Wheelchair vs. Staircase

Stephanie Cheng, Mario Sebasco, Savannah Skoran, Michael Vander Meiden
University of Miami
Department of Mechanical and Aerospace Engineering

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1 Abstract

The 2010 U.S census estimated that there were 3.6 million people in wheelchairs in the United States, 49% of which were between the ages of 15 and 64. Although many advances have been made to grant wheelchair users more freedom and independence than ever before, there are still some challenges to be addressed in the arena of disabled access. Using a staircase, for example, is often taken for granted but can be a source of difficulty and threatening danger for persons in wheelchairs. The objective of this project is to design a system that would allow a disabled person to traverse a set of stairs without aid from another individual. After completing preliminary research on all the ideas presently available, it was decided that the mechanism to be used would be a mechanical wheelchair lift. Picked due to its structural integrity, relative inexpensiveness, and straightforward design, the final system can be viewed from figure 37. A prototype was built and tested to yield deflection and ascension rate results. Platform deflection results exceeded standard requirements and Solidworks assembly models were used to investigate the unexpected behavior. It is hypothesized that imprecise/amateur machining led to disparities in Solidworks models and prototype assembly. Replacement of some Aluminum components with steel are proposed for future work. Although Ascension rate is satisfactory and performs according to ISO standards, additional disengaging and descent components require further testing and implementation. Considerations taken into account include: Structural analysis of components subjected to high stresses, biomechanic factors to offer optimum input from the user, and force input variations for added versatility. Concluding, the paper serves as an interim report on the manual lift endeavor with proposals for future work and research.

2 Objectives

The objective of the design project is to develop a mechanical system that would allow a disabled person to traverse a set of stairs without aid from another individual. It is important to define the term "disabled person(s)" in the context of this endeavor as referring to those individuals who are confined to wheelchairs, but exhibit full functionality of all appendages above the waist. An ideal user of the system could include young veterans of war who have been injured in active duty or otherwise healthy and active individuals who happen to require the use of a wheelchair for one reason or another.

Although there are currently several alternative methods of vertical transport for disabled persons - including ramps, elevators, stair-climbing wheelchairs (SCW), and inclined wall-mounted lifts - many of these require the aid of another individual or can be very costly due to heavy-duty machinery and electrical components. Ramps seem to be an exception but oftentimes there are space restrictions that prevent their installation. Ramps can also be difficult to integrate into existing buildings and structural systems due to the amount of ground coverage required to achieve common vertical displacements that must be navigated.To achieve the primary goal, the design will focus on cost-effectiveness, space-efficiency, reliability, and safety.

3 Background

When investigating existing solutions to the design problem, two main design approaches were considered: the Stair-Climbing Wheelchair (SCW) and a Wheelchair Lift system. Both exist as electrically or manually driven devices used to allow disabled individuals to climb staircases. The most significant difference between the two options is portability; with SCWs, the stair-climbing system is integrated with the wheelchair and is therefore portable and constantly available to the user at any point in time. A lift system would be permanently installed on a specific staircase for use only at that location. Both types would fulfill the design objective and require mechanical analyses in design. Prior work in both fields have been presented in the following subsections.

There are also several important design considerations, which should be taken into account for either type of system and are addressed under the Design Considerations section subsequent to the overview of existing work.

3.1 Literature Review

3.1.1 Electrical SCWs

Overall, electric wheelchairs come with a distinct set of advantages and disadvantages. For one, electric models save the wheelchair occupant physical exertion and is a more comfortable experience overall. Second, because the power is usually stored in a battery pack underneath the seat, there is a lower center of gravity which makes stability in design easier. Unfortunately, this battery pack also makes the wheelchair heavier and bulkier, so it may be harder to transport from place to place.

Electric wheel chairs also require the chair to be charged and this charge lasts for a limited amount of time, limiting some of the wheelchairs applications. Finally, electric wheelchairs initial cost and cost of maintenance are often much higher than manual wheelchairs which may be prohibitive for some consumers. Below are a few examples of existing electrical SCW concepts.

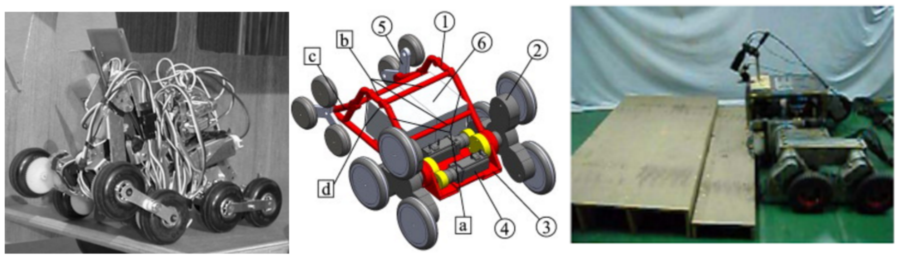

Figure 1: Left to Right: Chun-Ta,Giuseppe Quaglia, M.J. Lawn

One paper, published in The Industrial Robot by Chun-Ta, outlines a new design and prototype for an electric stair climbing wheelchair, with emphasis on stability analysis. The prototype is only a partial prototype of a proposed base for an electric chair. This prototype has 4 wheels with each wheel connected to a triangular support. While the chair is moving horizontally, the triangular supports are tucked in and the the chair moves as common wheelchairs would. When the wheelchair comes up on a set of stairs, the triangle supports rotate and press against the base of the stairs for stability. The back wheel then rotates around the front wheel onto the next step. This prototype was made to traverse steps of height 20cm and a depth of 26cm [19].

While Giuseppe Quaglia did not actually build a prototype he created designs and did an engineering analysis. This design consists of 6 rear wheels (three on each side, triangle formation) 6 smaller front wheels in the same formation. [20]

The final electric design was published in 2003 by M.J. Lawn and has the primary purpose to navigate across a single high step, such as the entrance into a van. This design varies widely from other methods and uses a series of linear actuators to extend and push off of the rear wheels. The design climbs stairs using a rotating set of two wheels on each corner. [18]

3.1.2 Manual SCWs

In terms of manual SCW designs, one of the earliest documented attempts was a patent filed in 1963 (as presented in Figure 2). The objective of the specified design is to construct a manual wheelchair capable of traversing stairs and other obstacles in a safe and comfortable manner. The patented wheelchair must, according to the description, be capable of accounting for differing step sizes. Furthermore, it must be able to be folded, be lightweight, easily transported, and economical.

Overall it is unclear if the system is efficient, or if it even functions correctly. The designer does not provide any clear dynamic measurements to support his claims, and aside from figure 5 which does not provide a great visual representation of the overall motion of the system not much else is shown. For example, it is unclear from the figures (and the text), if the smaller front wheel will

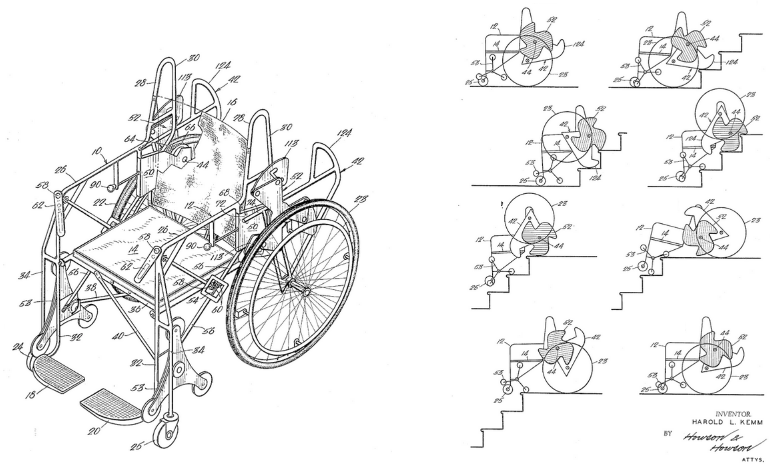

Figure 2: Patented wheelchair design

come into contact with the staircase, which may result in overall failure. How the device descends stairs, and how the person in the wheelchair powers the system raises even more questions which are not explained in the patent.

Conclusively, although the patent contains novel ideas, prototype testing is needed in order to gauge the full effectiveness of the chair. From first inspection, more variables need to be taken into account, such as slipping, and whether any of the wheelchair components will interfere with the stair climbing system. Furthermore, the structure requires two axes of rotation (one when on a flat surface, and one when traversing stairs), which can lead to excessive complications in the design that are just not necessary; it is also impractical to have a large 22-inch wheel spinning as shown in figure 5, if the operator is in a small stairway. Effectively, numerous improvements need to be made in order to make the system a viable solution to the objectives given. [15]

Figure 3: One Step wheelchair design

Another design was proposed at the University of New York in Stony Brook in 2012. The One-Step Climbing Wheelchair was designed to be capable of climbing only one stair, specifically a sidewalk curb (see Figure 3). Having the ability to climb a curb would save time and effort that would otherwise be spent locating a sidewalk ramp for wheelchair access. The main goal the designers of this wheelchair had, was to make it as cost efficient as possible. Compared to electric types, manual wheelchairs cost less because they require smaller amounts of material and time to manufacture. With this in mind, they used a common manual wheelchair as the base and built a permanent add on device that modified the existing wheelchair to be able to climb a curb. The one-step-climbing wheelchair consists of a front and a rear gear subsystem and a linkage subsystem. The front gear subsystem takes input power and transfers it to the rear gear subsystem by bicycle chains. A bicycle hub is used to transfer the input power to the wheels. The rear gear system transfers the power to the linkage system with a designed gear ratio. The linkage system plays a key role to overcome a curb. It has a slot path to relocate the wheel shaft and also a hexagonal shaped shaft locker at the end of the slot path. It mainly reduces the force input by converting the users weight to a torque on the wheel shaft. It also allows the user to put a portion of his/her weight over a curb prior to climbing and then lock the wheel while climbing.[12] The reported total cost for parts to build the One-Step-Climbing Wheelchair was $1500, which can be too costly for many average individuals to afford. Other disadvantage of this design is that it is very limited in its applications. It was specifically designed to be able to climb one step and only one step. It is incapable of traversing a series of steps or even a set of two steps one after the other. Another disadvantage is that the two levers located on the sides of the wheelchair constantly stick out farther than the wheelchair and may cause injury to other people in close proximity.

Figure 4: Vardaan wheelchair design

In 2012, a Gandhian Young Technological Innovation Award was given to Shanu Sharma, a student of the Indian Institute of Technology Kanpur, who designed the Vardaan wheelchair (Figure 4) for the purpose of traversing any number of stairs. Unlike the One-Step-Climbing Wheelchair, the Vardaan is capable of climbing whole staircases, not just one step. The most different feature of this design is that it is able to travel both up and down stairs. All other designs have focused mainly on traveling up, so the Vardaan is truly unique in this aspect.

Vardaan facilitates climbing up and down the stairs by using an innovative Y-shaped wheel that provides better grip and optimum braking along with a ratchet and a braking system. The Y-shaped wheel has been designed to mimic the human gait while climbing up. For climbing up the user pulls and pushes the lever ratchet attached on the axle of the Y-shaped wheel. The design makes use of the fact that people with lower limb disability have high upper body strength and therefore pulling/pushing the ratchet lever can be conveniently operated by them. While during the climbing down the user can use brakes to control the motion of the wheelchair as the Y-shaped wheel rolls down the steps. [13] [14]

The Vardaan is able to carry people weighing between 58 kg and 89 kg. It is also able to traverse all stairs that meet the National Building Code Specifications of step widths 230-350 mm and step heights 110-185 mm.

A disadvantage is the amount of strain placed on occupant operating the wheelchair. This design is based on the assumption that people with lower limb disabilities have strong upper bodies to make up for it. The design is also only suitable for traversing up and down stairs. The Y-shaped wheel, though adept at grabbing onto stairs, would make regular travel on a flat surface difficult. The wheelchair would stop every time the Y-shaped wheel turned to have the spacing between the legs of the Y facing the ground. The amount of effort to turn the wheel, even with the benefit of the lever ratchet system, would be far too much just to go a distance of a few feet. Finally, the unique design of the wheelchair does not use a common manual wheelchair as its base. Each part would have to be specially made just for this design. So, even if the materials were cheap, the cost to manufacture the wheelchair would be high.

3.1.3 Electrical Lifts

Electric stair lifts are situated on a track secured to either the wall adjoining the stairs, the stairs, or both depending on the design, manufacturer, and environment. Residential stair lifts usually run along a single track attached to the stairs with a small chair for the occupant to sit on. These lifts are small in size for increased spacial efficiency. Seat belts are included to prevent users from falling off and injuring themselves during transit. Lift speed is kept at a general walking speed of 5 mph as another safety precaution. Compared to residential areas, stair lifts located in urban areas are larger and more industrial in design. They must follow slightly different safety standards and need higher durability. This is because the amount of usage is significantly greater. The United States has an average occupancy of four people per residential home. So the amount of people using a residential lift is much smaller than the amount using a lift thats available for public access. This affects pricing as well. Residential lifts are cheaper because amount and strength of material used in manufacturing them is less than urban lifts. Prices range from $100 to over $7000 for residential stair lifts alone.[21] Most electric lifts use a battery to supply them with power. The batteries of residential lifts have an average lifetime of 2 years and regular maintenance is recommended. [22] Batteries cost around the same price as car batteries. The batteries must be replaced by professionals which is not a free service provided with purchase of an electric stair lift. Electric lifts arent widely available to everyone due to high cost.

3.1.4 Manual Lifts

Upon consideration of a lift type design, it was determined that a manual system would better suit the objectives of the current design project. Electrical lift systems would be more costly and would require additional considerations for protection of electrical components in outdoor weather conditions. After much research, only one published project was found that attempted to design and build a manual stair lift for wheelchair-confined persons. This project was carried out by a group of students at the University of Wisconsin-Madison. Similarly to the current project, they realized that those needing to ascend a steep environment in a wheelchair only had two options, powered lifts and ramps, and that powered lifts were often prohibitively expensive. Their design was submitted to the Rehabilitation Engineering and Assistive Technology Society of North American 2013 Student Design Competitions under the category of Wheeled Mobility Technologies. They dubbed their design the Funicular.

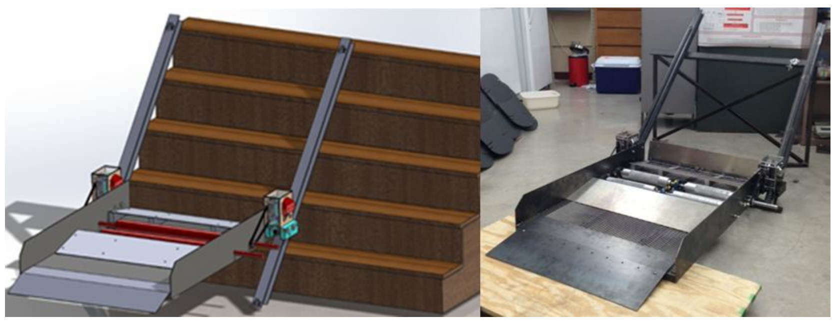

Figure 5: University of Wisconsin lift design

The Funicular design (Figure 5) is powered using two roller pins in the center of the platform. The wheelchair is navigated backwards up the small ramp onto the platform until the rear wheel is resting on the two roller pins. The user then pushes the wheels as he normally would which spins the two roller pins, operating a gear system, and pulling the users up a chain hoist.

The students tried to design the Funicular so that it could be operated with 5lbs of input force which is the industry standard. The design they created was under 500 dollars and had safety gates that also served as on and off ramps at the front and back of their platform. Overall their design was able to successfully ascend the stairs (although used slightly more than 6.5lbs of force).

3.2 Biomechanics/Anthropometric Factors

In prior work in manual systems the effort required of the user has been overwhelming and impractical. The human factor in design can often be overlooked, but is an essential component to the practicality and marketability of any design. In order to traverse a set of stairs, work must be done over a period of time resulting in a total power input requirement to complete the task. In the case of a manual stair-climbing wheelchair, the source of power/energy is the human body and related factors must be taken into consideration in the design.

