This locates the part such that the second bearing hole is exactly aligned and parallel to the first one. Additionally, when the first bearing hole was located off the adjuster plates, it was ensured that it was in the correct orientation relative to the spur gear.

Thus, while the welded frame was quite inaccurate, by using the mill and some tooling, the bearing holes are precisely aligned with each other, ensuring proper gear alignment.