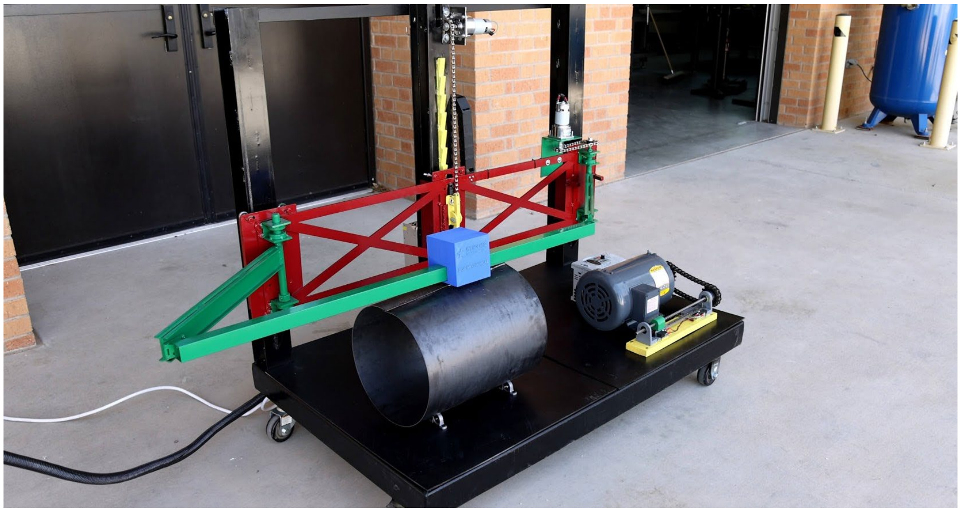
Improvements to a Welding Manipulator



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1 Abstract

As part of the new Mechanical Engineering program at Angelo State University, senior-level capstone students partnered with Wendland Manufacturing, to develop a new concept design to improve upon the existing design of a custom-built pressure vessel welder manipulator located at their production facility. Two main problems were presented by the client related to the existing design. The first problem was related to the safety of the operator and bystanders while the machine is in use. The second problem was related to the need for physical manipulation of the machine by the operator. The existing design presented multiple inefficiencies that could be addressed by developing new design solutions. These solutions were focused on adding mechanical systems to improve the raising and lowering movements as well as the rotational movements of the machine. Each design solution integrated new safety measures aimed at preventing injury to not only the operator but also the machine itself. Utilizing mechanical, structural, electrical, and industrial engineering theory, concept solutions are proposed that meet the requirements set forth by the client. Gearmotors, sprockets, roller chains were determined to be the best solution for handling the movement of the machine. Limit switches and a safety latch mechanism using shock absorbers were determined to be the best solutions to improving the safety of the machine and the operator. BGB Mechanical developed the proposed solutions with the help of Angelo State University faculty, industry professionals at Wendland Manufacturing and Indeco Industrial Electric, as well as help from other students at Angelo State University.

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2 Introduction

2.1 Client information

The client for this senior-level capstone project is Wendland Manufacturing Corporation, located in San Angelo, Texas. They have been manufacturing tanks and pressure vessels since 1921. They began with light gauge fabrications but have expanded to manufacture tanks for many different applications. These include hydropneumatic tanks used in municipal water districts to stainless steel tanks used in the distillery industry. This expansion in manufacturing was made possible by adopting the latest in welding techniques and incorporating new machinery into their production line. One of the newest additions to their production line is a custom-built piece of equipment called a welding manipulator that supports a sub-arc weld head, enabling it to be precisely positioned over a tank for welding. This equipment gives the welders the freedom to relax while the machine does the work. It allows for greater stability of the sub-arc weld head during welding procedures, improved weld quality, and increased production output.

2.2 Welding Manipulator Introduction

Wendland Manufacturing developed a custom welding manipulator for sub-arc welding circumferential and longitudinal seams on tanks and pressure vessels as seen in Figure 1. This piece of equipment was custom designed and fabricated by an employee of Wendland. The existing machine is attached to two I-beam columns approximately 24 ft apart. Uniquely fabricated sleds are attached to the machine to ride vertically along the front column flange to a max height of 20 ft measured from the top of the carriage, allowing for welding between 6 ft and 14 ft diameter tanks and pressure vessels.

Figure 1: Existing welding manipulator at Wendland Manufacturing

Each of the sleds contains multiple bearings which secure the machine to the columns, reducing friction between the sleds and columns during lifting operations as seen in Figure 2. The rectangular carriage is the main component that connects the entire machine and is attached at the ends of each of the sleds. Also, at each end of the carriage are attached jibs that span 14 feet from the carriage and can rotate in either direction. The ends of the jibs are connected by rectangular steel tubing used as a track to position the sub-arc weld head. The jibs' ability to rotate is primarily used to position the weld head above a tank during welding procedures but also enables the machine to be stored against the wall when not in use. Storing the machine close against the wall allows for the overhead gantry crane system to bring large tanks in and out of the building.

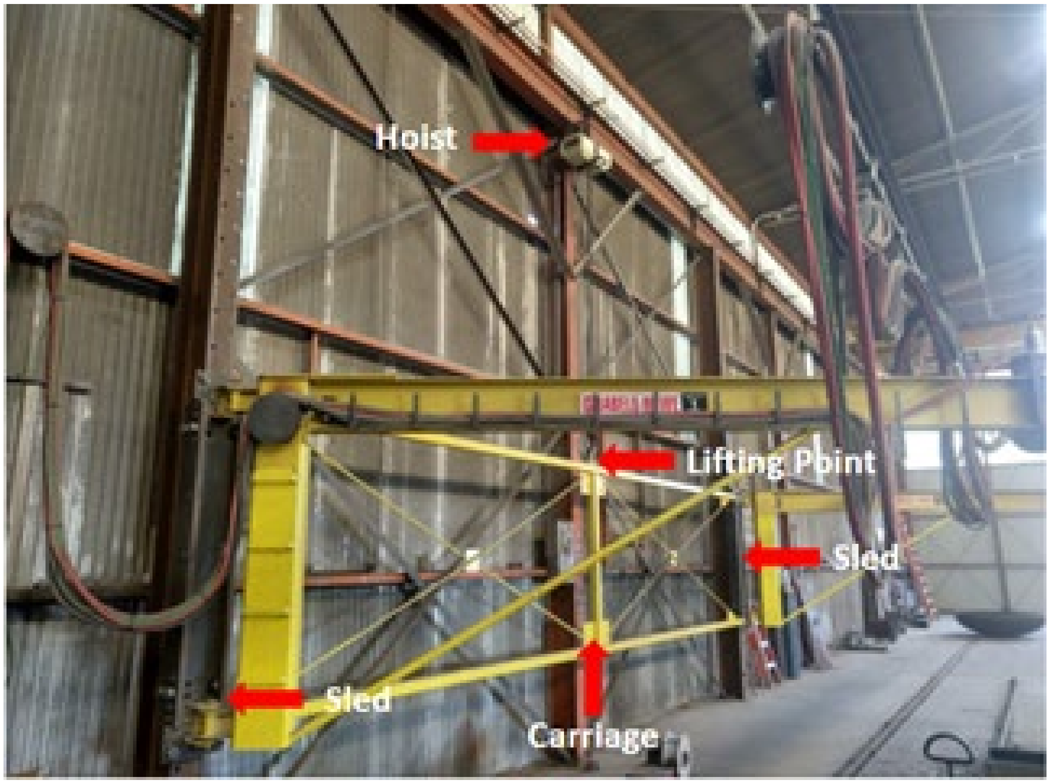


Figure 2: Machine Components

2.3 Problem Statement

The welding manipulator is designed to make the process of welding tanks and pressure vessels more efficient, but several problems exist that are slowing down operations and creating an unsafe working environment. The first problem is to move the machine in and out of the storage position, an employee must manually grab the machine from the end of the jibs and pull it using their strength until it is positioned for welding. This presents a safety issue to the operator because they are positioned beneath the machine during this task. The second problem is that the current lifting system has insufficient measures in place that would ensure the safety of the machine and the operator in the event of chain or hoist failure. The current safety measure requires the operator to insert a safety pin into predrilled holes of each column below the machine after it is positioned into place for welding. These pins are designed to prevent the machine from falling to the floor. But, while inserting the pins, the operator is once again beneath the machine which presents a safety hazard.

The human costs alone are reason enough for these improvements, but the financial costs that would arise in the event somebody is harmed using the machine are also the reason for the change. Improvements could also have the potential to increase productivity and decrease maintenance costs over the long run. If motors and remote controls were integrated into the existing system, employees would be able to safely operate the machine without the need to put themselves into hazardous positions. Employees may also see a decrease in the time it takes to finish a tank or pressure vessel when using a remote control to position the sub-arc welder over the spots to be welded.

2.4 Scope

The purpose of this project is to present engineered concept solutions to Wendland Manufacturing that would improve the safety and operation of the current welding manipulator machine. BGB Mechanical will develop improvement solutions for the machine that will include the addition of new mechanical systems for both the lifting and jib rotation operations according to industry standards and the design of safety mechanisms that eliminate the need for operator involvement according to industry standards. With these improvements, Wendland Manufacturing employees should benefit from the added convenience and safety of this piece of equipment, increasing productivity and job satisfaction.

2.4.1 *Client Requirements*

The client has provided a list of requirements that must be included in the proposed design. These requirements are as follows:

1. The system must be designed so that it is easily operated and requires very little operator training.
2. Maintenance on the system can be performed by current maintenance employees.
3. A variable frequency drive (VFD) must be used to control the machine's movements.
4. Components must be compatible for use with 240 Volt, 50 ampere, 3 phase power.
5. The machine must be able to be controlled from two separate locations, non-simultaneously (from the floor in the vicinity of the machine and from atop the tank where a welder operates the sub-arc welder).
6. Engineered solutions for operator and machine safety during lifting and rotational movement must be incorporated.