In a recent review of biomechanics and human powered vehicles [3], it was described that there exists a force, velocity, power relationship of human muscle capacity that should be considered. With a high velocity of contraction (and no load), minimum muscle force (and power) can be produced. As the load increases, the velocity of contraction decreases, and with a maximum load,

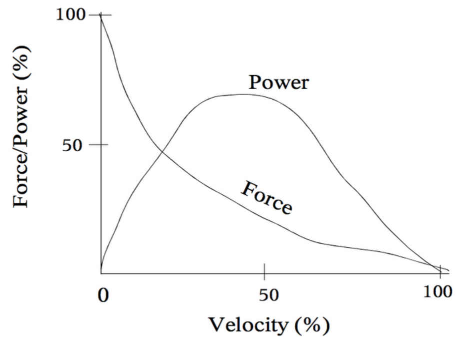

Figure 6: Power/Force versus velocity

the force of contraction becomes a maximal isometric one (resulting in zero power). The relationship can be observed in Figure 6.

To achieve efficient locomotion, human power output can be seen to maximize at between 33-66% of maximum force generation and contraction velocity. In addition, there is the effect of muscle fiber length in the generation of power in reference to cycling motion, which introduces the need to account for user-machine interface and orientation. It is evident from anthropometric studies [4] that the orientation of the body in relation to a mechanical device is essential in determining the force and power output of the individual. Since there exists a wide range of human geometries and forms, considerations must be made in terms of the target population for any assistive device. The manual stair-climbing wheelchair design is one better suited for use by fit and independent individuals as opposed to senior citizens and individuals whose strength has been compromised as result of disease or other physical ailment.

Although literature on the subject is incomplete, the ISO 7176-28:2012 provides standards on the maximum direct operating forces for manual stair-climbing wheelchairs and the combined standards for residential, manual lifts by the ASME A17.1.2004 provides good basis for design of lift type systems. Understanding that the average adult weighs more than the allowable operating force, there exists a need for mechanical design that magnifies the force applied in order to produce the necessary lift for ascension.

Multiple types of simple machines can be used to gain a mechanical advantage and thereby magnify the applied force, including gears, pulleys, lever arms, and hydraulic components. However, by a simple analysis of force, work, and energy relations, it can be shown that the less force required to operate a machine, the longer it takes for the machine to complete the tasks. There must exist a compromise between energy expended to traverse the stairs and the time it takes to do so.

Also described in the standards are a series of design types that have been recognized/established by the International Standards Organization (ISO), American Society of Mechanical Engineers (ASME), and Americans With Disabilities Acts (ADA). SCW types can be observed in figure 7 and a multitude of lift designs categorized by driving mechanism, orientation, support system, rail type, and more can be found in the corresponding standards.

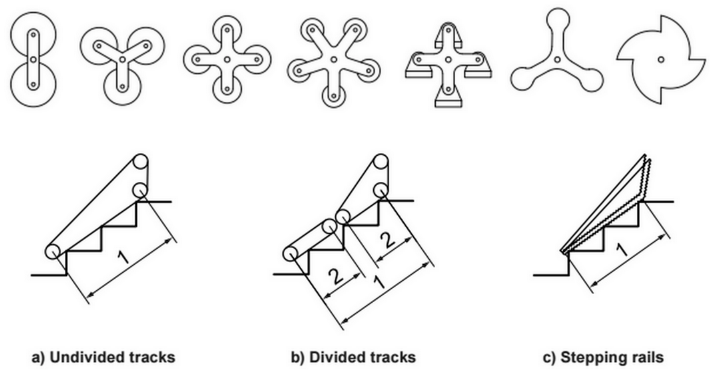

Figure 7: Wheel and tread types

3.3 Design Constraints

Economic:

The main economic consideration in the project will be cost of materials/production. Although maintenance and operational cost may be relevant, minimizing these costs will not be a focus of the endeavor. As part of the main objective, the goal is to develop a system that is less costly than currently available systems, while still maintaining high standards for safety and quality. If the final design system is to be of the manual lift type, the system should cost less or equal to the previously discussed funicular design from the students of University of Wisconsin-Madison. Costs will be reduced by: (1) optimization of the system to eliminate unnecessary weight/components; (2) using commercially available products that can be purchased wholesale or in bulk; (3) minimize/eliminate the use of electrical components which contribute to material, operating, and maintenance costs.

Environmental:

Since the design problem includes operating conditions that are greatly varied, it is essential to take into consideration the environmental factors affecting the design. For exterior stairways, the system will have to operate safely and effectively in non-ideal weather conditions: mainly referring to rain and wet stairwell surfaces. Accounting for potential slippage or malfunction of an SCW or manual lift system in the presence of environmental factors will be an important design consideration during both the design and evaluation stages of the project. Codes and standards exist which dictate the required injury/malfunction prevention methods to account for environmental factors.

Social, Political, and Ethical Impacts:

Due to the nature of the design project and the desired approach to the problem, there are no significant social, political, ethical or sustainability-based considerations to be made which operate as design constraints. It is ideal that the appearance of the device should be benign to reduce the potential negative stigmas surrounding disabilities and disabled individuals, however this social factor should not significantly affect the design process. Ethical issues may arise regarding to coherence with existing codes and standards, but are easily eliminated by following best practice procedures and using appropriate engineering standards.

Health and Safety:

Health and safety will remain a major priority for the project and will be addressed by constantly referring and adhering to the codes and standards governing the chosen solution. Main considerations include safety of the user and bystanders other than the user during both operating and non-operating conditions. Fall prevention, pinch-point reduction/labeling, and fail-safes should be included in the design as ordained by official standards and as judged necessary by the designers.

Manufacturability:

Manufacturability of the design components is directly related to the cost of the final design and will be taken into consideration during the design process. Allowable tolerance and clearances should be specified in a way that maintains reliability and operability of the system while minimizing cost. Components should require minimal and relatively simple machining work.

Sustainability:

Both system types proposed as potential solutions to the design problem will involve inert or benign materials that are readily available and should not impose unsustainable burdens on the environment, human health, or sociopolitical and economic stability.

Codes and Standards:

ISO 7176-28:2012 [11]

- Operating force will not exceed: 120N ($\sim$27lbs) for single arm operation, 240N ($\sim$54lbs) for two arm/hand operation, and 400N ($\sim$90lbs) for combined trunk and arms operation.
- Testing staircase will be consisting of eight steps, each having a rise of 180 mm (7 in). The overall pitch shall be 35° (see Figure 8).
- Center of Gravity of the wheelchair will be placed at the following location for testing purposes when actual is not known (using surrogate wheelchair):
  - Height from floor: 0.45$\pm$.05m (17.7$\pm$2in)
  - Distance from rear wheel axle: 0.15$\pm$0.05m (6$\pm$2in)
  - Centered between wheels

ISO 7176-1:2014[16]

- Determination of Static Stability of Wheelchairs

ADA Standards (308-309)[26]

- Establishes which ASME standards and codes are relevant to the design of various lift systems.
- Force required to operate should not exceed 22.2N (5lbs)
- Height of reach: 15 - 48" from ground level (unobstructed)

ASME a17.1.2004[27]

- From Hand elevator (4.3):
  - Platform Factor of Safety (F.S.) = 4
  - Rated load uniformly distributed over platform
  - Platform supports and connections are
    * Bolted
    * Welded
    * Riveted
  - Platform minimum rated load: 50 lb/ft2 (240 kg/m2)
  - Rails: shall not deflect more than 0.25in (6mm)
  - F.s. based on static loading
- Private residence inclined elevators (5.4)
  - Platform area not exceeding 1.4 m2
  - For areas up to 1.1 m2 (12 ft2) rated load must be 195 kg/m2 (40 lb/(ft2)) : 285lbs
  - Rated speed 0.20 m/s (40 ft/min)
  - Suspension means can be one wire rope or roller-type chain conforming with ASME B29.1
  - F.S. for chains must be at least 8
  - F.S. is 5 for beams
- ASME 2.23 for guide rails geometry and strength

3.4 Additional Considerations

It is also certain that the suitability of an SCW varies greatly depending on the type of stairs involved. Standards for staircases vary by region, as well as purpose, leading to an impossibly broad range of staircases for which a universal SCW will need to traverse. The focus of the current investigation will be narrowed down to the set of test stairs as defined by the ISO 7176-28:2012 used

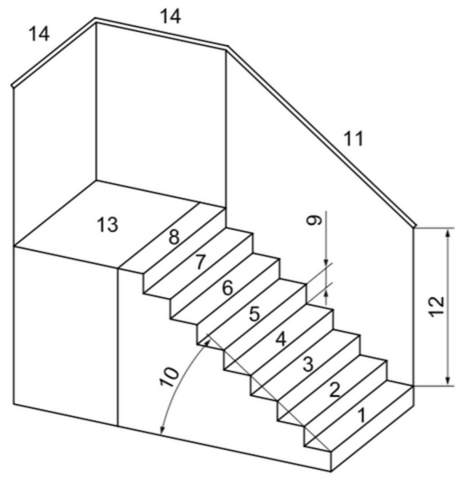

Figure 8: staircase testing standards

for testing SCWs (See Figure 8). Note that this standard differs from the building code standards established under law for the State of Florida as of 2004.[17]

Also, due to low cycles of usage in either type of system, the effects of fatigue were considered irrelevant and remain outside the scope of the current investigation.

3.5 Relevant Theory

In order to properly design a serviceable system, it is vital to correctly understand the stresses and motion of all the components. Doing so will require an in depth understanding of the physical governing equations, and their analytical (or numerical) solutions. These steps are an integral part of any mechanical design and ensure that the system will not fail, possibly resulting in severe injuries to a user. The following few sections will go in depth into the main considerations accounted for when constructing a safe and versatile apparatus.

3.5.1 Human Power Capacity

The primary consideration in designing an adaptable system, is the range of power that an average person is capable of maintaining. It is important that the design adhere to industry standards providing for use by persons who may not generally be in good physical condition, while allowing able-bodied individuals to operate the system with greater speed. Overall, the power required to traverse a set of stairs is equal to

$$P = \frac{W}{t} \tag{1}$$

where,t and W are time required and work done to ascend the staircase, respectively.

$$W = F \times d = m \times g \times h \tag{2}$$

with m being mass of the wheelchair and user, g being the acceleration due to gravity, and $d = h$ being distance over which force is applied and is equal to the total height of staircase that must be traversed. The speed of (or time required for) ascension is then directly related to the potential power output of the user. These are useful relations that are used during theoretical evaluation of a given system design. In order to carry out the design process, data on human power capacity was gathered from NASA and other anthropometry sources [4][3]. Power output and transmission concepts are essential to the design process and are further discussed in application.

3.5.2 Power Transmission

ADA Standards (308-309) require that the maximum force exerted by anyone on a wheelchair lift be 5 lbs. Similarly, ISO Standard 7176-28:2012 states that the force exerted by anyone using a wheelchair to traverse a set of stairs be limited to no more than 90lbs. As a result of these force restrictions, it is necessary to implement a system of mechanical advantage that will allow the user to complete a relatively demanding task without having to exert a substantial effort.

Common mechanical systems implemented in such cases include, but are not limited to, gear systems, pulleys, and levers. The specific mechanical advantage devices closely studied for this project are mainly limited to levers and gears, the former of which is highly applicable in this specific scenario, and the latter is compact and capable of being highly versatile.

The theory behind the mechanical advantage provided by a lever is quite simple. The torque applied by any lever is given by

$$T = r \times F \tag{3}$$

where r is the distance from the point of rotation to the force being applied, F is the force applied, and the torque is the cross product of these two quantities. The larger the lever arm, the greater the torque that can be achieved at the axle. Thus, if a small gear (or a similar mechanism) is placed concentrically with the lever, it can be capable of transmitting much larger forces (since the torque is the same, but the lever arm is much larger).

When a lever is not sufficient to obtain the desired force output, then multiple gears can be introduced. For the simple case of two gears in contact with each other figure 9, the forces acting on the contact point must be equal in each gear (given Newton's third law of motion). If this is true, then a simple relationship can be derived between the torque and the radius of the gears. That is to say, the torque of the small gear in figure 9 will be amplified in the big gear since the radius is larger. The drawback is that the small wheel will have to rotate more degrees than the larger one (thus having a higher rpm). The advantage of using a gear system is clear in this application, especially when the force reductions can be transmitted over long distances with the use of chains. Some simple relations can be made for gears such as the ones shown in the figure.

$$\frac{T_A}{T_B} = \frac{RPM_B}{RPM_A} = \frac{Teeth_A}{Teeth_B} \tag{4}$$

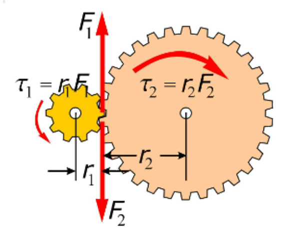

Figure 9: Forces and torques acting on two gears in contact [24]

Although, different variations of gear systems may be implemented for specific applications in order to improve cost or account for space requirements, the basic equations always hold. Other fundamental equations that may be useful when applying these mechanical systems include energy and power equations given above (Eq. 1 and 2).

3.5.3 Lever Drive Mechanism

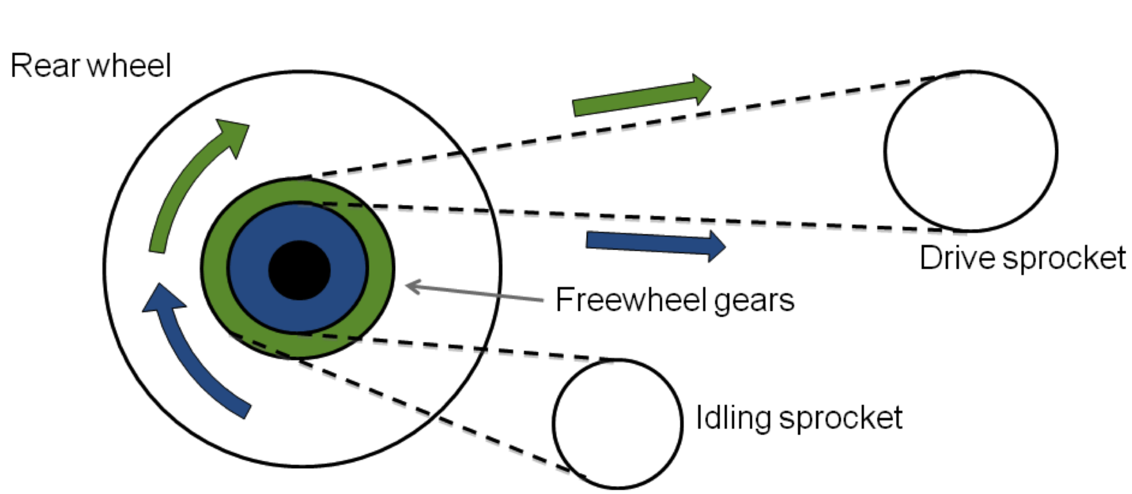

Figure 10: Illustration of lever drive components and concept [25]

The lever drive as pictured below (Figure 10) is a simple sprocket and chain subsystem that can be used on wheelchairs and other devices to improve ease of use. It is composed of a drive sprocket (whose shaft is fixed to a lever arm), idle sprocket, and 2 freewheel gear/sprockets oriented such that back and forth motion on the drive sprocket causes unidirectional rotation of the freewheel sprocket shaft. For both an SCW and a lift system, the lever drive is superior to a simple ratcheting device by allowing the user to produce work at the output from both pushing and pulling of the lever. In terms of sizing, the two freewheel sprockets may be of any size relative to each other (unlike the diagram where one is larger than the other). For the current application, a larger drive sprocket to freewheel sprocket is generally desired to obtain greater rotations at the output shaft from each input shaft rotation. Of course, a greater ratio also results in greater torque input required at the drive sprocket. From the equations governing power transmission (Eq. 1 - 4), the ideal sprocket ratio will have to be a balance between the desired output and possible input in terms of both force

(Eq: 5) and displacement (Eq. 6). Table 1 of Appendix D demonstrates the effects on output, force, and displacement based on average human power output and sprocket ratios for the case of a manual lift in which displacement is measured as vertical ascension, y (Eq. 7), and R_3 in equations 5 & 6 refer to the output sprocket radius.

$$F_{out} = \frac{LR_2}{R_1R_3}F_{in} \tag{5}$$

For equation 5, $F_{out,in}$ are output and input force, respectively; L is the length of the lever arm which drives the input sprocket; R_i is the sprocket radius, with index $i = 1, 2, 3$ referring to input, freewheeling, and output sprockets, respectively (assuming freewheel sprockets are of equal size). Note that if the output sprocket is replaced by a wheel for the case of an SCW, R_3 would simply represent the radius of the rear wheel.

$$s_{out} = \theta_3R_3 = \theta_1R_2 = \frac{s_{in}R_2}{L} \tag{6}$$

For equation 6, $s_{out,in}$ are the output and input circumferential displacements, respectively; $\theta_1 = \frac{s_{in}}{L}$ is the input angular displacement; $\theta_2 = \theta_3 = \frac{\theta_1R_2}{R_3}$ is the output angular displacement;

$$y = \sin(\theta_{incline})s_{out} \tag{7}$$

For equation 7, $\theta_{incline}$ is the incline of the stairwell and is equal to $tan(\frac{h}{d})$ where, h is the height of the stairs and d is the depth of the stairs.