2.4.2 *Objectives*

The objectives of the welding manipulator improvement project to be accomplished by BGB Mechanical are as follows:

1. to developed concept solutions for lifting operation improvements that include a new mechanical system to replace the existing hoist and chain.
2. to develop concept solutions for the jib rotation movement that includes a new mechanical system to replace manual movement by the operator.
3. to develop concept solutions for a safety mechanism that will replace the existing safety pins used in the machine columns.

4. to design a remote-control system that will control both the lifting operations and the jib rotation movement at two separate locations.
5. to incorporate the client requirements into the design of all improvements to the machine.
6. to propose possible future improvements that could be made to the machine.

2.4.3 ***Deliverables***

The deliverables for the welding manipulator improvement project for Wendland Manufacturing provided by BGB Mechanical are as follows:

1. A complete design concept for proposed improvements to the existing machine that include the new mechanical systems for lifting and rotating the machine, along with the new safety mechanism design.
2. A detailed report of the concept design to include calculations and equipment recommendations.
3. A non-scaled prototype of the design to demonstrate core design concepts and their functionality.

2.4.4 ***Milestones***

- August 2020 – Project idea introduction to group
- December 2020 – Design improvement options presented to client and faculty for input.
- January 2021 – Prototype design and construction begins.
- February 2021 – Finalized design improvements and prototype progress presented to faculty for review.
- May 2021 – Final presentation of the project to client, faculty, and IAC members, including a completed prototype demonstration.

2.4.5 *Limits and Exclusions*

The following is a list of what is not included in the improvement project:

1. Wendland Manufacturing is responsible for primary electrical wiring to supply power to the system.
2. Wendland is responsible for foundation inspections to account for the added weight of the machine.
3. BGB Mechanical will not be responsible with creating or providing CAD drawings to Wendland.

3 Machine Lifting Assembly Improvements

In this section, the lifting capabilities of the existing machine are analyzed, and then suggested improvements to the machine are discussed. First, the existing conditions of the lift assembly are discussed along with safety concerns that need to be addressed with the proposed design. Next, design boundary conditions are presented that will allow for a basis of design. Then, several design options are presented and discussed. These will be evaluated based on client needs and the applicability of each to determine the best design solution for improving the lift capability of the machine. Next, calculations are performed to determine component sizing. Finally, a conclusion reporting the final design decision and justifications are presented.

3.1 Existing Conditions

The existing assembly utilizes a hoist lifting mechanism that uses a chain and hoist to lift the carriage from a fixed point, referred to as the lifting point, as shown in Figure 3 below.

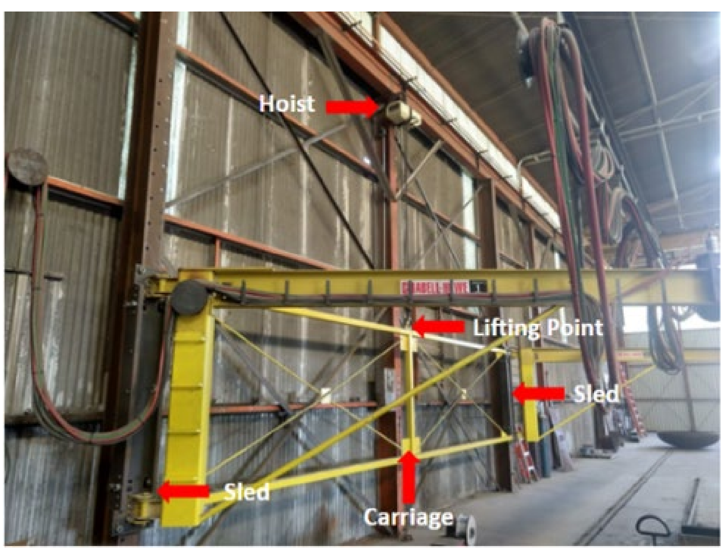


Figure 3: Current Machine Hoist and other components

The machine utilizes a frame (carriage) that is attached to two sleds, that roll vertically up and down the columns during vertical movement of the machine, as described in section 2.2. The 12 bearings installed on each of the sleds, contact the front, back, and sides of the column flange. The bearing placement can be seen in Figure 4 below.

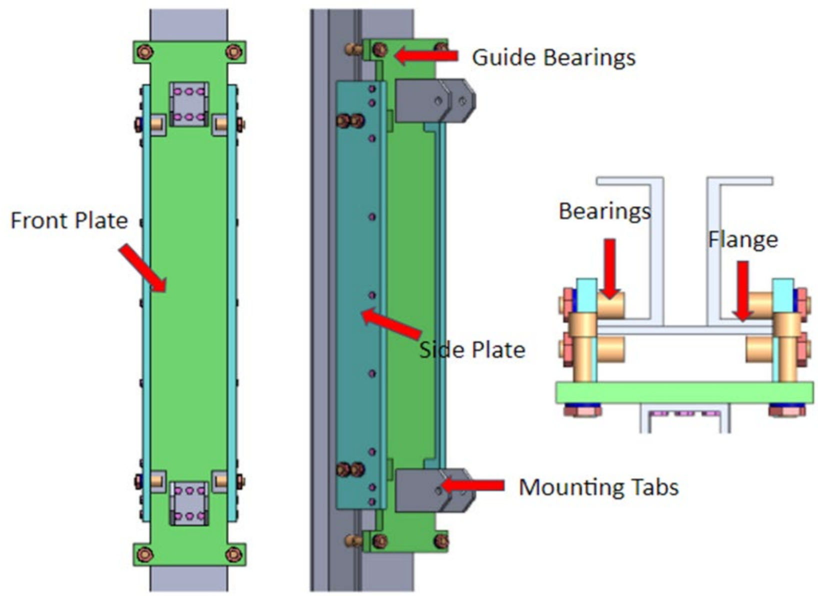


Figure 4: Current Sled Design

There are four guide bearings attached to the front plate of each sled and four more bearings attached to each side plate of the sled. The guide bearings help the sled to stay centered on the column. The bearings on the side plate serve to keep the sled against the column, sandwiching the flange between a pair of two bearings. All these bearings help to keep the entire frame aligned parallel with the floor of the building. Depending on the placement of the weld head on the weld track of the jib assembly while being lifted, the carriage assembly frequently binds and shakes during the lifting movement. This binding appears to influence the lifting speed and has become an area of concern related to the hoist capability and safety of the operator. This binding could be assumed to introduce uneven wear on the bearings. The current hoist is rated to lift 2 tons, however, if either the hoist fails or the chain breaks, the entire machine could fall to the floor. The only existing safety mechanism to prevent this are safety pins inserted into the columns designed to catch the carriage during such a failure.

3.2 Design Boundary Conditions

The design boundary conditions for the lifting movement of the machine are used to determine a basis for calculations and improvements. The first design constraint is the weight of the machine. This value will be used in determining the sizing of almost all components in the proposed improvements. The second design constraint is the speed at which the machine moves vertically up and down the columns. The third design constraint is that the machine must be able to safely move into operating position, ensuring that in the event of failure, the machine will not injure operators. To address these constraints, a weight analysis containing the weights of all the current components are reported, current lifting speeds are assessed, and safety measures introduced.

3.2.1 *Weight Analysis*

To begin performing any calculations, the weights of each of the components of the machine are needed. These weights are shown in Table 1 below.

Table 1: Weights of Individual Components of the Machine

Component	**Weight (lb.)**
Carriage	545
Sled Bearing Plates	765
Weld Track for welder	510
Welder (not shown)	318
Two Jibs	1490

Summing these weights together results in a final weight of 3628 lbs. To account for components added due to our analysis, existing and new hoses, and miscellaneous fasteners on the machine, the total weight used in all calculations was increased to 5000 lbs. This increase in weight will also serve as a type of design safety factor, in addition to any safety factors to be mentioned later.

3.2.2 *Lifting Speed Analysis*

To determine the speed of the carriage that is being lifted by the current hoist system, a video was taken of the lift operations. When observing the machine's lifting operation, the lifting point on top of the carriage was tracked to determine how far that point travels vertically up the columns. This determines the minimum and maximum height of operation, which are 5ft and 15ft, respectively. To determine the amount of time needed to travel this distance of 10 feet, the video was analyzed. The total time that it took to raise from 5ft to 15ft (10 feet or 120 inches), was 21 seconds. The average velocity during this time was determined to be 120 inches per 21 seconds or approximately 6 inches per second. The lifting direction can be seen in Figure 5 below. This information will be used in future calculations.