Note that the sprocket radius used in calculations should be derived from the pitch diameter rather than the outside diameter. The following equations (8 - 9) relate tooth number, N_T; outside diameter, D; pitch diameter, P_D); and pitch, p.

$$P_D = \frac{p}{\sin(\frac{\pi}{N_T})} \tag{8}$$

$$D = P\Big(0.6 + \cot(\frac{\pi}{N_T})\Big) \tag{9}$$

3.5.4 FBD

In order to be able to understand where a design will fail (mechanically), it is necessary to find locations of high stresses and ensure that no material limits are breached. In order to do this however, it is first necessary to calculate the forces acting on the body. This can be accomplished with the aid of a free body diagram. If one assumes the problem to be a static one (which can be done in this case since the accelerations are very slow and any inertial forces will be insignificant), and the system to be a rigid body, then the equations of motion are simply

$$\sum F_x = \sum F_y = \sum F_z = \sum T = 0 \tag{10}$$

As an example, figure 11 shows the external forces acting on a mechanical lift, composed of the center of gravity of the lift, and the corresponding forces on the wheels (vectors not drawn to scale). It is important to note that the wheels are not capable of applying the force necessary to balance the diagonal (parallel to the ramp) component of the weight of the system, for this another system is required (thus in this diagram only the perpendicular component of the weight is taken into account). Given the dimensions of the system, solving Newton's laws (equation 10) yields the contact forces that the wheels experience (and therefore the railing). This force, which is likely the greatest in this system, can be studied in order to anticipate any failures in the wheels, railings, and any bolts that attach the wheels to the body.

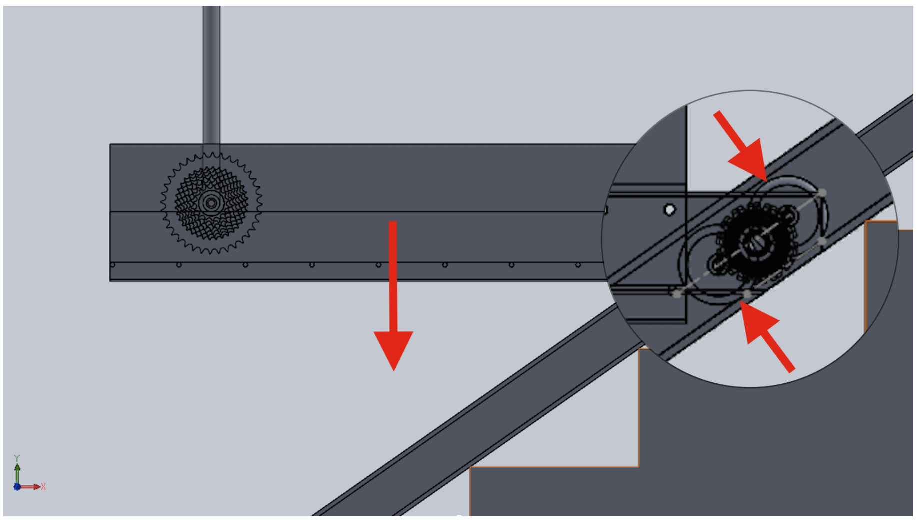

Figure 11: Free body diagram of the external forces acting on a wheelchair lift

Similar analyses can be done on wheelchairs by using contact forces at any points touching the ground, and taking into account the center of mass of the system. Of course, a more in depth analysis of the forces acting on individual components can become considerably more complicated, but the theory behind it remains the same.

3.5.5 Stress Analyses

Knowing the stresses that act on a component is the final step in determining if a system is structurally stable. Methods for finding stress vary, although more complicated geometries tend to use numerical analysis in order to solve (a popular choice for analyzing these types of problems is the computer software ANSYS). Despite any disparities between methods, there are certain assumptions and theories that can always be used. Two simplifications can be made for the case of very thin or very thick materials, for the latter plane strain can be used, and it assumes that the strain perpendicular to the face being studied is zero. The former, applied to thin objects, assumes that the stresses perpendicular to the face being studied are zero. These assumptions can simplify problems from three-dimensions to two, greatly reducing calculations.

In addition to plain stress and strain, beam theory is another application of structural mechanics that may be applicable when looking at wheelchair lift systems. Although, once again, numerous computer software can be used in order to solve these kinds of problems, it is important to know the assumptions that make them possible. Specifically, beam theory assumes that the deflection of the beam is small compared to the length of the beam, and that the material is homogeneous, such that the deflection can be described by a circular arc. Beam theory is highly applicable in engineering, and is simple to use as it is a one-dimensional problem.

Lastly, when viewing the results of a stress analysis it is necessary to choose a yield criteria in order to detect any failures. Although several different ones are commonly used, a favored option is the Von Mises stress (particularly when looking at ductile materials such as metals). The Von Mises criterion states that yielding (and therefore failure) occurs when the elastic energy of distortion reaches a certain critical value. In essence, the Von Mises criterion examines all of the stresses acting on a certain plane and determines if it is likely to fail. The most general form of the Von Mises equation is given as

$$\sigma_v = \sqrt{\frac{1}{2}[(\sigma_{11} - \sigma_{22})^2 + (\sigma_{22} - \sigma_{33})^2 + (\sigma_{33} - \sigma_{11})^2 + 6(\sigma_{12}^2 - \sigma_{23}^2 - \sigma_{13}^2)]} \tag{11}$$

where σ_{xy} are the normal and shear stresses in all directions. If this stress surpasses yield stress then the structure is said to fail. In fact, it is common application for engineers and scientists to use a factor of safety in order to maintain the material far from yielding.

4 Significance

The 2010 U.S census estimated that there were 3.6 million people in wheelchairs in the United States alone. Other studies performed in France, the UK, and the Netherlands found that approximately 1% of the European population uses wheelchairs [2][1]. Developing countries such as Cambodia and Afghanistan experience several times these numbers with percentages ranging from 5-15% due to the inefficiencies in treating diseases like cerebral palsy, and violent attacks.

Using a staircase is often taken for granted but can be a source of difficulty and life-threatening danger for persons confined to wheelchairs. In some emergency cases, such as fire or evacuation, the ability to descend a staircase independently could mean the difference between life and death, or severe injury. The use of elevators are often restricted or banned for building fires and other emergency situations, meaning disabled persons will often need to find alternative methods of evacuation. Although commercial evacuation products exist, these devices are used only for descent, require the individual to transfer from one vehicle to the other (and mobility after descent becomes restricted if not impossible due to the loss of the original wheelchair), and still often require the aid of another individual [9] [10].

Multiple studies in psychology have also shown the relationship between disability and emotional health [5]. The loss of mobility and independence can often be crippling for the mind; leading to depression, feelings of inadequacy and emotional trauma. The effects of social stigma are also significant to an individuals self-perception and mental health [6]. The standard wheelchair has already granted independence and mobility to the disabled, but further progress on improving wheelchair functionality would facilitate disabled individuals to have a positive role in society, a necessary condition for the advancement of disability rights and the reduction of social stigmas [7].

Additionally, developing countries often lack many of the amenities for disabled persons that have become standard practice here in the United States, including alternative routes (ie: ramps, elevators, etc.) for wheelchair occupants. According to the World Report on Disability, a survey of 36 countries in Asia and the Pacific showed that ∼28% did not have accessibility standards for the built environment or public transportation [8]. Although stair-climbing wheelchairs (SCW) and similar technologies do exist to address the issue, many current solutions are slow, costly, and/or arduous.

5 Design and Study Approaches

Over the course of the design process, SCW type systems were first considered with evaluation of the design proposals to follow. Ultimately though, complexity and cost of an SCW system led to the development of a manual lift system as the solution that more closely aligned with the design objectives and constraints.

5.1 SCWs

Several manual wheelchair designs were studied before finally deciding to change the direction into lifts. Manual wheelchairs were preferred over electrical ones due to their relative simplicity and overall inexpensiveness. The few designs considered are explained in more depth below.

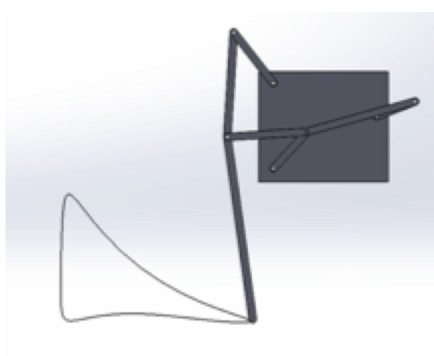

Figure 12: Klann linkage mechanism showing the motion of the leg

One potential idea reviewed was the use of linkage systems attached to a wheelchair that would mimic a motion capable of traversing stairs. Specifically, the Klann linkage, patented by Joe Klann in 1994, appears to approximately follow the movement of a human leg. Of particular interest was the Klann linkage's ability to have a high rise displacement relative to its horizontal movement (making it potentially adaptable to climbing stairs). Figure 12 shows the motion that this linkage system creates. It is easy to see from this figure that such a movement could definitely have the potential for ascending or descending stairs. Figure 13 shows what a potential configuration of this type could look like. In the image there are 8 individual Klann linkages (with 4 moving in unison, and the other 4 at a 180°phase shift). Such a system would always have four legs on the ground, making it stable, and very unlikely to slip when climbing a set of stairs. Despite this, the entire system is simply too large, bulky, and contains too many moving parts, making it a poor design to attach to a wheelchair.

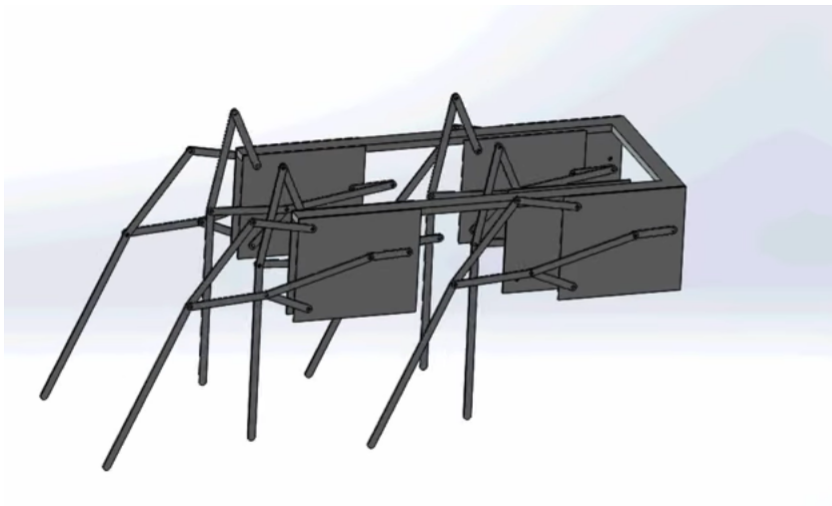

Figure 13: Possible configuration of Klann linkage system

Another design considered was an already existing idea built and tested by the Mechanical and Aerospace Engineering Department at King Mongkut's University of Technology North Bangkok (KMUTB) in 2013. The design (shown in figure 14) is powered by rotating the large wheel and is capable of both ascending and descending stairs at a moderate pace. Although seemingly bulky, this design is perhaps the most feasible of it's type, as it allows the user to use relatively little force while providing good stability. Unfortunately, the design suffered from a fatal flaw. In many occurrences, as can be seen from video footage, the wheel-like components in contact with the stairs (while climbing) would land on the edge of the staircase, when this happened the device was likely to slip and result in the person potentially falling over. As safety is a primary consideration, the design was deemed hazardous and discarded.

Figure 14: University of Technology North Bangkok [28]

5.2 Manual Lift

5.2.1 Design Progression

The initial design proposal is pictured below (Figure 15) with a rough idea of what the main components would consist of and how the system would generally operate. Note the lack of realistic attachment/support points for the platform and railings. At this point the design resembles a simple inclined rail-car system with a platform as the car and a manual drive mechanism that is facilitated by a lever drive system. The appropriate gear sizing is not present and large wheels are used to provide normal support to the platform.

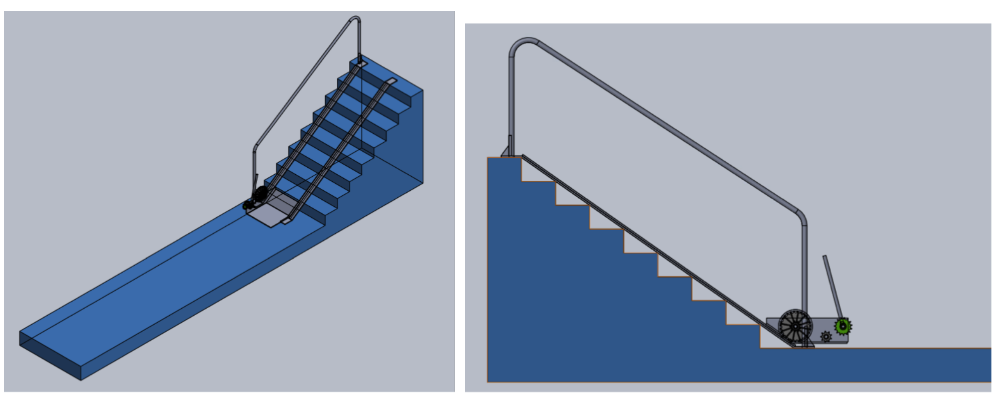

Figure 15: Diagram of initial manual lift proposal: isometric view (left) & side view (right).

From this point, further evaluation and refinement of the proposed design led to the interim design as depicted in figures 15 - 18. Realistic attachments, supports, and brackets are included, as well as more accurate, relative sizing of sprockets in commercially available sizes/types. Dimensions that are independent of strength analyses are included according to standard specifications. The rails are secured to the staircase and all components are present except for the chain and braking system. Note also, the inclusion of a bicycle sprocket cassette that is used to change the sprocket ratios on the lever drive so that users can adjust mode of operation to their liking and ability. The gear switching mechanism will resemble that of the derailleur systems commonly found on bicycles with gear switching capabilities.