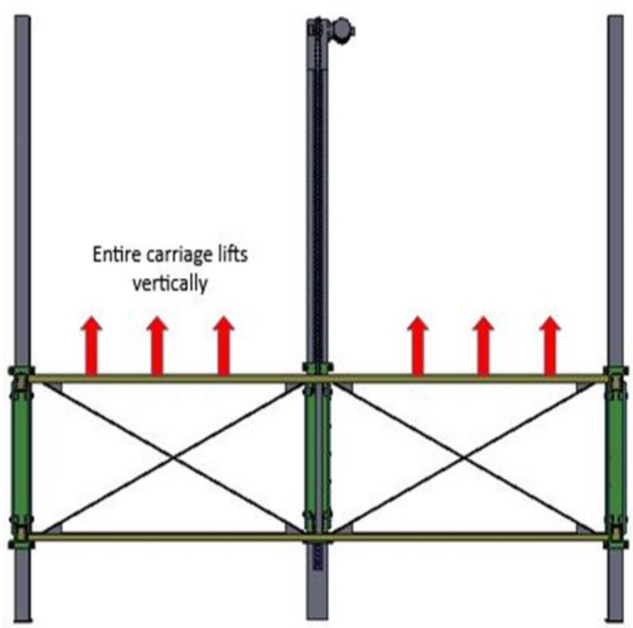


Figure 5: Carriage lift direction

The movement of the carriage is an important aspect to analyze when it comes to acceleration and deceleration of the carriage during travel as well. Piecewise functions are used to plan out the movement of the carriage where S is the position of the lifting point of the carriage in feet, V is the velocity in feet per second, and A is the acceleration in feet per square second. The first piecewise function determines the positioning of the carriage with respect to the lifting point using Eq. (1) between 0 and 1 second, Eq. (2) for 1 second to 20 seconds, and then Eq. (3) for 20 seconds to the final time of 21 seconds. The second set of equations determines the velocity of the carriage using (4) between 0 and 1 second, Eq. (5) for 1 second to 20 seconds, and then Eq. (6) for 20 seconds to the final time of 21 seconds. The final set of equations determines the acceleration of the carriage between those specified times.

Position:

$$S = \{\ 0 < t < 1 : +0.25t^2\ \} \tag{1}$$

$$S = \{\ 1 < t < 20 : -0.25 + 0.5t\ \} \tag{2}$$

$$S = \{\ 20 < t < 21 : -100.25 + 10.5t - 0.25t^2\ \} \tag{3}$$

Velocity:

$$V = \{\ 0 < t < 1 : 0.5\ t\ \} \tag{4}$$

$$V = \{\ 1 < t < 20 : 0.5\ \} \tag{5}$$

$$V = \{\ 20 < t < 21 : 10.5 - 0.5\ t\ \} \tag{6}$$

Acceleration:

$$A = \{\ 0 < t < 1 : 0.5\ \} \tag{7}$$

$$A = \{\ 1 < t < 20 : 0.0\ \} \tag{8}$$

$$A = \{\ 20 < t < 21 : -0.5\ \} \tag{9}$$

The red dashed line, as seen in Figure 6, shows the position of the carriage at any time from 0 to 21 seconds. The green solid line, as seen in Figure 7, shows the velocity of the machine at any time between 0 and 21 seconds. The black dashed line, as seen in Figure 8, shows the acceleration of the machine between 0 and 21 seconds. It is important to note that from 0 to 1 second, the machine experiences acceleration, and from 20 to 21 seconds, the machine experiences deceleration. From 1 second to 20 seconds there is no acceleration because there is a constant velocity of 6 inches per second.

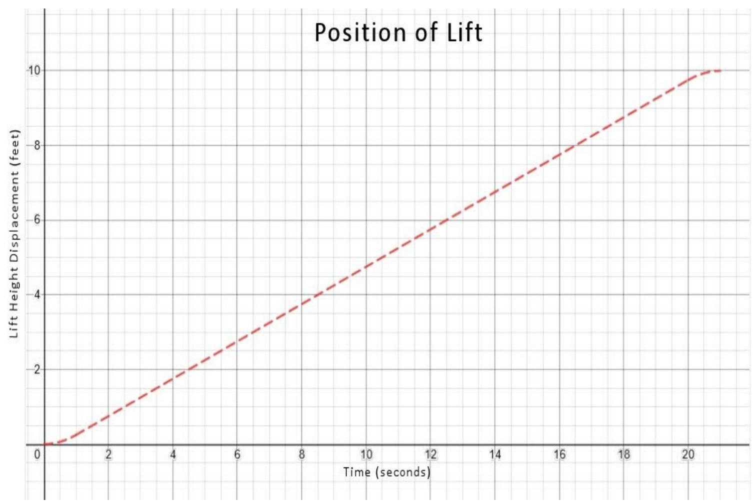


Figure 6: Position of lift

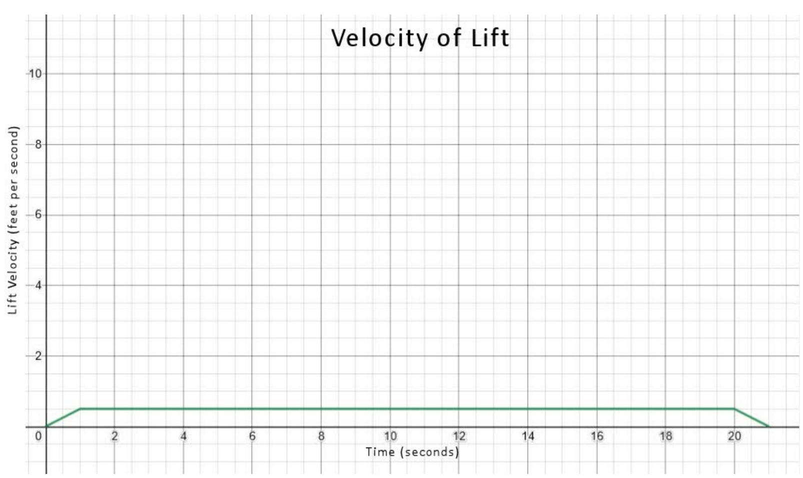


Figure 7: Velocity of Lift

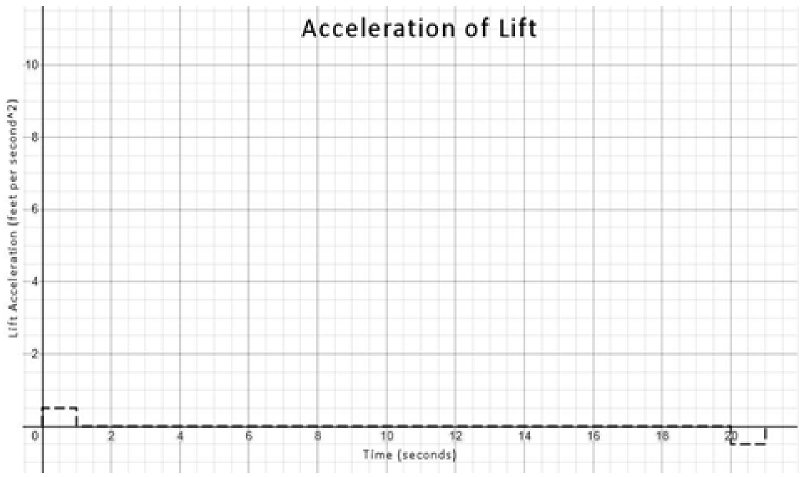


Figure 8: Acceleration of lift

Determining the acceleration and deceleration of the machine is important to prevent what is called "jerk". When moving equipment from no acceleration to some constant acceleration or deceleration value, jerk can occur, which can introduce harsh movements and impacts into a machine and its components. Sudden changes in acceleration are quantified by high magnitudes of jerk. Thus, reducing the magnitude and maintaining a continuous jerk versus time curve has advantages on machine loading. This occurs at the transition point where constant velocity is achieved, shown in Figure 9, as the flat line of the trapezoidal profile.

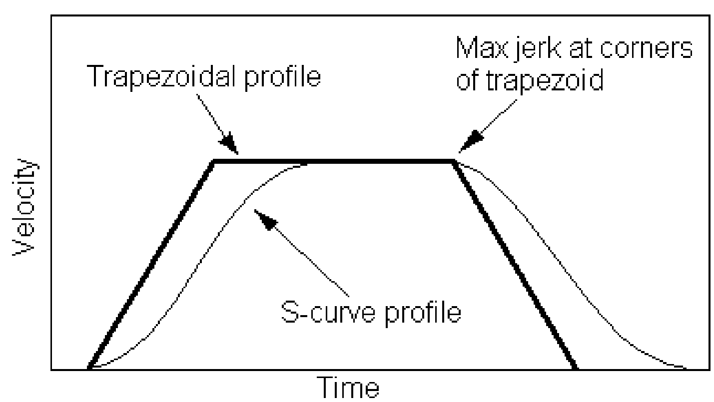


Figure 9: Trapezoidal Velocity Profile

To alleviate this, the velocity profile can be modified from the current trapezoidal profile, into an S-curve profile, which eases the machine in and out of the acceleration values chosen. Trapezoidal velocity is not ideal in these applications involving motors. Sharp changes in acceleration cause hard starts and stops, leading to vibrations or jerking movements. The S-curve profile is more applicable to real-world examples involving motors. This is accomplished by utilizing a motor drive that will be discussed in further detail in section 5.2.2.

3.2.3 *Safety*

The current machine has several safety concerns which need to be addressed through the design proposal. It is necessary to include features to the system which reduce the risk of injury to the operators and catastrophic damage to the machine. The current safety mechanism used on this machine requires the manual insertion of safety locking pins below the carriage assembly. These pins serve as a failsafe if the hoist chain were to break, preventing the machine from crashing to the floor. To insert the pins, the operator must stand beneath the machine. This could potentially be a dangerous situation if the hoist chain were to break. A replacement for the manually inserted pins is desired to eliminate the need for manually inserting and removing locking pins when the carriage is being moved to its operating position. Safety measures for this system will be considered into the different design options.

3.3 Design Options

3.3.1 *Lifting System Design*

To accommodate the client's request and design constraints of the machine, it was necessary to determine some different design options. The first design option analyzed used a winch and pulley system shown in Figure 10 and utilizing the 2-ton hoist that is currently being used to distribute the weight of the carriage evenly to reduce binding and shaking experienced in the current lift. The design creates a low complexity design which uses minimal moving parts to create a 2 to 1 mechanical advantage. This means the force required to pull the carriage vertically, would be half as much force as the weight of the carriage.

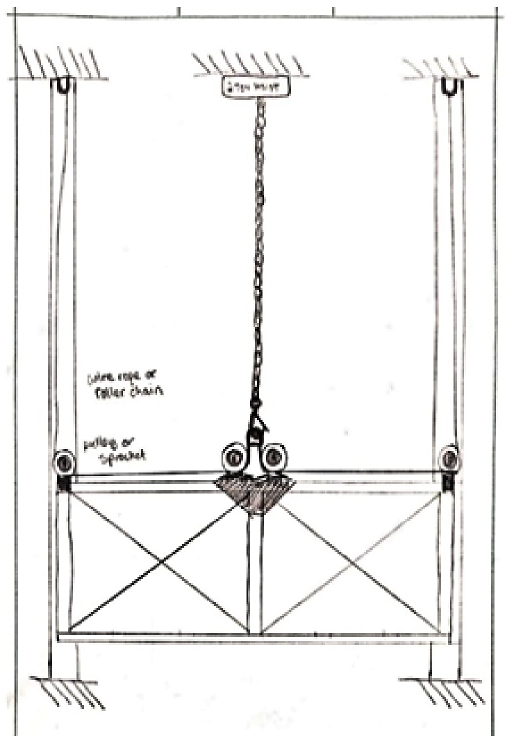


Figure 10: Option 1-Mechanical advantage pulley system

The second design option analyzed is a dual hoist system, which uses two hoists located on each column as seen Figure 11. The design evenly distributes the loads to each column allowing it to have a higher lift capacity. However, due to the complexity of a dual hoist system, a motor speed synchronization issue may occur causing uneven lifts creating more binding and shaking issues than currently experienced.

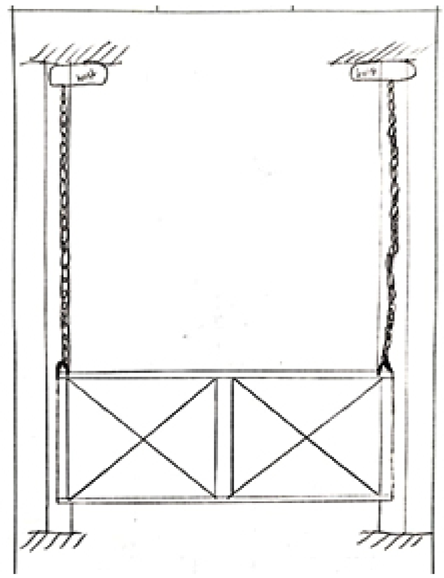


Figure 11: Option 2 - Dual hoist system

Exploring the third design option of a hoist, chain, and sprocket combination as seen in Figure 12, moves the hoist over one of the columns and incorporates a chain and sprocket system to distribute an even load to the opposite column. This design also has a high lifting capacity, but a high complexity due to the number of moving parts. It utilizes one single hoist connected to a system of chains and sprockets similar to a pulley vehicle lift currently used by mechanics.

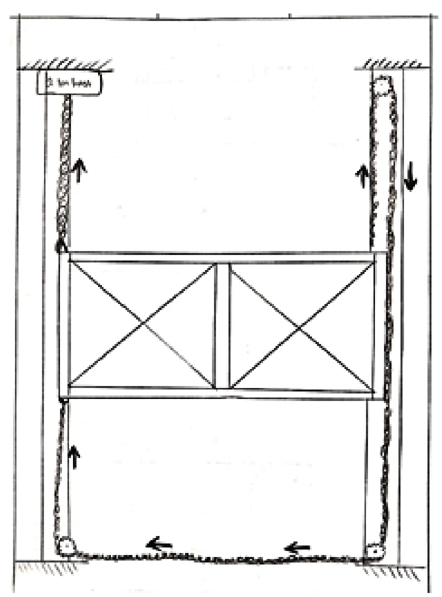


Figure 12: Option 3 - Hoist, chain, and sprocket combination.

The fourth design option is a variation of the previous hoist, chain, and sprocket combination but would incorporate a hydraulic piston that is connected to the bottom of the carriage on one side and the chains and sprocket setup is used to pull the opposite side of the carriage vertical and evenly. In this design, a hydraulic piston would need to be installed into the building. The design, however, was confirmed by the client, to not be desirable considering the implementation of hydraulics that would need to be made. An example of this design can be seen in Figure 13 below.

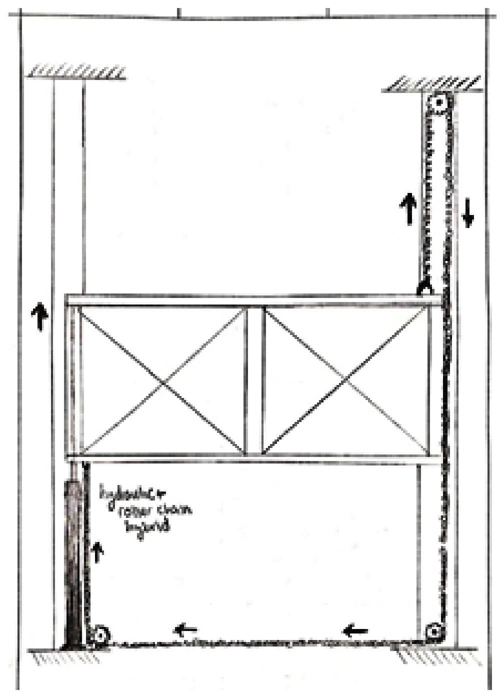


Figure 13: Option 4 - Hydraulic piston with a chain and sprocket system

The final design option introduces a third column installed directly center of the machine assembly and incorporates a mounted motor that uses a system of sprockets and roller chains to lift the carriage. An example of this design can be seen in Figure 14 below.

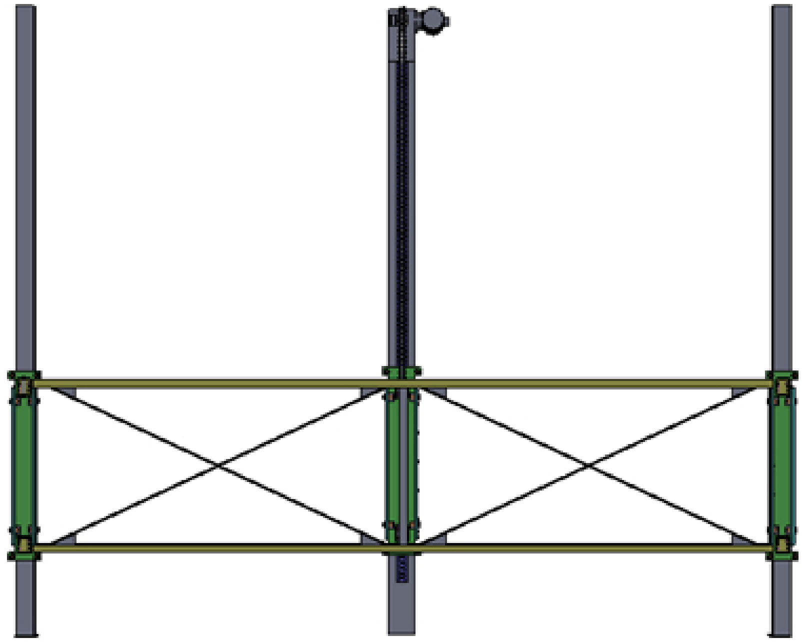


Figure 14: Option 5 - Motor, chain, and sprocket combination

In addition to adding a third column to this design, a third sled which will be like the two outer sleds will be attach to the carriage. This design would minimize binding and shaking of the machine because of the added stability from the third column and sled.

From discussions of the design options with the client, based on client requirements, it was determined that extensive use of chains and pulleys were not preferred, due to the amount of dust and dirt present in the shop. It was also discussed that the use of the motor would be ideal instead of utilizing a chain hoist. Using a motor instead of the hoist will also allow for a variable frequency drive to be installed to control the speed of the movement of the machine as requested by the client. Also, incorporating a sprocket and roller chain system would provide beneficial stability due to the rigidness of the chain and alignment of sprockets. After analyzing the five different design options and researching all possibilities that fit within the client's scope for the project, option 5, using the motor, chain, and sprocket design with the incorporation of a third column was chosen to use for this application.

3.3.2 ***Column Design***

After the lifting system design for the machine had been determined, the next step in the process was to design the third column that would be capable of assisting in supporting the weight of the machine, in combination with the existing outer columns. It was determined that in order to accommodate a new motor, chain and sprocket assembly with a continuous looped chain, the center column would need to be a built-up column, that was open on the back side to allow for chain movement, as seen in Figure 15.

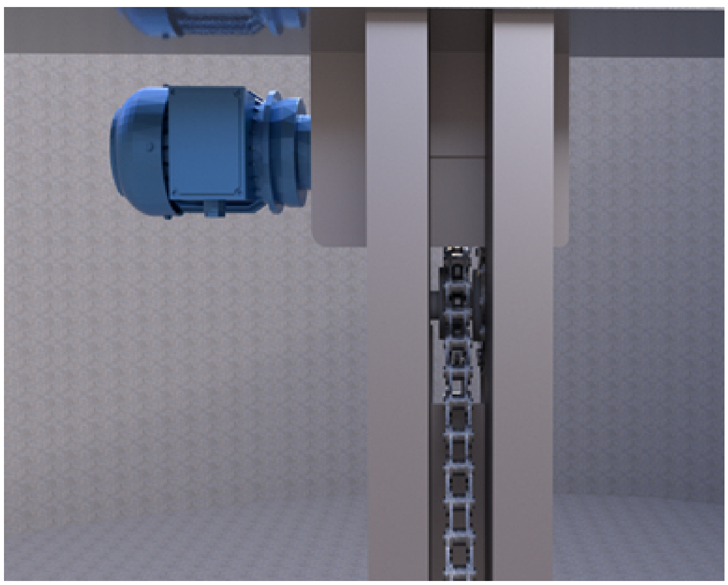


Figure 15: Backside of center column with chain located within

The built-up column design would consist of 2 flanged 10 x 30 c-channel sections placed back-to-back with a gap between them to allow for the chain to drop behind the machine. The column would be welded to an anchor bolt plate on the floor and welded to an overhead beam on the building's ceiling. The two c-channels would be reinforced with a front plate that would span the entire length of the column. Along the back side of the column, stiffener plates would be welded intermittently to the c-channels, providing support to the column to prevent twisting. The cross section of the column can be seen below in Figure 16. A continuous ¼" fillet weld would span the entire length of the column, merging the front plate and the c-channels together. The column would also be tied into the existing column's diagonal bracing to add additional support.