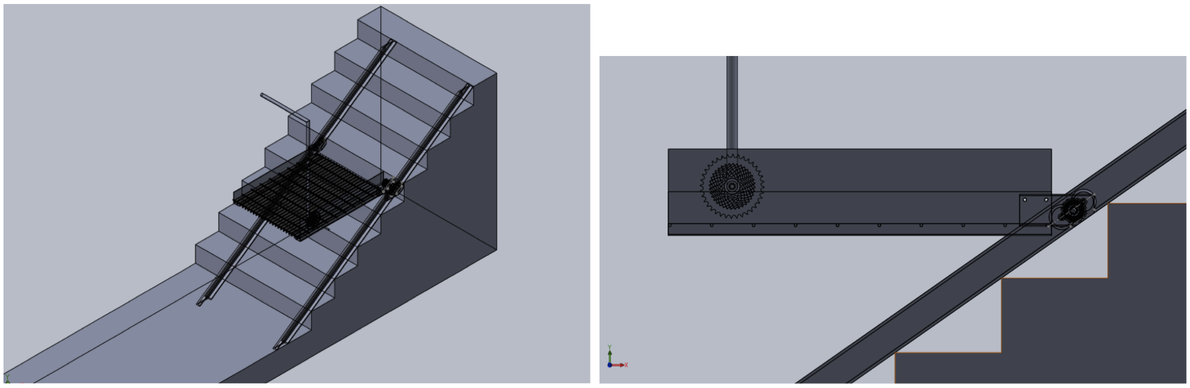

Figure 16: Diagram of final manual lift design: isometric view (left) & side view (right)

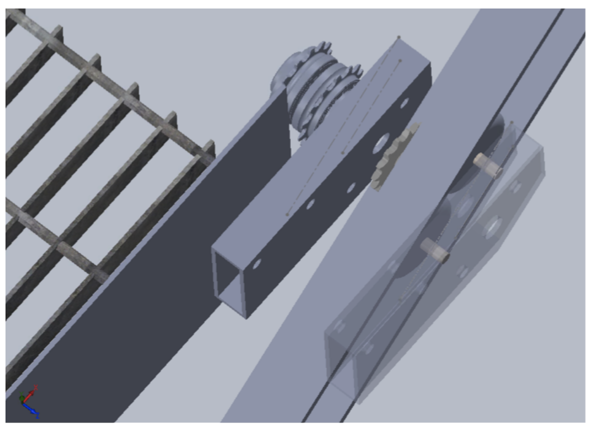

Figure 17: Diagram of final manual lift mechanism

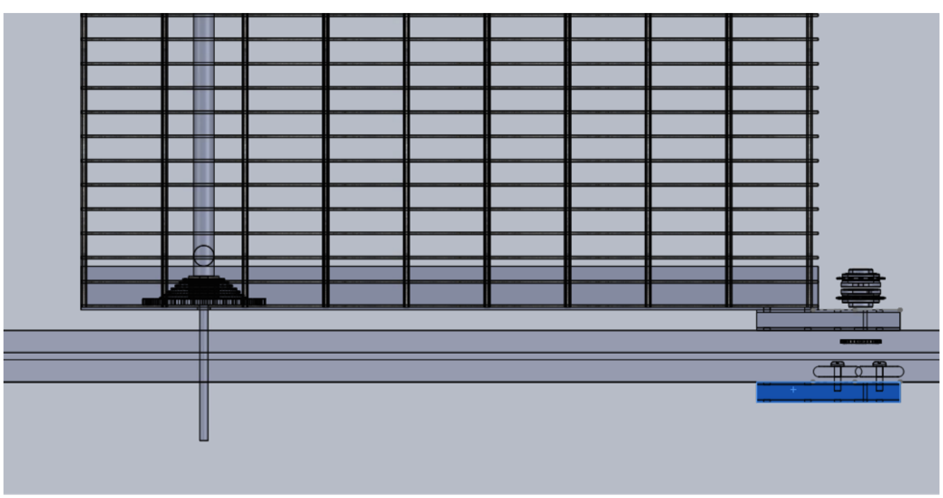

Figure 18: Diagram of final manual lift mechanism: top view

In the current mode, the lever drive would allow for ascension of the stairs, but a braking system is needed to prevent slippage - once the lever is released - and a method of descent is required. Under consideration for brakes were self-locking brakes, a "ratcheting brake", and a Weston-Brake. Although it was determined that the self-locking mechanism would not be a feasible option, the latter two methods are briefly explained below.

Ratcheting Brake:

A simple ratcheting mechanism could be mounted on the platform and joined to the end of the shaft to act as a brake. The solution would be relatively cost effective by using an existing ratcheting socket wrench secured to the output axle machined with a hexagonal male end. The wrench would then be grounded to the lift platform and/or brackets. However, this proposed solution would require a separate method for descent. involving the disengagement of the socket wrench, or a directional switch on the ratcheting device. The directional switch mechanism used must then disallow the possibility of inappropriate ratcheting direction during ascent or descent that could cause the platform to disengage and slide down the rails, uncontrolled (major safety consideration).

For descent, a rotary damper placed on the output shaft was considered. It was proposed under the assumption that a ratcheting or self-locking brake would be used to prevent slippage. The rotary damper would be sized in order to achieve a shaft rotation rate that would allow for comfortable and safe descent of the platform. The sizing would have to be calculated according to the maximum speed allowed and under maximum loading. Further research of the topic revealed that dampers of the required size were not available or were too costly. A viable alternative may involve disc or drum-brakes such as those used on Go-karts and mountain bikes.

Weston Brake:

The Weston brake (depicted in figure 19) was developed by Thomas Weston in the late 1800's, and has a design that gives it some very interesting properties. The benefit of the Weston brake is in that it allows the user to rotate an axle in both directions while still being sure that any large force will cause the entire system to lock. As seen from figure 19, as the user rotates the handwheel in one direction, the threaded portion of the pinion shaft will tighten all of the components together allowing everything to spin in uniform. When the user lets go and the force of the item being lifted tries to spin the brake in the other direction the ratchet disk and the pawls prevent any movement. Now, when the device is spun in the other direction, the thread becomes lose, and the friction plates lose their contact, sending the load into free fall, however as this happens the load spins the threaded bolt back in, everything locks once again, and the ratchet prevents further motion. This back and forth movement allows a controlled continuous descent to take place, making this device very practical for this particular situation.

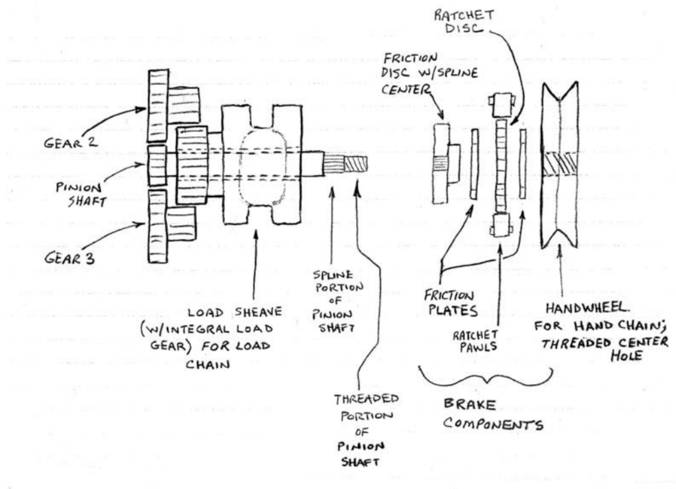

Figure 19: Components of the Weston brake [29]

5.2.2 Plan of Analysis and Evaluation

In order to evaluate whether the system is effective and achieves the stated objectives, stress and strain analyses were performed to obtain component dimensions which satisfy the codes and standards (the lift should operate as described and deflections should remain within those specified). High-stress components were evaluated using ANSYS Mechanical Analysis software. Following, preliminary cost analyses were performed to evaluate the total cost of the system; based on these results, the design was modified to optimize for cost and weight. A preliminary prototype of the lever drive mechanism was also constructed to provide a proof of concept validation before continuing in the design process.

The system prototype was then built to evaluate the performance of the proposed design. The resulting stresses on components using load cells were not obtained experimentally for verification of theoretical results due to funding constraints, however the manual lift could be tested with maximum loading to verify that standard specified capacity rating was achieved. Final cost of components were conducted based on wholesale pricing for comparison with other methods of achieving the same objective.

Note that, for the prototype, rails were not anchored permanently to the sample staircase being used; temporary attachment methods were used during evaluation of the design. Although the ideal prototype is full-scale, the length of the railing was reduced for cost and ease-of-testing purposes.

Over the course of the design study, modifications to the design were implemented as deemed necessary, the most significant of which will be discussed.

5.3 Theoretical Design Modifications

Theoretically based modifications refer to those which were made as a result of ANSYS modeling (Appendix A), manual calculations, commercial availability, and consultation with machining/manufacturing professionals prior to preliminary prototyping and testing. Significant changes to the preliminary design included the substitution of a C-Channel with additional chain guides for the I-beam as railing, wheel size and orientation, a hinged ramp/foot-guard, bolt sizing, elimination of the bike derailleur concept, a disengaging mechanism, and a controlled gravity-descent mechanism.

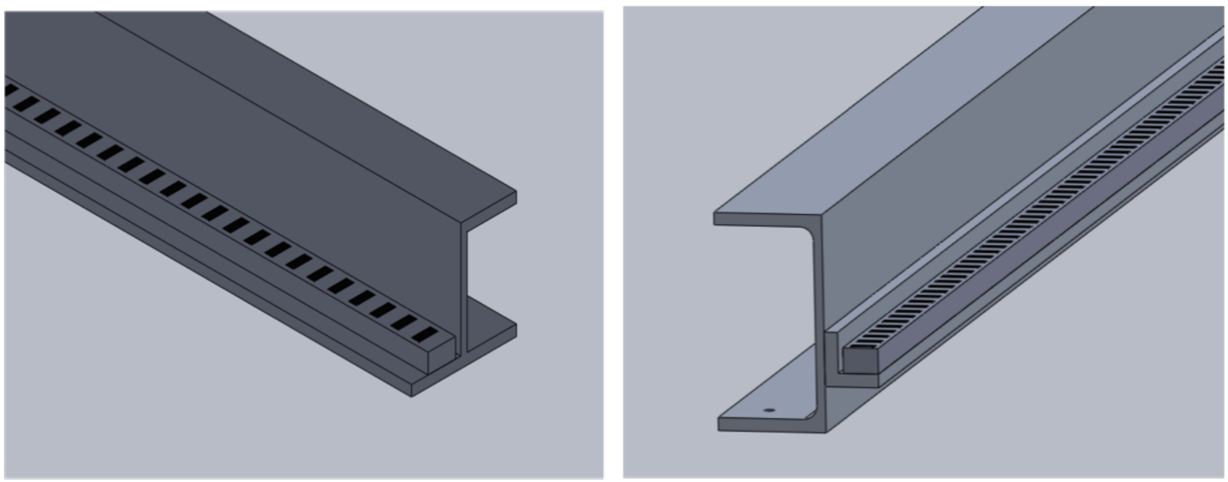

Figure 20: Solidworks images of I-beam and C-channel rail geometries for comparison. C-channel railing with angle bracket was chosen for ease-of-machining purposes.

Due to limitations in local machine shop capability, the I-beam rails were replaced with C-channels to which a small angle is attached as a chain guide. The design is modular, simpler to manufacture and produce, and cheaper in terms of machining costs. C-channel dimensions are an American Society for Testing and Materials (ASTM) 4in. standard Aluminum channel (web 0.16in. thick) and able to accommodate the 3in. diameter wheel used. Results of ANSYS demonstrate a Factor of Safety (FS) greater than 10 for the dimensions chosen.

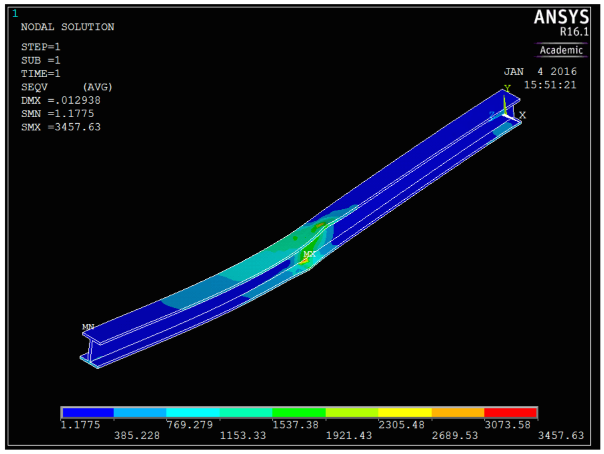

Figure 21: ANSYS Static Mechanical Analyses revealed F.S. greater than 10 for Aluminum 6061-T6 (yield strength ∼45000psi).

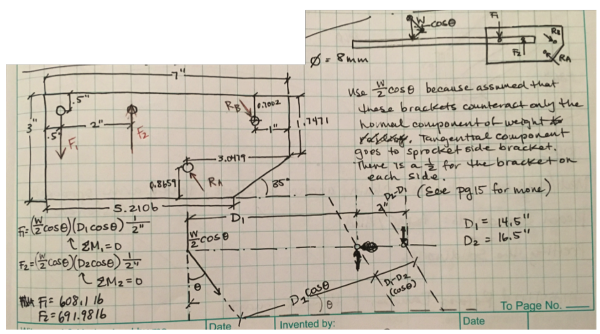

Figure 22: Manual Calculations of force caused by weight of platform and external load of operator indicating the need to increase the distance between wheels.

Results of product search and force calculations resulted in the selection of a 3in diameter nylon wheel rated for 660lbs loading. Spacing between the wheels was increased from ∼2.5in. to ∼6in. to reduce stress on each wheel as seen in Figure23.

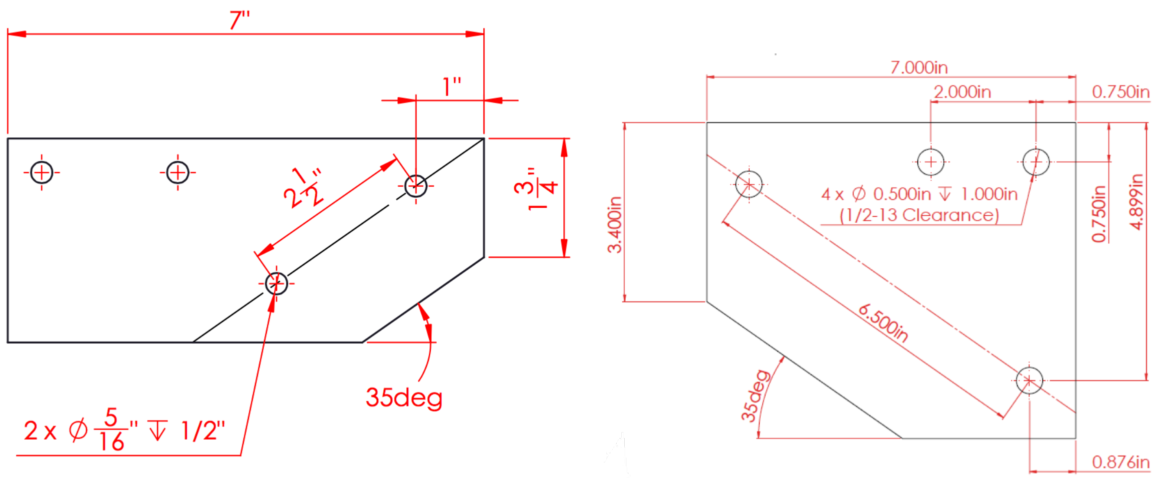

Figure 23: Drawings of old design (left) and new design (right). Increased spacing between wheel axles reduces the stresses induced on both.

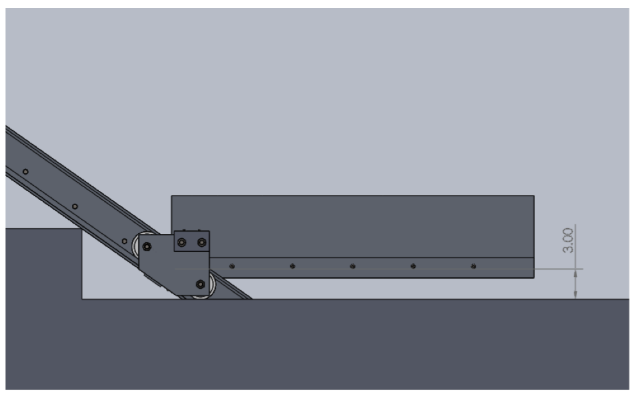

Figure 24: Image of Solidworks model demonstrating gap between platform face and ground necessitating a ramp/foot-guard device to aid in user mounting of lift.