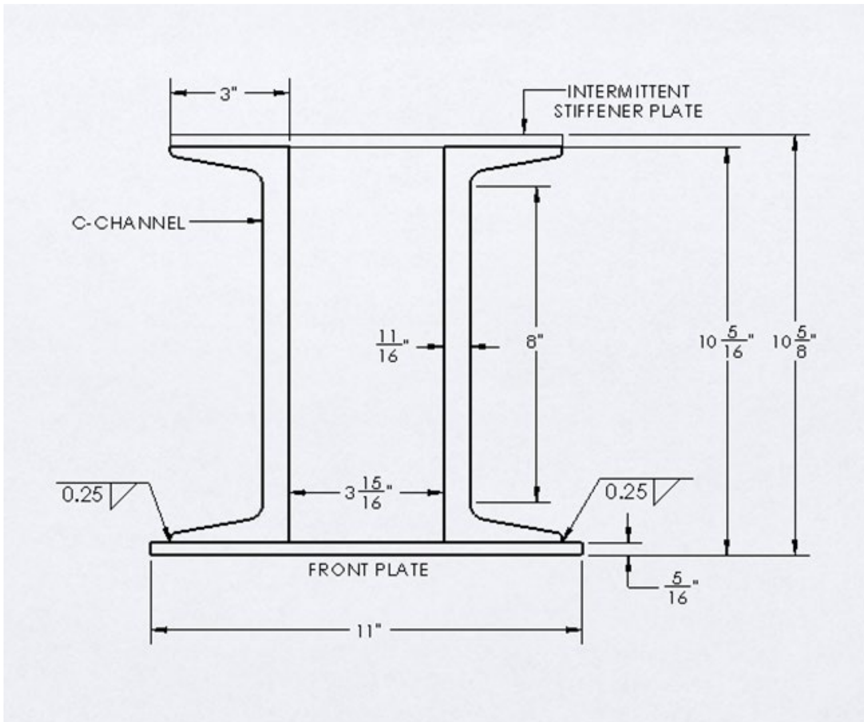


Figure 16 : CAD model and dimensions of the built-up column

3.3.3 *Chain Drive Assembly and Platform*

When considering the addition of a third column as previously described, mounting the components to hoist the machine into the air, must be considered. The selection of components and how they mount will vary the orientation of the assembly, however, a general design option can still be considered and is described as follows. At the highest point, and on the front plate of the center column, a motor platform will be installed that will support the lift shaft assembly that is connected to the motor. The platform will be welded to the front of the column near the top, in a position that will allow for the motor to drive a separate shaft that will serve to support the weight of the machine. The motor brake, as seen in red, will be discussed later in this section. This setup can be seen in Figure 17.

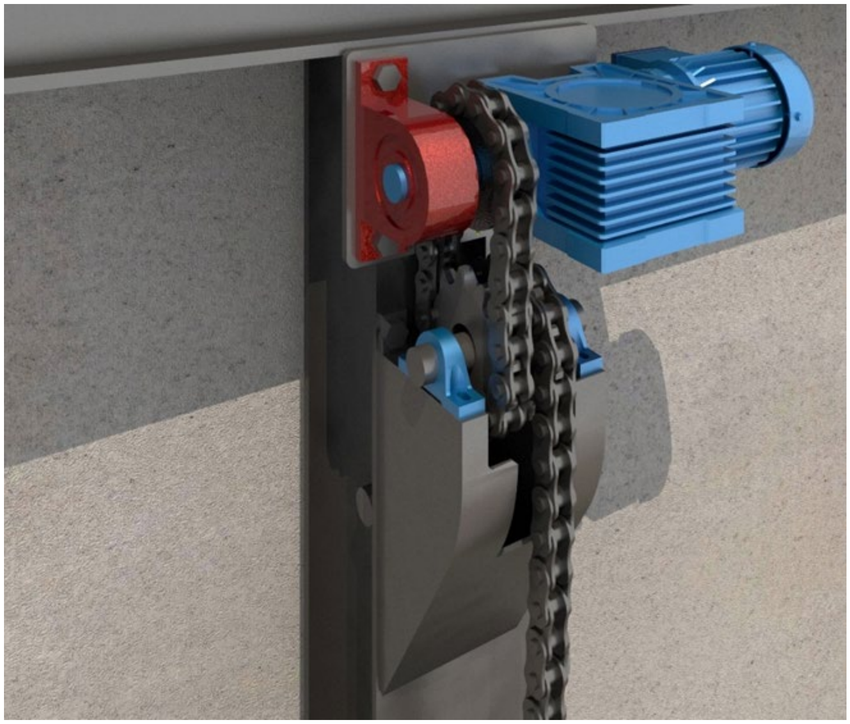


Figure 17: Lifting Shaft Assembly Platform

On top of the platform will be two pillow block bearings, in blue, that are bolted to the top surface of the platform [16]. A driven shaft with a sprocket will be positioned into the pillow block bearings and then connected by a chain to the driving motor shaft sprocket. This setup will serve to reduce the overhung load on the motor gearbox, extending the life of the motor and gearbox. In addition to the extended life of the gearbox, the separate drive shaft allows for clearance issues to be addressed where the roller chain might rub against the center column, by extending the lift point out, and away from the columns. This will allow more room for the 6 in sprockets to be installed without being interfered with by the columns. This platform can be scaled in size to accommodate different size sprockets and roller chains, as needed.

3.3.4 ***Safety Latch Design***

The current safety mechanism for the machine to prevent it from falling to the floor in the event of a chain failure, involves operators manually inserting pins into pre-drilled holes located on the fronts of the outer columns seen in Figure 18.

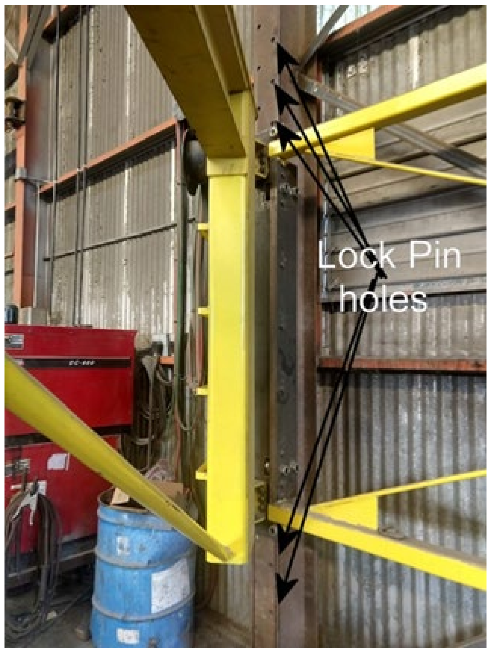


Figure 18: Locking Pin Holes in the Column

These pins consist of a bolt with a nut that when inserted into the column would not allow the carriage to fall past those points of insertion. This method is robust, in a sense that the physical connection, made by hand, can reasonably be determined to be sufficient to withstand the energy from a falling weld head carriage. The problem with this safety measure, however, is that it can put the operator in a dangerous position beneath the machine while inserting the pins. To eliminate this method, several alternative safety mechanisms were designed.

There were three main design options that were considered. The first design option was to use a solenoid with a latch and sawtooth system, as seen in Figure 19.

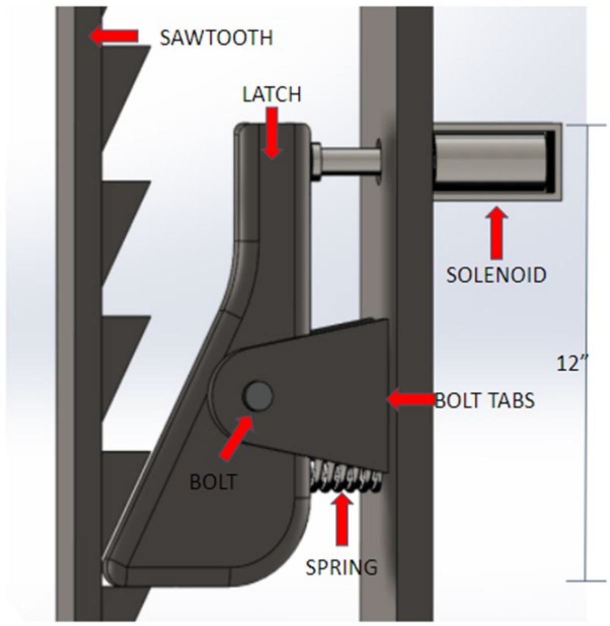


Figure 19: Option 1-Solenoid Safety latch

The second design used a tension cable method with a latch and sawtooth system, as seen in Figure 20. This design utilizes a pair of synchronous latches that are engaged upon release of tension in the chain.