Evaluation of a Solidworks model of the complete assembly revealed an initial platform elevation approximately 3in. above the ground. A hinged platform design was proposed to act as a both a ramp and foot-guard for the final design. This component was not incorporated into the prototype due to budget limitations.

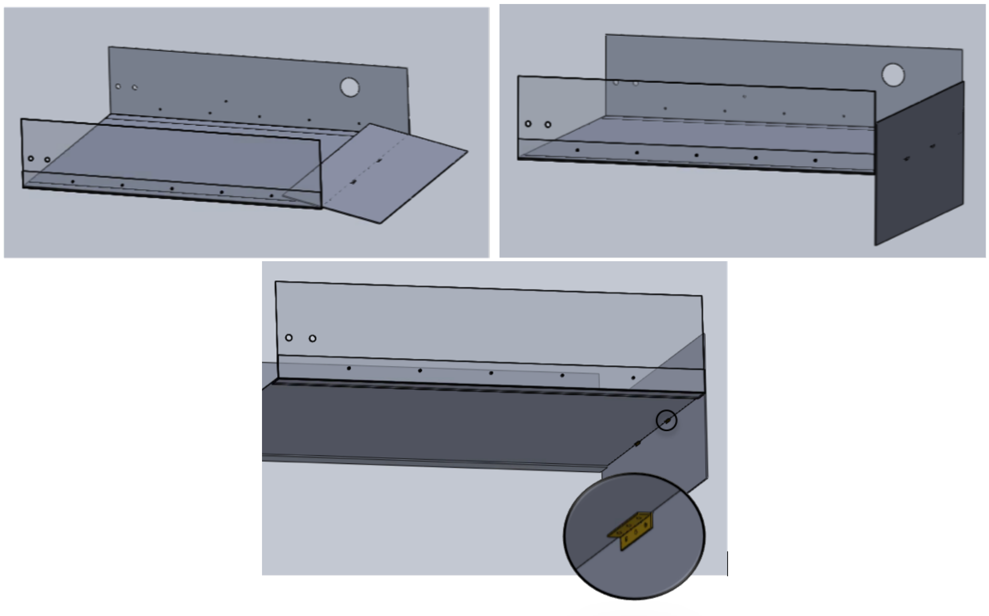

Figure 25: Solidworks models of proposed, hinged ramp/foot-guard design

Main, load-bearing bolts were increased from 0.25in. diameter to 0.5in. diameter based on manual calculations of shear stress, where force obtained through a moment diagram of the loads as seen in Figure 11.

$$\tau = F/A \tag{12}$$

Upon consultation with bicycle experts, it was determined that the driving sprocket in the current application was spinning at too low an rpm (and high torque) to use a derailleur in switching gears on the sprocket cassette. Although versatility is decreased in doing this, the idea was thought likely to fail and was removed altogether from preliminary prototype testing. Future designs may implement an alternative transmission device for sprocket/gear switching. Despite derailleur omission, preliminary testing of the lever drive mechanism was executed as pictured below (Figure: 26) to validate the mechanism concept design.

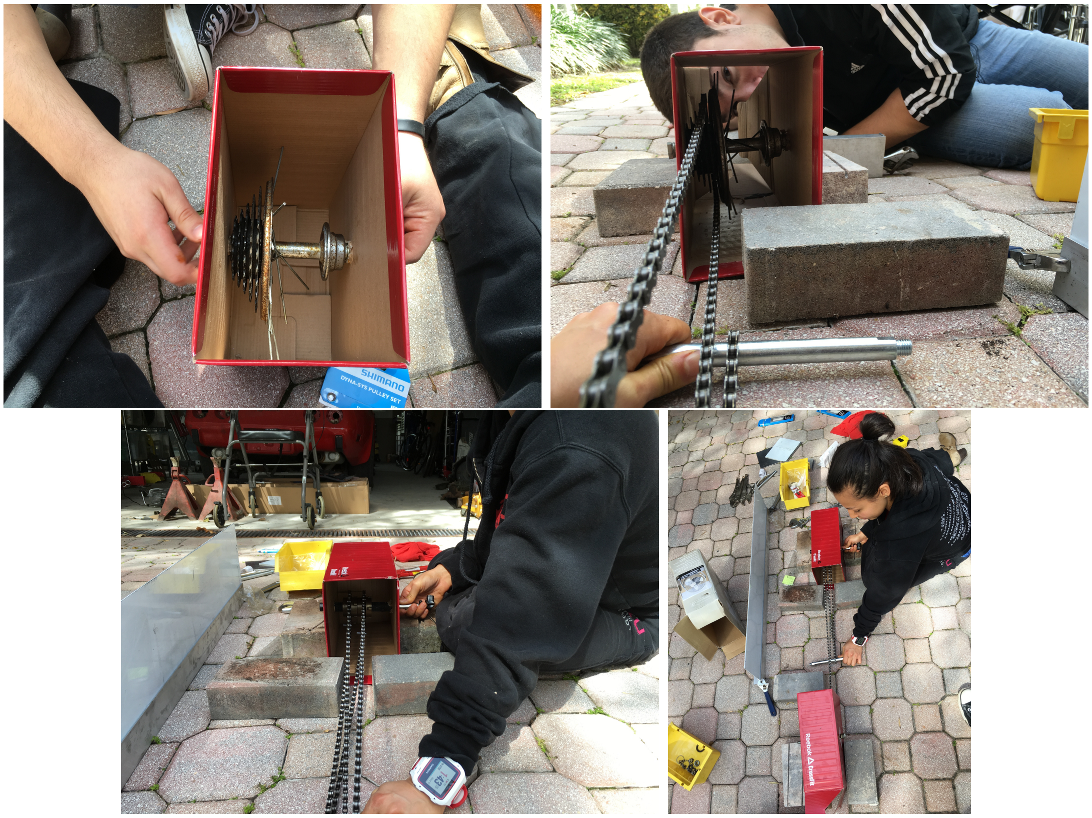

Figure 26: Pictures of preliminary prototyping to test drive mechanism as a proof of concept.

It was discovered early on in the prototyping stages that the lever drive mechanism presented will not allow the axle to spin in the reverse direction (and thereby letting the whole mechanism descend). In fact, there is no simple solution consisting of freewheel sprockets. This is because if the ratchet catches the pawl when rotating the axle clockwise (for example) then it will also do so when the axle tends to rotate counterclockwise. Although a seemingly trivial fact, once observed, it was evidently a significant factor. In order to make the lift fully functional a mechanism had to be designed to fix this problem. The solution, as seen in figure 27, is comprised of the two freewheel sprockets (which would have to be joined together in such a way that motion is not restricted, ie: welding or epoxy) attached to an internal ring gear. This assembly does not spin with the axle, although it is restricted from moving along the axle. A matching spur gear, keyed to the axle but not restricted from lateral movements, then comes in and attaches to the internal gear to lock everything together and ascend. If descending is desired then the spur is simply disengaged from the system, at which time the drum brake (described below) will activate and allow for controlled descent. The Cad drawings portray the basic movement and joint types that would be required in order to engage/disengage the mechanism from a location suitable for the lift rider. One important feature of the assembly is the springs. They're function is not only to reengage the setup, but also to ensure that even if the gear teeth do not initially match, everything will fall in place when the sprockets are turned a small amount.

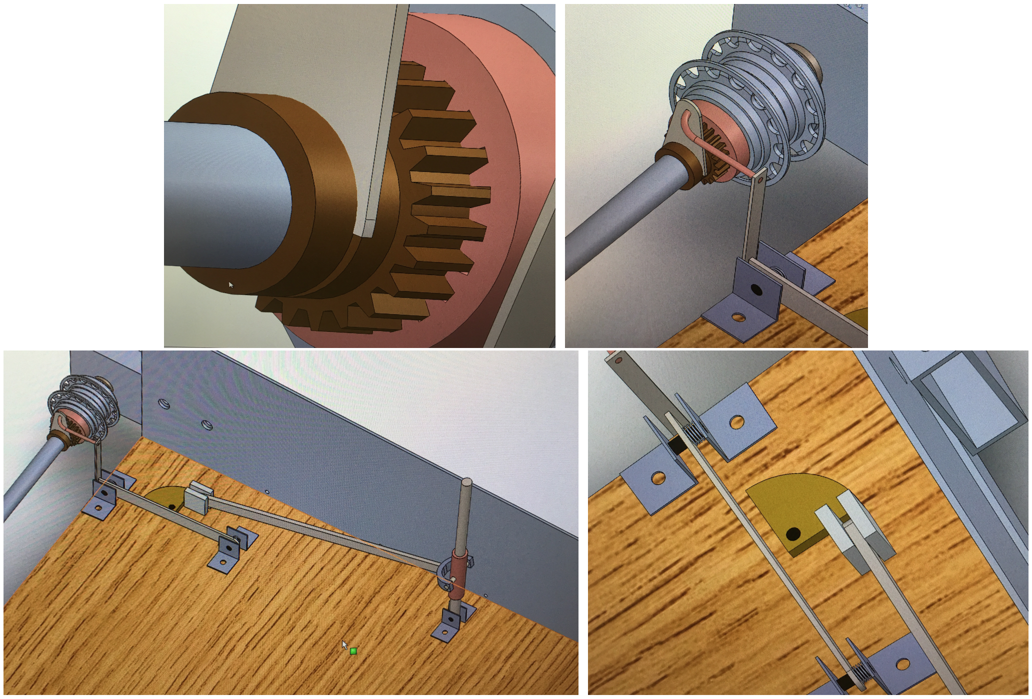

Figure 27: Engaging/disengaging mechanism.

Lastly, a descending mechanism (which had not been previously discussed) was designed so that the lift could be fully functional. Although not applied to the prototype (due to shortages in both funding and time), the assembly consists of a Go-Kart drum brake system mounted on an auxiliary axle. Seen in figure 28, the brake drum is constantly held taught by use of the spring in the brake handle. As a result, the smaller auxiliary axle is not allowed to spin at any time. The main axle, attached to the smaller one through the two free wheel sprockets and chain setup, can only rotate in the direction in which the ratcheting is occurring, thus preventing the system from rolling down uncontrollably under any conditions. If the objective is to descend then the brake is released slowly so as to allow the auxiliary axle (and thereby the main axle) to rotate in the other direction. In order to be a safe system, the brake needs to be calibrated such that even when the handle is compressed completely the band on the drum still provides enough friction to prevent uncontrolled descent. Suffice it to say that the design needs to be prototyped and tested before a conclusion about its efficiency and capabilities are stated, however if the brake does not provide a smooth motion then a bicycle disk or an inner drum brake assembly can be implemented using the same principles.

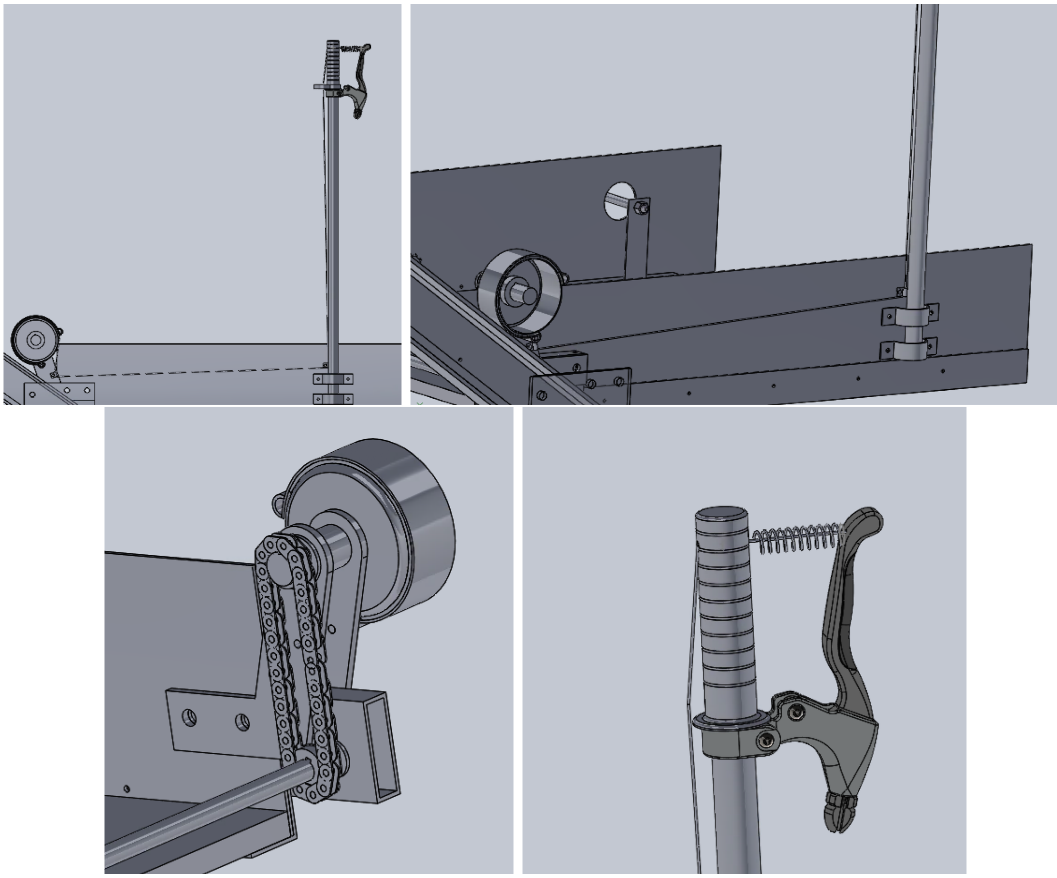

Figure 28: Braking/Descending mechanism.

6 Results and Analysis

6.1 Practical Design Modifications

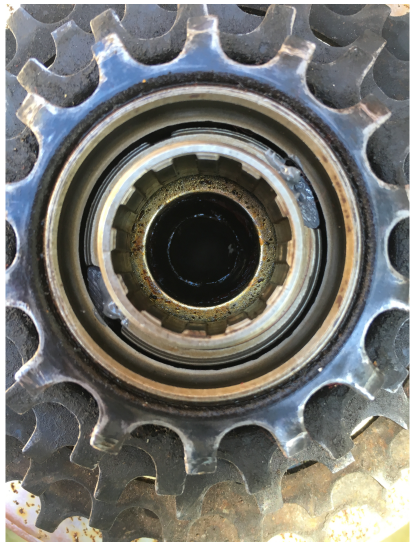

Figure 29: Picture of implemented sprocket cassette fixture design with steel pins to maintain "activated" or outward pall position.

Upon preliminary testing of the drive mechanism, several design flaws were detected and accommodated for. Firstly, the main drive sprocket obtained (from an old bicycle) was a freewheel sprocket cassette. This presents a problem since the design requires a fixed sprocket in order to be able to apply tension on the chain when both pushing and pulling the lever. Therefore, the cassette was made a set of "fixed" sprockets by securing the pawls in an outward or "activated" position to disallow freewheeling of the sprocket. To secure pawls against high torque on the sprocket cassette, steel pins were inserted and epoxied in place such that significant deformation of the solid steel rods would be required for failure of the component.