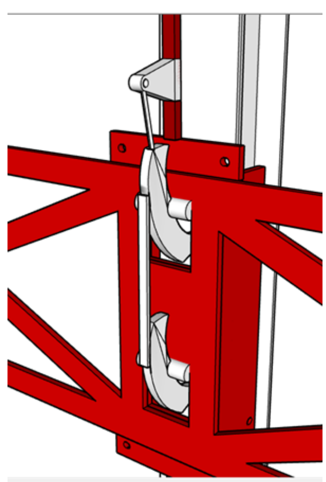


Figure 20: Option 2-Tension cable, double latch safety mechanism.

The third design option utilizes a latch and sawtooth design, like option 1, but without a solenoid to disengage the latch as seen in Figure 21. Rather than utilizing a solenoid to disengage or engage the latch, the force from the weight of the carriage combined with positioning of the latch about a point of fixed rotation is used. This will accomplish an instantaneous safety latch engagement in the event of failure from the lift system.

Figure 21: Option 3: Cad model of weight driven safety latch

All of these designs, in part, are based on safety latches that have been used on other smaller weld head manipulators or lifting mechanisms such as vehicle lifts. An example of a smaller welding manipulator can be seen in Figure 22 below.

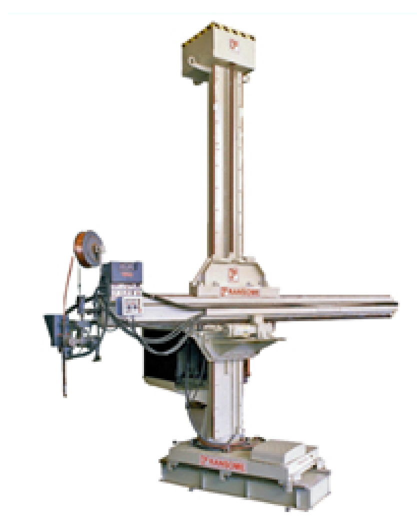


Figure 22: Small welding manipulator [21]

The safety mechanism on this type of machine, which can be seen in Figure 23, utilizes a safety latch and sawtooth track. The safety latch is connected to the chain that lifts the machine vertically up the column. This particular design has tension in the chain, keeping the safety latch disengaged at all times, unless chain tension is removed, allowing the latch to engage and rest on the top of the nearest sawtooth track section. The latch locks into place, preventing the machine from lowering. The latch is disengaged when tension is applied to the chain, which occurs during lifting, or lowering operations, assuming the rate of lowering does not exceed accelerations consistent with an object in freefall. This safety mechanism is common in these types of machines and heavily influenced the third design option of the safety latch system.

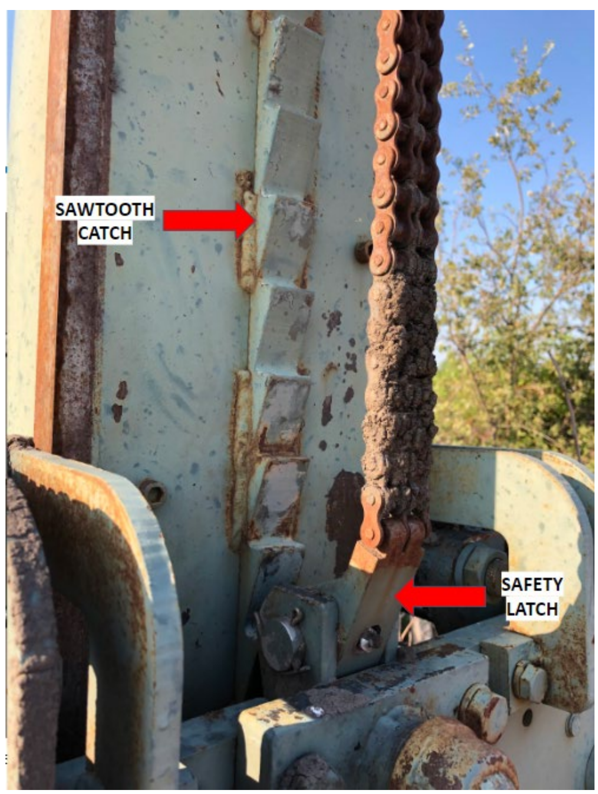


Figure 23: Weld manipulator safety latch example.

The first design option considered was based on the same idea as the small weld manipulator, but the addition of a solenoid added complexity and more points of failure that could be avoided with other options. By connecting the chain to a separate lifting point on the carriage, the safety latch is only engaged or disengaged utilizing an external electrical energy source through something such as a solenoid. A solenoid is an electrical device that uses a solenoid plunger or actuator, that is activated by an electrical current. When the solenoid is activated, a spring mounted behind the latch would compress, keeping the safety latch dis-engaged. When power is removed, or is non-existent, the spring force would cause the safety latch to engage with the sawtooth track. The latch would be in constant contact with the sawtooth plate when the solenoid is not engaged, preventing it from being lowered. Once the solenoid is activated, the plunger would push the top of the latch forward causing the spring to compress and disengage from the sawtooth plate, allowing the machine to be lowered. This design is intended to keep the safety latch in constant engagement as it is lifted or not in use and can only be lowered when the solenoid is engaged. A downfall, however, is that to move the carriage downward, the carriage would first have to be raised, to release the force on the latch, and then lowered.

The second option for the safety latch mechanism is to utilize the same latch design as previously discussed but instead of using a solenoid to engage and disengage the latch, a tension cable would be used. This tension cable would be connected directly to the chain and can be seen in Figure 20. The purpose of the tension cable is to detect when there is a chain failure. The geometry of the chain link that would have to be manufactured to allow for the addition of a tension cable, would be designed such that when chain tension is increased, torque is introduced to the chain link. This would rotate it in such a way that the tension on cable is increased as well. The shape of this chain link would likely be triangular, as this would maximize the resultant tension on the cable depending upon the geometry of the triangular shaped chain link. When tension of the cable is not present due to a chain failure, a set of latches would engage into the sawtooth track. The latches would be linked together by a thin rigid steel plate, to ensure simultaneous movement. A spring would be located behind the bottom of the latch and attached to the back side of the carriage to push the latch forward when tension was released. This safety latch is meant to be constantly disengaged during movement, until the immediate loss of tension is experienced due to chain breakage. This would cause the latch to engage into the sawtooth track, preventing catastrophic failure.

The third design option utilizes a spring-loaded latch very similar to what was just described, connected directly to the roller chain to create a constant tension on the latch. This design does not require external assistance from an electrical source, keeping it disengaged from the sawtooth track. This track can be seen in yellow in Figure 24.

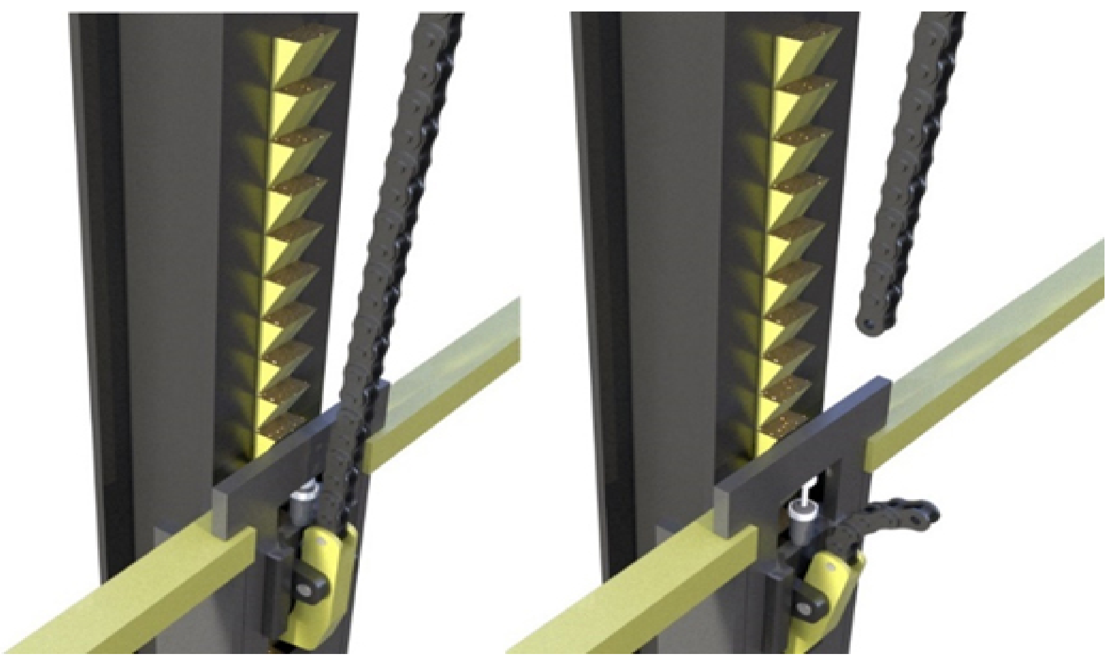


Figure 24: Sawtooth Latch System

This is nearly identical to Figure 23, where the chain pulls on the latch to keep it disengaged from the sawtooth pieces. As seen in Figure 25, the spring attached to the front of the top portion of the latch and the front of the carriage sliding plate assembly (hidden from view in-between components), will engage the latch into the sawtooth track. This system can engage as-is, without a sliding plate of steel, as discussed later in this section, however, the energy from impact in the event of a chain failure could cause damage to the bolt, or other components of the safety latch.

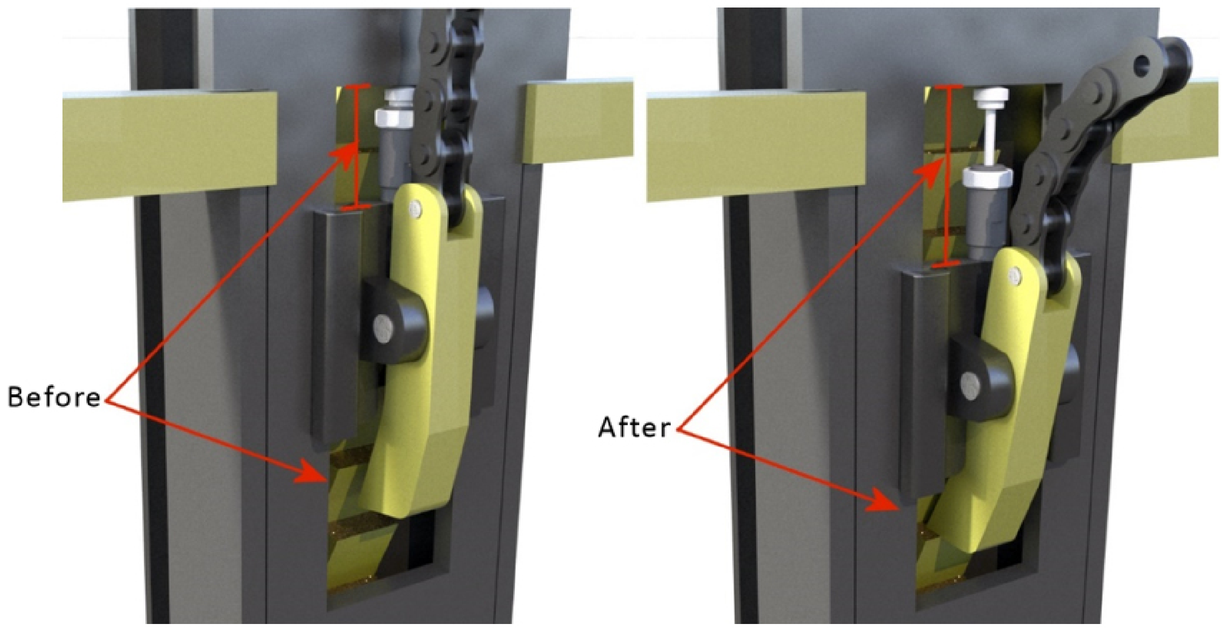


Figure 25: CAD model before and after a chain break

To alleviate this, a mount for the safety latch is integrated into a shock absorbing mounting plate as seen in Figure 26. A rectangular section would be cut out of the center column sled to allow for the sliding plate and shock absorber to be installed. When the latch engages from a loss of tension in the chain and quickly rotates into the sawtooth track, a lot of energy is involved. The energy from the 5,000 lb carriage that would normally be absorbed into the safety latch mounting tabs and bolt, will be transferred into the shock absorbing oil filled cylinder, or spring, reducing the chances of damage to the safety latch system.

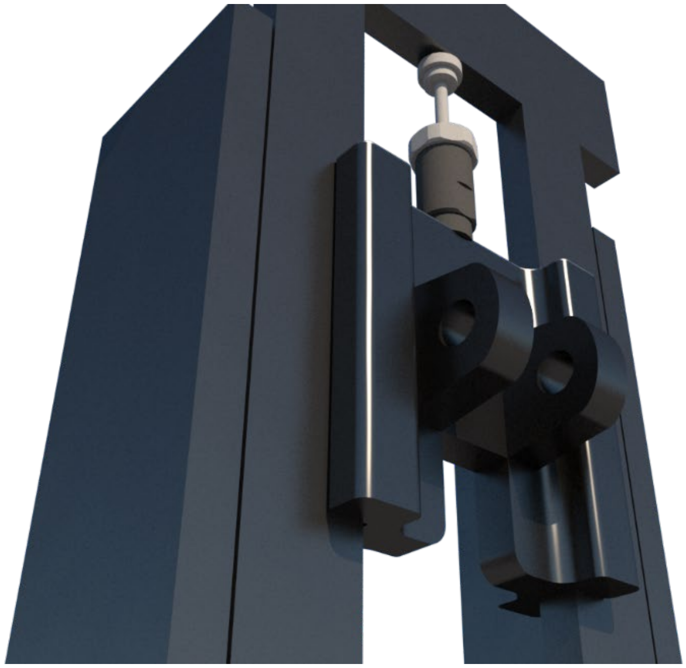


Figure 26: CAD model of Safety latch sliding mounting plate

It is important to note that when sizing the absorbers for this impact, this failure event is not expected to be a normal occurrence. Shock absorbers convert kinetic energy to heat, and if this were to be a repeating occurrence, heat considerations would need to be made, and absorbers of all types are available that made for those types of situations. If chain failure ever occurs, causing this safety latch system to engage, an inspection would be required before operation resumes to ensure that deformations or cracks are not present in the system resulting from the impacts involved.

3.3.5 *Sled Design*

The original two outer sleds that are part of the existing machine were built by an employee at Wendland as shown in Figure 27 and will not be redesigned. It was ideal to continue to use this sled design instead of introducing a new design for those columns. The proposed design will now include a third sled that must move along the center column. By modifying this existing design, the safety latch mechanism can be integrated into the sled while also widening front plate to account for the wider center column.

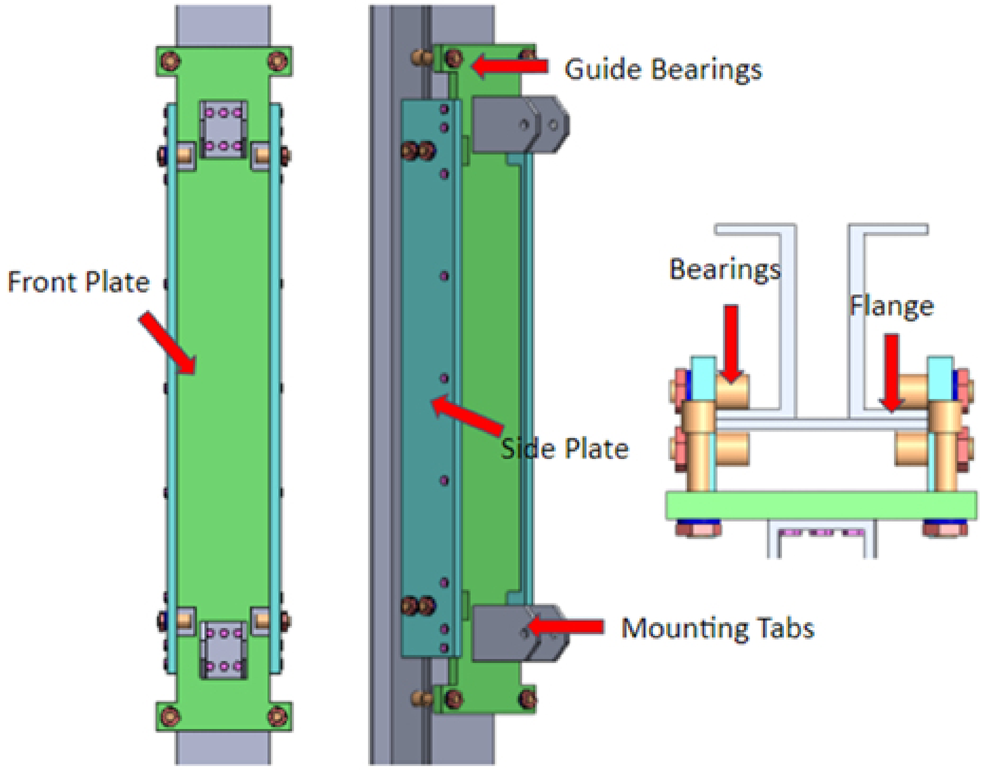


Figure 27: Sled Design on existing Machine

A rectangular section of material will be removed from the front plate, allowing the safety latch, as previously described, to mount within the sled. Widening the front plate will also be necessary to bring the total width that the bearings contact with to 11”, which is the width of the front plate on the center column, as shown in Figure 16, within section 3.3.2. The contact of these bearings, as seen in Figure 27, will still ride the edge of the front steel plate. In order to hold the sled tightly against the column, 4 pairs of bearings will also contact the front and back side of the built-up column flanges.

3.4 Calculations

In this section, calculations are conducted to determine minimum criteria for component selection. The first set of calculations is related to the column design, determining eccentric buckling forces and then the weld strength of the fillet weld used to join the c-channels to the front plate. The second set of calculations are related to determining the horsepower needed from the motor to adequately lift the assembly vertically. The third set of calculations are used to determine the static torque needed from the motor brake and the fourth set of calculations are used to determine sprocket and chain sizing. The fifth set of calculations are used to determine the shaft diameter needed to support the load of the machine and the last set of calculations related to the safety latch bolt and fabricated parts.

3.4.1 *Column Calculations*

3.4.1.1 Buckling Analysis

To verify that the proposed column addition will sufficiently support the weight of the machine, an eccentric column buckling analysis was done. This analysis is done to determine the maximum load a column can withstand before it collapses. In an eccentrically loaded column, meaning the load is acting at the edge of the column or on a bracket attached to the front, there is a moment that is combined with the axial loading. The stress distribution that acts over the cross-sectional area of the column is highest at the edge of the column closest to the applied load shown in Figure 28.