Hertzian contact stress calculations (see Eq: 13 - 17) verified the safety of the design solution.[31]

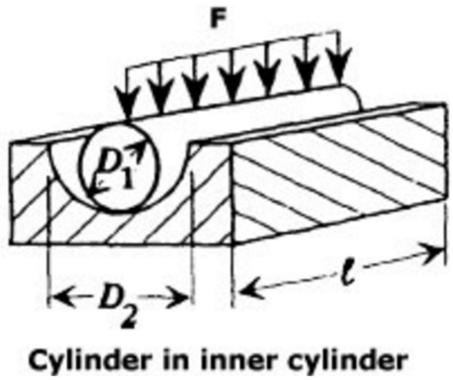

Figure 30: Diagram of Hertzian contact stresses resulting from force F applied to two cylinders, one of which is concave (referred to as cylinder 2).[32]

$$b = \sqrt{\frac{2F}{\pi L}\frac{\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}}{\frac{1}{d_1}+\frac{1}{d_2}}} \tag{13}$$

$$p_{max} = \frac{2F}{\pi b L} \tag{14}$$

where, b is the half-width of rectangular contact area between the two cylinders, p_{max} is maximum pressure, F is the applied force, L is the length of the contact area, ν_i is Poisson's ratio, E_i is the elastic modulus, and d_i is the diameter of the cylinders; the index i referring to cylinders 1 or 2. Note that d is made negative for the concave cylindrical surface ($d_2 < 0$).

$$\sigma_1 = -2\nu p_{max}\left(\sqrt{1+\frac{z^2}{b^2}} - |\frac{z}{b}|\right) \tag{15}$$

$$\sigma_2 = -p_{max}\left(\frac{1+2\frac{z^2}{b^2}}{\sqrt{1+\frac{z^2}{b^2}}} - 2|\frac{z}{b}|\right) \tag{16}$$

$$\sigma_3 = \frac{-p_{max}}{\sqrt{1+\frac{z^2}{b^2}}} \tag{17}$$

where, $\sigma_{1,2,3}$ are the principal stresses.

Following assembly of the system prototype it was discovered that the use a of a single chain on one aluminum channel to provide platform lift resulted in an imbalance of forces, causing the entire platform to slant and thereby exert an unaccounted twist on the wheels and their supporting structure (Figure 31).

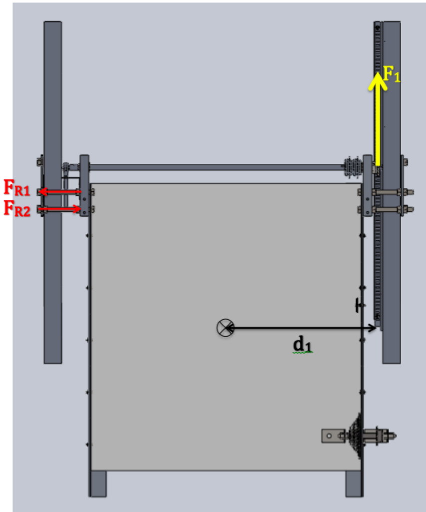

Figure 31: Free-body diagram of forces on the platform with a single chain drive. The force, F_1, generates a moment about the center of gravity (c.g.) to generate a moment, $F_1 d_1$ that must be counteracted by the reaction forces, F_{R1}, F_{R2}. Since the wheels are not confined to the railing, in this direction, the entire platform rotates counterclockwise about the c.g. until the socket-wrench on the output axle strikes the railing.

The asymmetric distribution of forces was an overlooked design defect, causing a catastrophic inability for the system to work efficiently. To counteract the moments, a caster wheel was originally installed between the platform brace and the non-drive side channel (Figure 32). Although the caster did not fail in compression, the forced applied were too great and caused the wheel to seize. Seizure then produced friction between the caster and the channel causing system failure. Instead of the caster, a second chain was then added to the C-channel in order to properly balance all of the forces as seen in figure 33.

Figure 32: Picture of failed attempt to implement caster wheel solution to balance the forces.

Figure 33: Assembly depicting dual chains.

Although the design modification required the removal of the ratchet from the end of the axle, the braking mechanism proposed previously made the need for the ratchet obsolete. Surprisingly, however, the complete functionality of the lever drive mechanism was overlooked. Although it is perfectly capable of spinning the axle in one direction, the opposing forces created by the chains on the sprocket cassette make it impossible for the axle to spin the other way. Overlooking this fact brings complications when a transition from ascending to descending is required. This development led to the understanding that the freewheeling sprockets in the drive mechanism would have to be disengaged simultaneous to brake application for controlled descent. Potential design solutions are discussed for future directions, but were not implemented or tested in the current design due to budgetary and time constraints.

The lever drive mechanism initially proposed also encountered difficulty during implementation. Although the mechanism operated smoothly under low torque conditions, high torque coupled with short idler pulley (obtained from derailleur mechanism) teeth caused the bicycle chain to slip during operation. Consequently, an alternative method of chain threading was used to achieve the same objective as shown in figure 34 below. The new setup operates on the same principles as the lever drive. The primary difference is that it uses two chains (one arranged in a loop and another in an eight configuration) and thus does not require use of an idler. The largest and 3rd largest sprockets in the cassette were used for this method. It should be noted that there is no flaw with the initial design and it can be carried out with more careful considerations and precise machining.

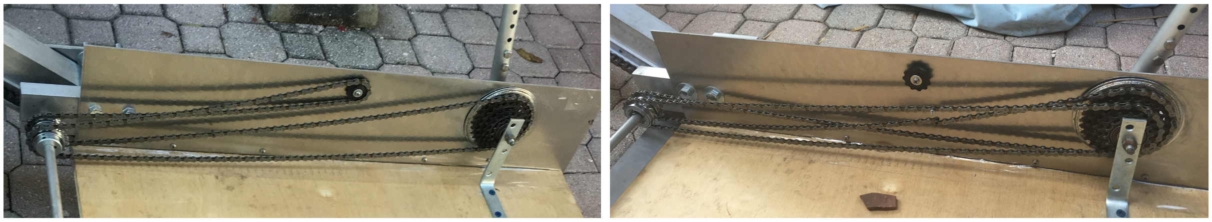

Figure 34: Initial chain setup using idler sprocket (left) vs. current model (right)

Further testing also revealed an unacceptable deflection in the platform under simple self-weight of the platform. Initial design calculations did not account for the scissoring of the bolts connecting the wheel structure to the main body. This design flaw, coupled with imperfections in machining and tolerances resulted in excessive tilting of the platform downwards, making it impossible for a person to safely operate the device. In order to accommodate this issue, further reinforcements were added between the main body and the wheel structure including additional 1/4in. aluminum plating and angle brackets to provide necessary stiffness. The results of the improvement can be seen below

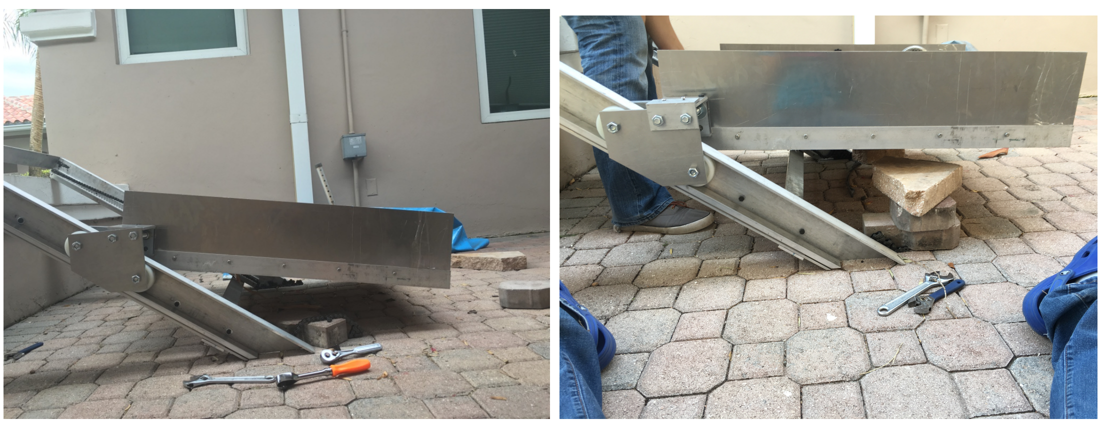

Figure 35: Comparison of platform deflection before and after the application of reinforcements.

Slighter issues required simpler modifications to the prototype and overall design which do not require detailed discussion in the current paper. Examples include: keywayed freewheel sprocket adapters used to properly secure output axle, aluminum reinforcements to prevent C-channel displacement during testing, C-channel aluminum filler bars to reduce clearance between wheels and rails, copper tubing used as spacer for the rectangular tubing (as compressive forces from bolt-nut tightening could cause tubing deformation or collapse), and passing the output chains over the output sprockets as opposed to below them (similar to rack and pinion arrangements). The last solution was found to improve slipping issues, and did not demand as stringent dimensional requirements as those for the chain passing under.

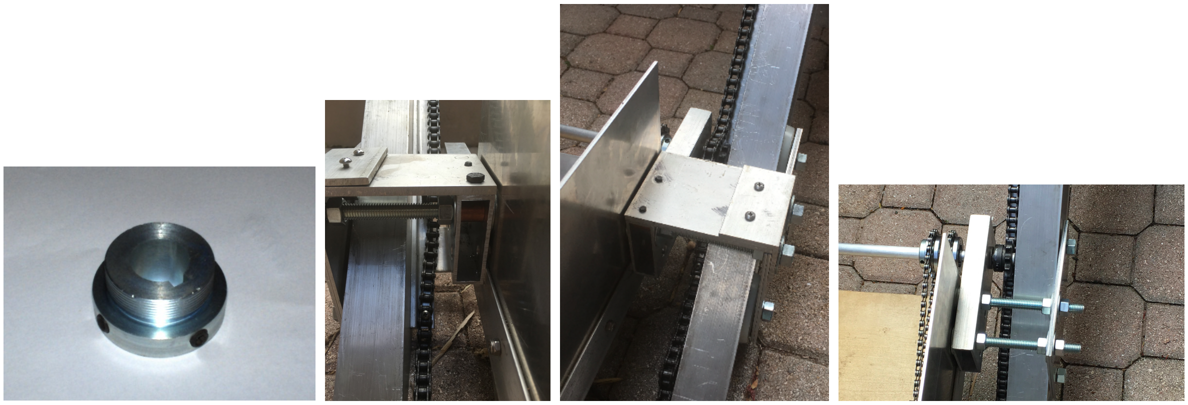

Figure 36: Pictures of thread-on, keywayed, freewheel adapters[33], implemented copper tubing spacers, and comparison of under and over chain orientations (left to right)

Numerous other modification were made during and after machining and testing, but the majority (apart from the components mentioned above) did not comprise any major design changes. Many post design modifications were made in order to facilitate or expedite the machining process. The final design along with the corresponding results are discussed in the following sections.

6.2 Final Product

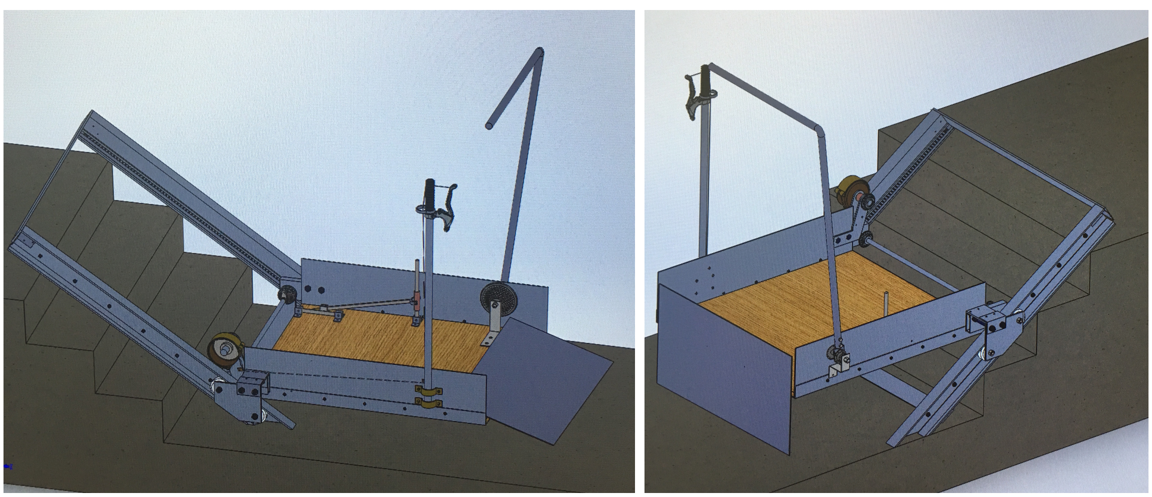

Figure 37: Final CAD model of completed lift assembly.

Figure 37 shows the final design with all of the subassemblies incorporated. This system, if properly built, should be fully capable of performing all of the functions required of a lift of this kind. Any additional modifications were not researched in this project, although a few suggestions are presented in the "Future Work" section below.

6.2.1 Final Prototype

Figure 38: Final prototype built for testing.

Above, in figure 38, are photos of the final prototype that was used for testing. Based off test results (to be discussed), future improvements are suggested and can be found in the "Future Works" section of this paper. Note that the prototype differs from the final design due to budgetary and time constraints.

6.2.2 Drawings & Components

Generic/Purchased components are listed below with details on manufacturer/vendor in Appendix B.

- 4 of 3in. OD Nylon Wheels, 0.5" ID, 660lbs. capacity
- Fasteners from Mitchell's Welding and Machine Shop
  - Load-bearing bolts of Grade 5 Galvanized steel, while others are generic metal alloy screws, nuts, and washers.
- 2 of No.40 ANSI industrial chain, 5ft long
- 3 of Shimano HG54 10-speed bicycle chain, 3/32" wide
- 1 of bicycle sprocket cassette from 10-speed road bike
  - Used Schwinn 1983 World Sport 10-speed Racing Bike (19in. frame)
- 2 of 14T freewheel sprockets for 10-speed chain and compatible keywayed adapters (thread-on assembly) 5/8in. ID, 3/16" square keyway
- 2 of 40BS11 Output Sprocket, 5/8in. ID, 3/16in. square keyway

- 3/16in. square keystock, 4in. long
- 2 of steel pins, 0.125in. long, #12 gauge diameter
- Used, Standard Issue Walker (Medical Device) from 2005
- Angle/Corner brackets and 2 wood screws from Home Depot
- 2 of 5/8in ID Oil-Embedded Flanged Bronze Bushings, 7/8in. OD, 1 1/4in. long

Engineering drawings of major components requiring design-specific machining or manufacturing can be found in Appendix C. All aluminum components were of the 6061-T6 alloy, mild load fasteners were generic metal alloy materials, prototype platform was made of 3/4in. plywood, though formal design would incorporate expanded aluminum/steel or safety floor grating. Raw metal materials were purchased at Simmons - Surplus Stainless and are listed with prices in the cost analysis section.

6.2.3 Cost Analysis

Total cost of generic/purchased components totaled $229.72 from receipts (excluding 7% sales tax applicable in Miami-Dade County, FL). Total cost of raw materials was $193.53 and estimate of machining costs approximated to be $406.60 (quote provided by Mitchell's Welding and Machine Shop) for professional labor/services. As previously mentioned, almost all machine work for the current study were performed by the students who author this paper. Machining of axle and bushing installation were performed by professionals, *pro bono* (acknowledgments to Dr. Michael Swain, Dr. Matthew Swain, and Mr. Angel Morciego). Also note that retail prices of salvaged materials were estimated by Manufacturer Suggested Retail Prices (MSRP), or local business quotes when available, and totaled $37.50. Detailed breakdown of cost analysis can be found in Appendix F.

It is important to note that, throughout the design process, special attention was paid towards the cost of manufacturing and maintenance. In order to reduce maintenance and repair costs, the design is almost completely modular so that repair and replacement of parts would be both time and cost-efficient.

6.3 Testing

6.3.1 Procedure

The measurement of several variables is necessary to evaluate the manual lift system properly. Firstly, the load capacity and platform deflection of the lift should, exceed and be less than that required by the codes and standards, respectively. Secondly, the ascension rate achieved through comfortable operation should be determined through direct testing and compared to that of "The Funicular" project.

To test platform strength and deflection, the platform was loaded with incrementally increasing weight and the resulting deflection was measured. The test was performed in static condition only.

Weights were applied at the center of gravity as specified by the ISO 7176-28:2012 and ideal location of wheelchair relative to the platform (at ∼18in. from platform rear and centered, laterally).

Evaluation of the drive mechanism was performed without external loading (self-weight of lift platform only) and with the operator standing next to the drive side of the manual lift. The user operated the manual lift by the comfort of his own range of motion to obtain results (achieving ∼30deg angular displacement per push or pull, ∼60deg per push/pull cycle), such that ascension rate was measured in the units of inches per push/pull cycle. These results could then be extrapolated using average human power output to a time rate of ascension with certain assumptions. Note that elimination of the derailleur concept from the design consequently eliminates the need to test the drive mechanism under loading when obtaining ascension rate per push/pull cycle since sprocket ratios remain constant. Only the time-rate of ascension would require loading of the platform.

6.3.2 Platform Strength and Deflection

Although standards require the platform to sustain at least 300lbs (F.S. = 4), rail deflection less than 0.25in, experimental data far exceeded the expected/design values for safe operation, even following design improvements. Rails did not deflect by any measurable degree, however the platform deflection of approximately 0.9in. under simple self-weight (vertical difference between front and rear) was deemed completely unsatisfactory for safe operation. Figure 39 below represents experimental deflection data in degree decline (see Appendix E for data).

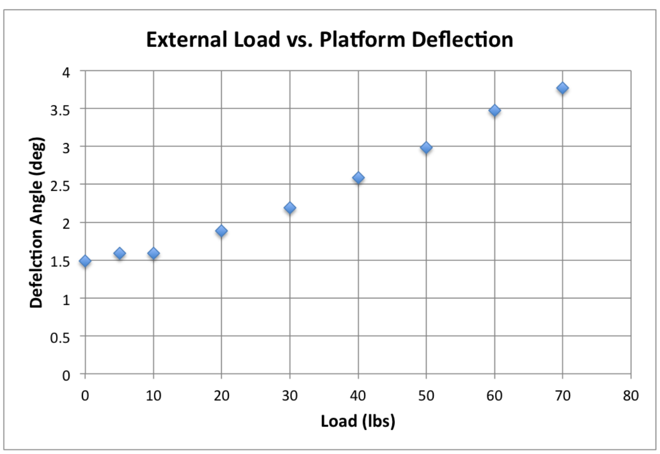

Figure 39: Graph of degree decline of the platform versus external load applied in accordance with standard testing procedure.

To investigate the reason for the disparity between expected system capacity and resultant prototype behavior, Solidworks static modeling techniques were used.

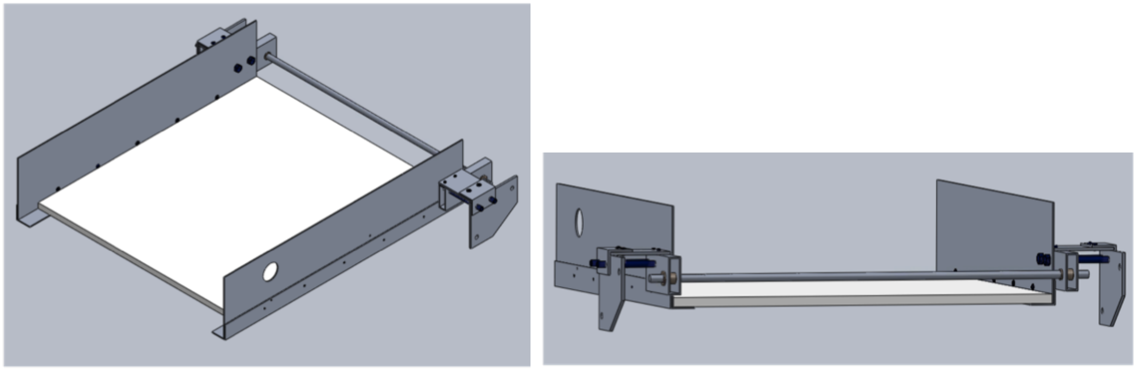

Figure 40: Image of simplified model used to run Solidworks static mechanical analyses.

A simplified assembly (see Figure 40) was tested under static conditions with boundary conditions at wheel axles and output axles as hinged and fixed fixtures, respectively. External loading included gravitational body force and 300lbs distributed force over the platform. All materials in the model were matching those used in the physical model. Global contact condition was set to "no penetration" and shrink fit contact sets were applied at the bushing-to-rectangular tubing junctions. Only load bearing bolts and screws were included in the analysis, remaining fasteners were left to be converted as fasteners through the automatic Solidworks simulation package. Curvature based meshing was used and the study was run using discrete solver. Global friction condition was set to that of Aluminum-Aluminum contact (0.57) at room temperature and 50% humidity [34]. Results of the analysis can be seen below in figures 41 - 43 with maximum deflection = 1.32in. (2.1^{o} decline), maximum strain = 0.8736%, and maximum von mises stress = 149,000psi.

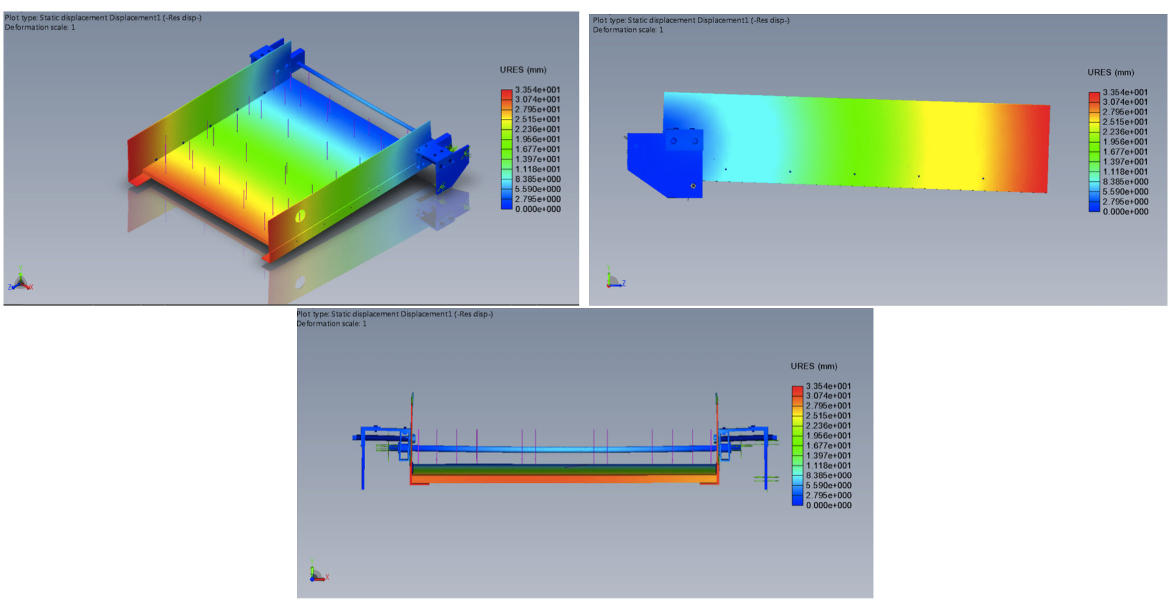

Figure 41: Image of displacement results for static analyses under 300 lbs distributed loading on the platform and gravity included for self-weights.

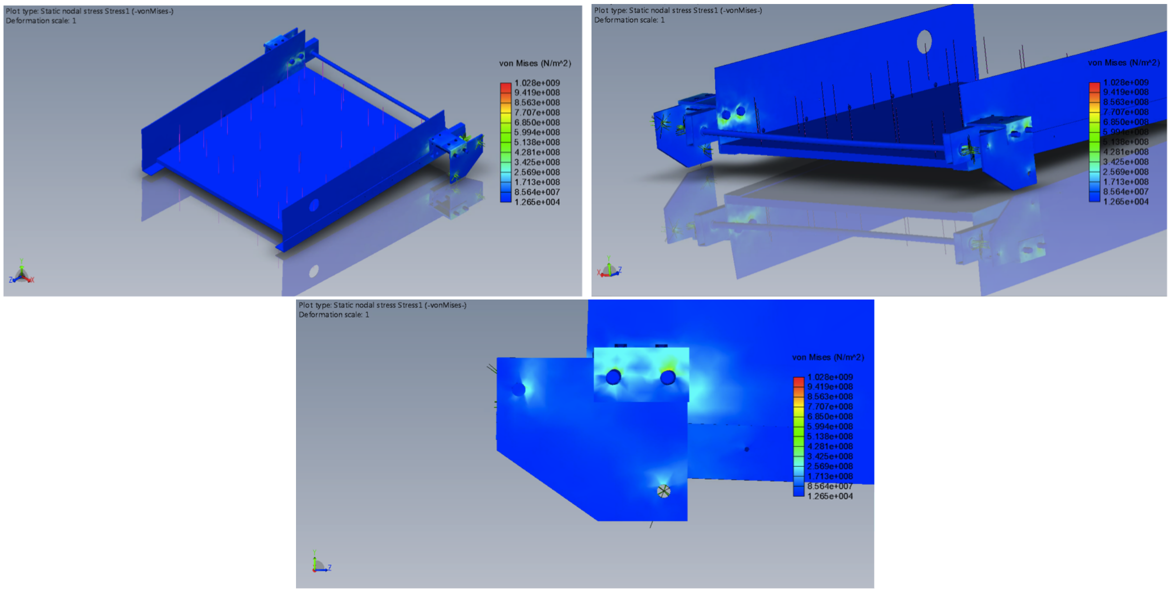

Figure 42: Image of stress results for static analyses under 300 lbs distributed loading on the platform and gravity included for self-weights.

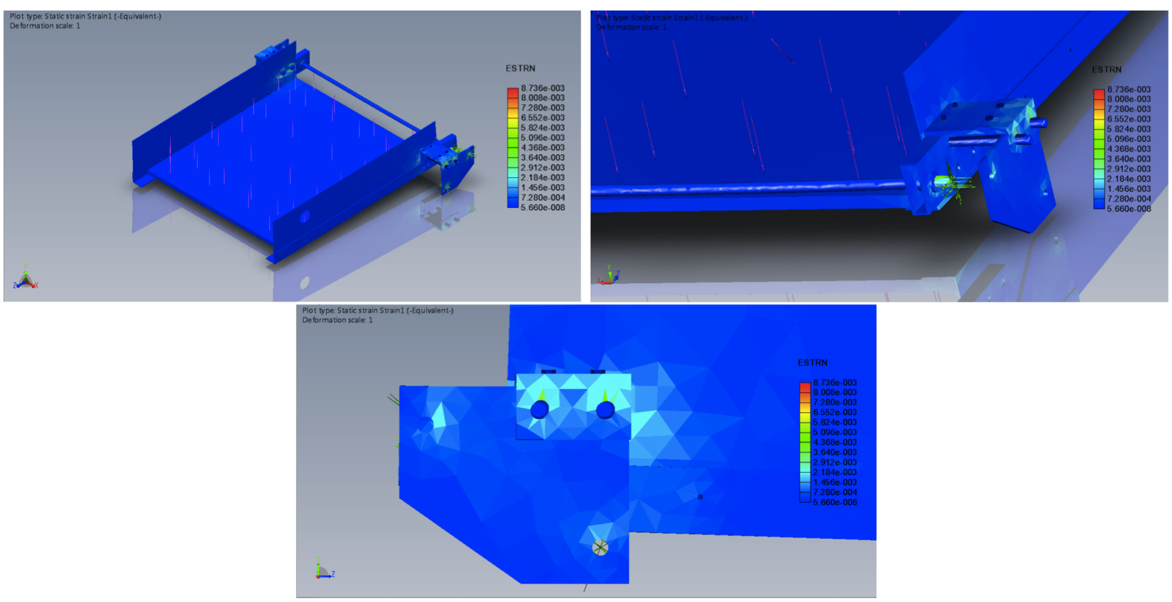

Figure 43: Image of strain results for static analyses under 300 lbs distributed loading on the platform and gravity included for self-weights.

Comparison of the model with system prototype reveals a gross overestimation of performance in the model. To speculate upon the reasons for the disparity, it is hypothesized that excess clearances are the cause for this disparity.

Should the excess deflection under self-weight be a result of material yielding or plastic strain on the components, there should be visible sign of these deformations on the parts. Upon inspection of the components following testing, no such deformation was discernible. Therefore the observed disparities are hypothesized to be mostly a result of poor quality machining.

The machining work was performed by the students who author this paper, who have had little to no prior experience working with precision machining of metal components. Thus, it is possible, and evident from various mistaken machining trials, that the Solidworks model dimensions may not precisely mirror those of the actual components despite efforts to measure exact component feature locations. Caliper and ruler measurements are subject to various subjective human errors and the true interaction between parts can be difficult to replicate in model simulations. Additionally, when the model was updated to include the actual hole dimensions, the model was not solvable in Solidworks due to instability of the assembly. The clearances caused displacements which were too large for the simulation to handle, even when run under the "large displacement" condition.

Although every hole diameter in the assembly (tapped and clearance) was determined using standard clearance charts [37] (see Appendix H), upon later evaluation with machining experts, it was determined that the chart overestimated necessary clearances (even when the close-fit standards were used). Furthermore, the majority of holes were drilled using a standard drill press since the Computer Numeric Control (CNC) machine was out of order in the University machine shop; the manual measurements and drilling likely contributed to machining inaccuracies and looser fits. It is suspected that more precise machining would improve the performance of the lift. Supporting this theory are the results of the Solidworks simulation trial run under self-weight and prototype hole sizes, wherein deflection totaled ∼0.24inch (see Appendix G). At this magnitude of deflection, a vertical hole misalignment near the railing of 0.04inches (∼1mm) could cause the observed experimental deflection under self-weight (see figure 44 and equations 18 - 19).

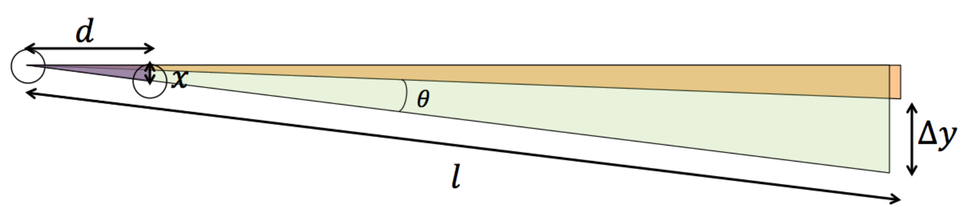

Figure 44: Diagram demonstrating how minimal vertical misalignment at the holes can cause large deflection in the system. ($x = 0.04$in., $l = 36$in., $\Delta y = 0.6975$in., $d = 2$in.)

$$\arcsin \frac{\Delta y}{l} = \theta \tag{18}$$

$$d \tan \theta = x \tag{19}$$

Assuming that proper machining is performed on the parts, subsequent static simulations with varying materials (mainly replacing aluminum with steel) were performed to gain an approximation of what changes could be made to design for future prototypes and improvements. The yield strength of stainless steel 304 (most prevalent alloy) is approximately 31,200psi, which is weaker when compared to Aluminum 6061-T6 at 45,000psi. However, the steel is much stiffer with an elastic modulus (29,000ksi) almost 3 times stiffer than that of the aluminum (10,000ksi). Although stainless steel is both heavier per unit weight and more expensive, it is thought that the additional self-weight of the material relative to the required capacity can be traded in compromise for greater stiffness. Additional costs (raw material and machining) will be necessarily incurred and accounted for.[35][36]

An example of one such study is shown below, where high stress components (rectangular tubing and side panels) were replaced with stainless steel, see figures 45 - 47.

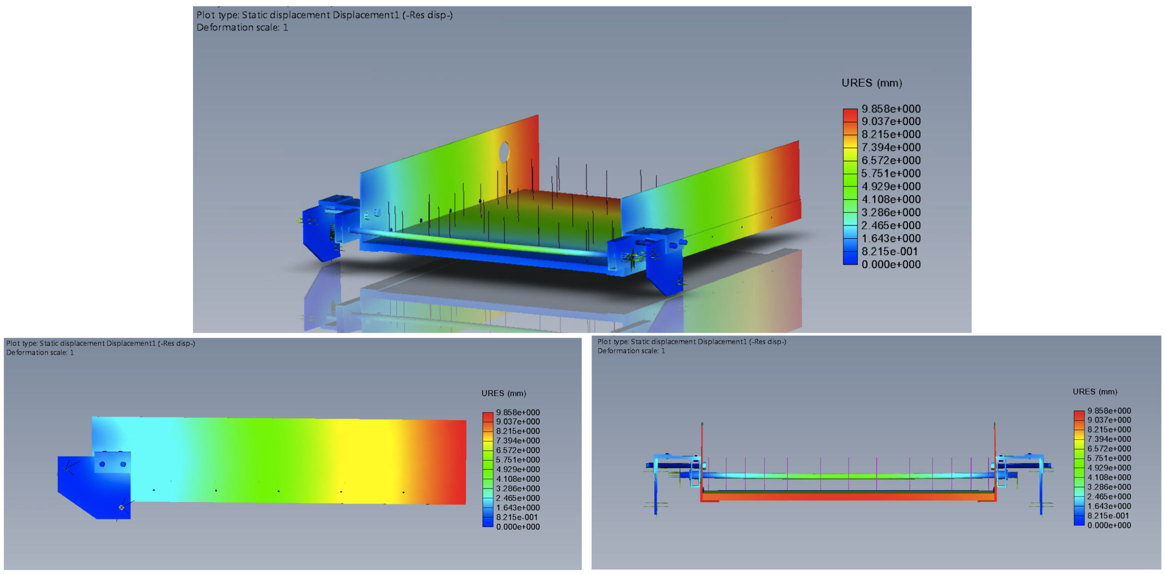

Figure 45: Image of displacement results for static analyses with steel components under 300 lbs distributed loading on the platform and gravity included for self-weights.

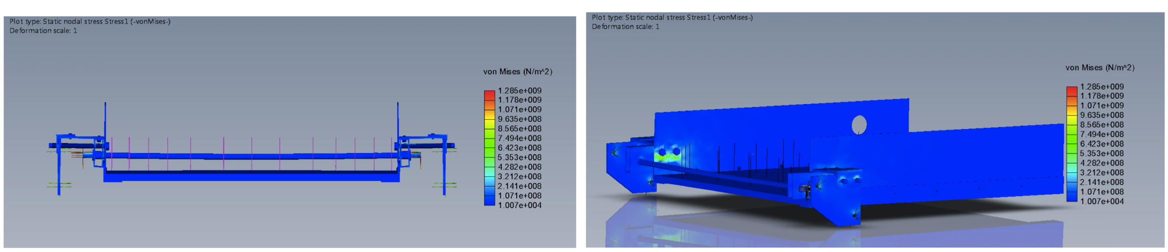

Figure 46: Image of stress results for static analyses with steel components under 300 lbs distributed loading on the platform and gravity included for self-weights.

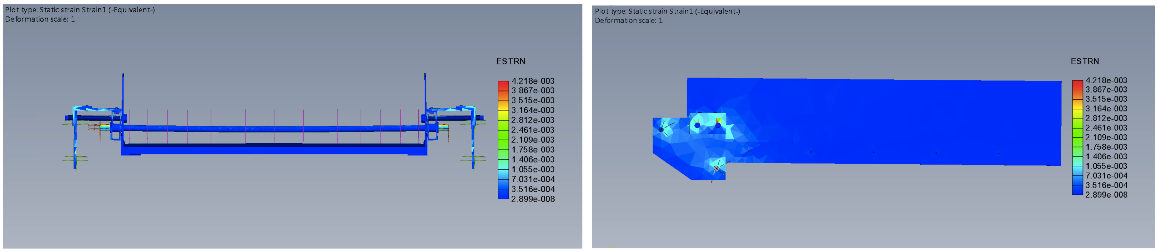

Figure 47: Image of strain results for static analyses with steel components under 300 lbs distributed loading on the platform and gravity included for self-weights.

Overall deflection in this model was approximately 0.35in (a 70% decrease in deflection and equal to a 0.56^o decline), with max strain = 0.42% and max von mises stress = 186,000psi. Although these deflections still do not comply with codes and standards for residential and manually operated lifts, it is believed that further studies and trials will allow for the development of a design that would comply. Particularly, the use of load cells could be used to determine the true applied forces on various components to obtain a better understanding of why the experimental prototype digresses from model predictions. Unfortunately cost and time constraints limit the availability of such data for the current paper.

6.3.3 Ascension Rate and Force Output

Theoretical data for ascension rate based on component dimensions are exhibited in figure 48, while figure 49 shows a comparison of theoretical and experimental values (see data in Appendix E). A linear regression performed on the data resulted in a 0.48in./push-pull cycle average rate of ascension. Compared with theoretical values, the prototype required 176% more push-pull cycles than predicted.

	Fin (lbs)	theta1	L (in)	R1 (in)	R2 (in)	R3 (in)	Fout (lbs)	theta3	Disp. (in)	Height (in)	Power (W)	pushpull cycles (n)	time (s)	Avg n	Avg time (s)
By Pitch	5	1.57	38.5	2.233	1.123	0.887	170	3.12	2.77	16	45	10.1	7.6	12.1	9.2
Diamter	5	1.57	38.5	1.598	1.123	0.887	170	2.23	1.98	16	45	14.1	10.7		

Figure 48: Table of theoretical data with prediction of ascension rate under "Avg n" for average number of push-pull cycles to travel upwards 16 inches. Time rate of ascension based on human power output assumptions and system operating force of 5lbs.

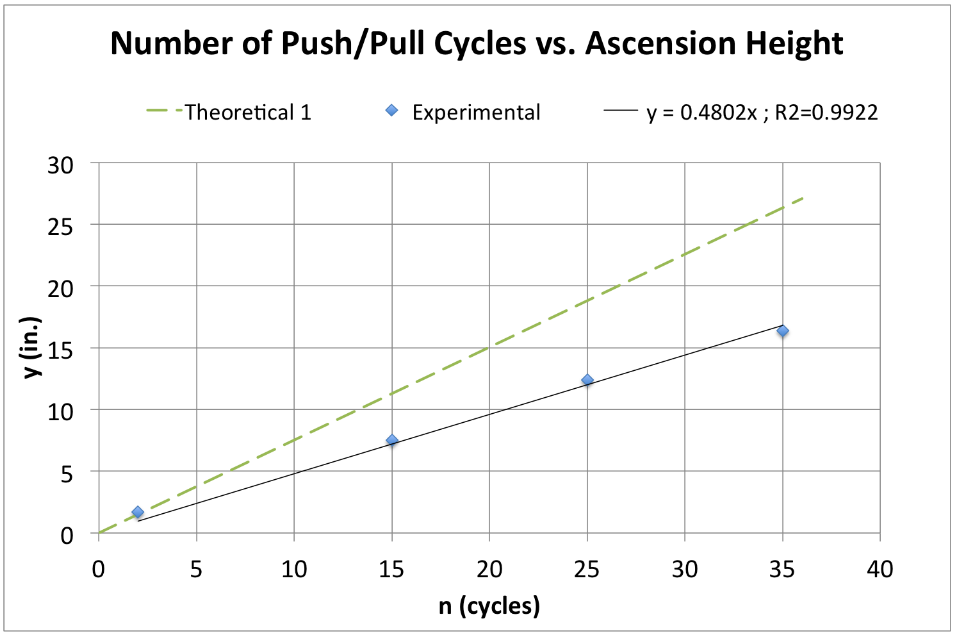

Figure 49: Graph of ascension per push-pull cycle as gathered from experimental testing compared with theoretical prediction under self-weight only (no external load). Input degrees per push-pull cycle is 60deg.

The disparity between theoretical and experimental results may be attributed to several factors. Of primary significance is the fact that freewheel sprockets tend to have a range of "lag degrees", during which no power is being transferred when switching direction of motion. This behavior is inherent to the design of typical freewheeling mechanisms where a pall(s) must catch on a toothed rotating component in order to shift from freewheeling behavior in one direction of rotation to fixed behavior in the other (see Figure 50). Consequently, each time the user switches from a pulling motion to a pushing motion, there are several degrees of rotation that are lost and which contribute to the inefficiency of the drive system. This "lag degree" was measured to be approximately 13^o. Accounting for this in the theoretical calculations (Figure 51), the difference between theoretical and experimental values is reduced to $\sim$1.7%. The predicted time-ascension rate using 5lbs input force, neglecting inefficiencies due to friction (bearings and joints well lubricated), and assuming a human power output of $\sim$45 W, is 0.59s/in. or 21.24s/yard. Comparison of new theoretical can be observed in figure 52.

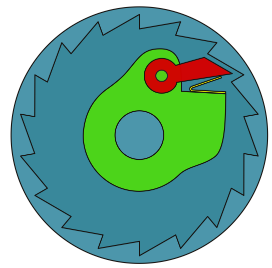

Figure 50: Illustration of simple freewheel sprocket design, with toothed rotational component on the circumference and internal pawl depicted in red. [30]

	Fin (lbs)	theta1	L (in)	R1 (in)	R2 (in)	R3 (in)	Fout (lbs)	theta3	Disp. (in)	Height (in)	Power (W)	pushpull cycles (n)	time (s)	Avg n	Avg time (s)
By Pitch	5	0.59	38.5	2.233	1.123	0.887	170	1.18	1.05	16	45	27.3	7.8	32.8	9.4
Diamter	5	0.59	38.5	1.598	1.123	0.887	170	0.84	0.75	16	45	38.2	11.0		

Figure 51: New theoretical data with prediction of ascension rate under "Avg n" for average number of push-pull cycles to travel upwards 16 inches. Time rate of ascension based on human power output assumptions and that system operates with 5lbs of force.

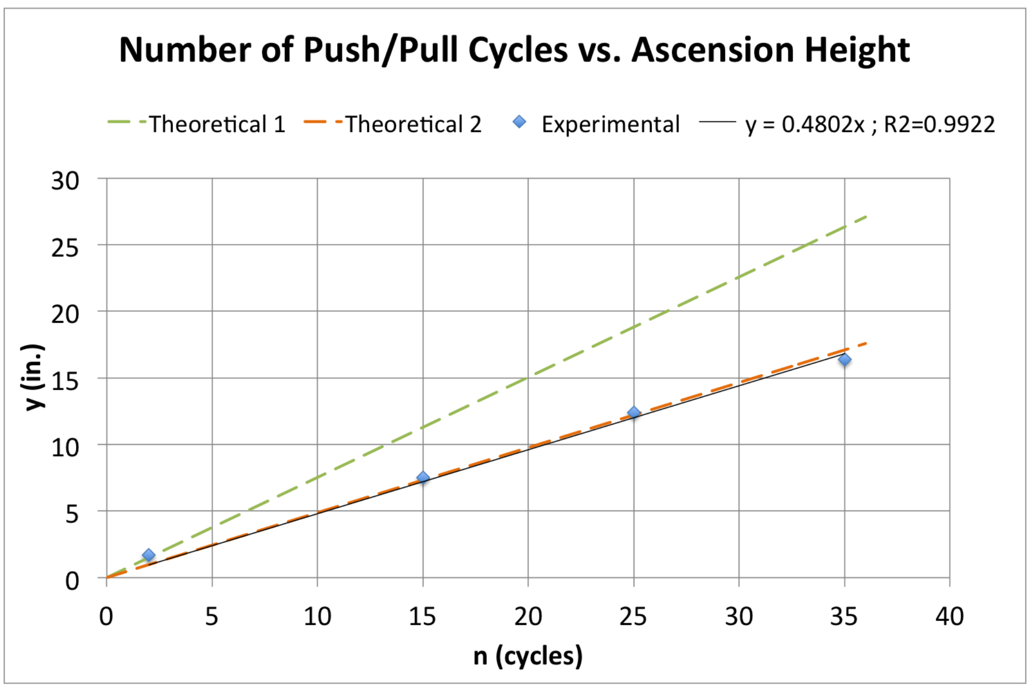

Figure 52: New theoretical ascension data compared with initial theoretical data and experimental data.

Additionally, components for the prototype were, in some cases, purchased erroneously such that sprocket dimensions diverged from the theoretical ideal. It should be recognized that sprocket ratios implemented in the prototype decreased the number of required push-pull cycles while simultaneously reducing the torque output obtained from each cycle. In summary, improvements to the prototype to operate within standards would increase the number of cycles required to ascend an equal height (see Appendix E for Torque vs. Sprocket ratio data). For example, if both input sprockets were to be 17 teeth, the reduced radius would reduce the rate of ascension to ~0.356in./push-pull cycle (approximately 25% decrease).

7 Future Work

Upon pursuing a fully functional commercial model, there are some modifications and additions that would be made. First, the descending and disengaging mechanisms shown in figures 28 and 27 respectively would have to be built and tested. More detailed drawings and prototyping of these two devices are mandatory as they allow the lift to be operated at full capacity. Aspects such as what joints to use and how to cover the disengaging mechanism's open components should be important considerations for the final build. Further amendments should include the replacement of aluminum with steel of key components mentioned earlier. Such an action is required to meet the deflection standards set forth. Lastly, as designed in figure 25, the addition of a hinged platform to act as a ramp and a foot guard to the platform needs to be added for complete passenger protection. Another ramp, similar to the one used as a foot guard, needs to be included at the rear of the platform, so that the rider is capable of entering/leaving through the back without hitting the axle. Of course, it is needless to say that a functional lift will require accurate machining. In the interest of saving money this prototype was not machined by professionals, however a final commercial product available to the public needs to follow more strict requirements.

Although not as crucial, several other changes can be made in order to improve the functionality of the system. The rate of ascent can be improved through the use of a gearing system to allow for higher force input. Modified freewheel sprockets to minimize the "lag degrees" referred to in this paper can improve the efficiency. A chain tensioner may be a necessity when testing higher weights on the platform. The addition of this to the rectangular tubing could be easily done and can prevent unnecessary risks. Finally, a hinged lever arm is needed in order to allow a passenger to comfortably mount the platform.

Further research and testing into this project may of course result in the arrival of new problems and further modifications. The writers of this report present this section as the next stepping stone into this project, although not necessarily the last one.

8 Conclusion

The use of an early stage prototype was key in illuminating all of the flaws present with the initial design. As described previously, numerous alterations were made to the physical model, and although great improvements were seen, there are still many more discrepancies to handle before the lift is fully functional. The solutions to these discrepancies, due to lack of time and funding, have been rendered to theory rather than practice. However despite any setbacks the lessons learned during the entire process allows the final design to be confidently presented as one that, if built with precise machining and improved materials/dimensions, is capable of smoothly performing its intended functions.

Looking back at the objectives initially defined, a properly built manual lift of this form should be able to meet all the requirements. These include allowing a disabled person to, without any assistance, traverse a flight of stairs safely and efficiently. Comparing the lift to electrical counterparts, the proposed design is a great deal cheaper, and at no great loss to speed [38]. When compared to other manual lifts such as the Funicular, the lift presented here has the ability to be faster, and thereby more efficient (due to the ability to do work both in the pushing and pulling movements), and comes close in total cost (excluding machine work). It should be noted that the current design is modular and easily adaptable to stairs of varying dimension (by simply adjusting the hole angles in the wheelplate and the angled cut of the rails). To reiterate, further work has to be done in order to truly confirm any of the claims made above, but the lift has been designed with strong foundations in engineering and almost all angles considered. There are high hopes for it to succeed in what it was set out to do.

9 Acknowledgments

- We would like to extend special appreciation and gratitude to Mr. Angel Morciego and staff at the University of Miami Machine shop for their counsel and instruction over the course of this study. We learned a great deal of practical knowledge and are grateful to have gained the acquaintance.
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11 Appendix