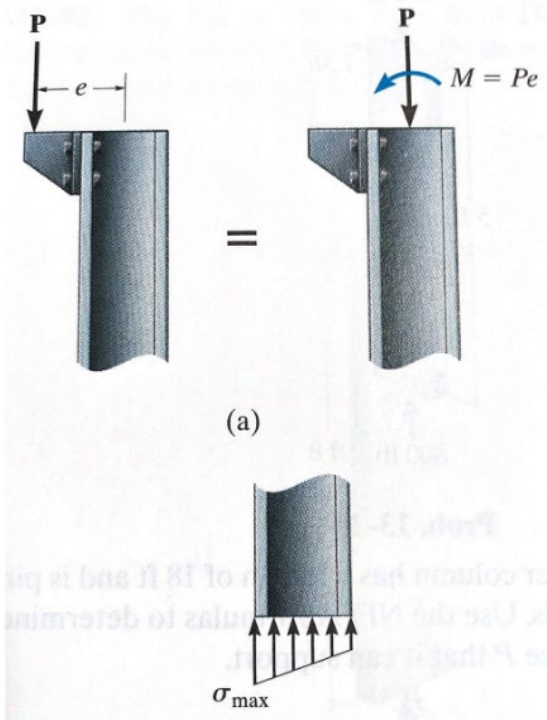


Figure 28: Example of eccentric loading on a column [21]

This is a result of a superposition of both the axial force P and the bending moment $M = Pe$, where P is the load and e is the eccentricity, or the distance from the centroid to the load. The variable c is the distance from the centroid to the outer fiber of the column. The maximum compressive stress is therefore found using Eq. (10).

$$\sigma_{max} = \frac{P}{A} + \frac{(Pe)c}{I} \tag{10}$$

If it is conservatively assumed that the entire cross section is subjected to the uniform stress σ_{max}, then it can be compared with the allowable compressive stress, σ_{allow}. If $\sigma_{max} \leq \sigma_{allow}$, then the column will support the intended loading. The first step is to calculate the slenderness ratio of the column about the y-axis. To calculate this value, Eq. (11) is used where, K is the effective length factor for the column, r is the radius of gyration, and L is the length of the column. For this column the height, L, is equal to 20 feet.

$$slenderness\ ratio = \left(\frac{KL}{r}\right)_y \tag{11}$$

To be conservative in this case, K =1, meaning that both ends are assumed to be pinned. The radius of gyration r is determined by using Eq. (12), where I_y is the moment of inertia along the y-axis of the cross section and A is the cross-sectional area.

$$r = \sqrt{\frac{I_y}{A}} \tag{12}$$

Substituting these values into Eq. (12) results in a slenderness ratio of 3.67. Next the lower bound of the slenderness ratio needs to be calculated. This is done by using Eq. (13), where E is Young's Modulus for A-36 steel, and σ_y is equal to the yield stress of A-36 steel.

$$\left(\frac{KL}{r}\right)_c = \sqrt{\frac{2\pi^2 E}{\sigma_y}} \tag{13}$$

Substituting 29,000 ksi for E and 36 kip for σ_y, results in a lower boundary slenderness ratio of 126.099. If $\left(\frac{KL}{r}\right)_y \leq \left(\frac{KL}{r}\right)_c$ then the condition for the slenderness ratio is satisfied and Eq. (14) can be used to determine the allowable stress, σ_{allow}.

$$\sigma_{allow} = \frac{\left[1 - \frac{\left(\frac{KL}{r}\right)_y^2}{2\left(\left(\frac{KL}{r}\right)_c^2\right)}\right]\sigma_y}{\frac{5}{3} + \frac{3\left(\frac{KL}{r}\right)_y}{8\left(\frac{KL}{r}\right)_c} - \frac{\left(\frac{KL}{r}\right)_y^3}{8\left(\frac{KL}{r}\right)_c^3}} \tag{14}$$

Substituting the values for the slenderness ratios results in an allowable compressive load of 16.57 kip. To determine the maximum compressive stress, σ_{max} , Eq. (10) stated earlier can be used. The values found in Table 2 are used to determine σ_{max}.

Table 2: Values used to determine the maximum compressive stress.

P =	5000 lb
A =	21.10 in2
e =	9.5 in
c =	5.5 in

Substituting the values found in Table 2, into Eq. (10) results in a value of 1.16 kip. Now comparing the two stress values, σ_{max} = 1.16 kip is less than σ_{allow} = 16.57 kip which means that the column will support the weight of the machine suspended from the face of the column [21].

3.4.1.2 Weld Strength Calculations for the Column Front Plate

To determine the strength of the ¼" fillet weld that is used to join the front plate and the c-channels together, the weld will need to be analyzed for shear. This calculation will determine whether the fillet size is sufficient for the shear stress that the column would be subjected to. If the load applied is not perfectly perpendicular to the fillet weld, the weld is in shear and its load carrying capacity is greatly reduced (33). Because of this reason, when designing welds, it is assumed that the weld will be loaded in shear as seen below in Figure 29.

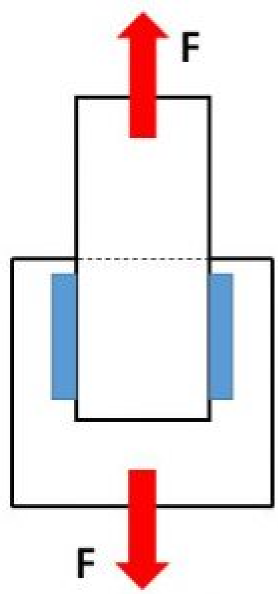


Figure 29: Fillet welds loaded in shear [29]

In this case the applied load is parallel to the welds. The forces are pulling the members being joined in opposite directions, which places the welds under shear. When a weld is in shear, the tensile strength of the filler metal can no longer be used to determine the strength of the weld. Instead, the tensile strength is reduced by a factor in order to assure safety. Clause 2 of American Welding Society: Structural Welding Code for Steel, section D1.1 requires that the minimum tensile strength of the filler metal be multiplied by 0.30 to obtain the allowable shear stress on the weld [29]. To determine the maximum shear stress of the weld τ, Eq. (15) is used where F is equal to the force the weld can handle and A is the effective area of the weld.

$$\tau_{allowable} = \frac{F}{A} \tag{15}$$

The effective area of a weld is calculated by multiplying the length of the weld times the throat of the weld. For design purposes, the theoretical throat is used as shown below in Figure 30.

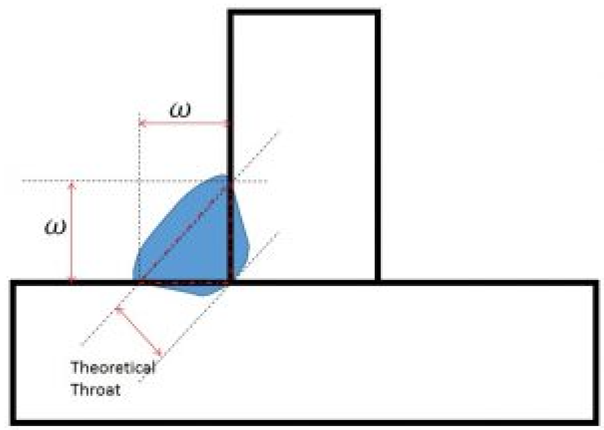


Figure 30: Throat weld diagram[29]

In Figure 30 above, ω is the fillet weld leg size. The theoretical throat is calculated by multiplying ω times the cosine of 45˚ which is 0.707. For all fillet welds with both legs being of the same size, the theoretical throat will be 0.707 x ω. The chosen fillet weld leg weld size for this built-up column is ¼" weld, spanning 240 inches. Therefore, the throat of the weld will be ¼" x 0.707 which results in a throat size of 0.177". The effective area of the weld is found by multiplying the length of the weld (240 inches) by the throat (0.177) and then by the number of total welds, which in this case is 2. The resulting effective area is determined to be 84.96 in2. The allowable shear stress value is determined by using the minimum tensile strength of the filler material used for the weld. In this case, ER70S-6 filler material will be used which has a tensile strength of 70,000 psi. Using Clause 2 from the AWS code book, to account for a safety factor, the tensile strength is reduced by 70% or multiplied by 0.30. This results in an allowable shear stress of 21,000 psi. Rearranging Eq. (15) to solve for the force that the weld can handle, results in a force of 1,784,160 lbf. This resulting value is more than sufficient for this application. To save cost and time, the lengths of the welds could be reduced, or the weld could be reduced in size.

3.4.1.3 Weld Strength Calculations for the Drive Shaft Assembly Platform

The strength of the weld that joins the drive shaft platform to the front plate of the column is determined using the same calculation method as in the previous section 3.4.1.2. For simplicity, the same size fillet weld of ¼" is used to calculate its strength in shear. In this calculation, the weld length used is 14" in length. The throat size is 0.177". Multiplying the throat size (0.177") by the length of the weld (14") and number of welds (2) results in an effective area of 4.956 in2. The filler material used is ER70S-6 with a tensile strength of 70,000 psi. Using the reduction in tensile strength of 70% according to the AWS code book yields an allowable shear stress of 21,000 psi. Rearranging Eq. (15) to solve for the force that the weld can handle, results in a force of 104,076 lbf. This resulting value is divided by the load of the machine and results in a safety factor of more than 20. The weld size is more than sufficient for this application.

3.4.2 *Motor Calculation*

The motor that is chosen to drive the chain and sprocket assembly must be capable of providing enough torque to lift the carriage at a minimum rate of 6 inches per second or 0.5 ft per second. This value was determined from observing the actual machine in operation. It is going to be assumed that a carriage will have a conservative acceleration rate of 0.5 ft/s^2 at the beginning of the lifting movement and a deceleration rate of -0.5 ft/s^2 at the end of the lifting movement. This will provide for a constant velocity during most of the carriage travel. To determine the power required to maintain this lifting speed, Eq. (16) is used, where P is equal to the power needed, W is equal to the weight of the machine, h is the height displacement discussed in section 3.2.2, and t is equal to the time it takes to travel the height displacement.

$$P = W * h/t \tag{16}$$

Substituting 5000 lbs for the weight of the machine, 10 feet for the height displacement, and 21 seconds for total travel height into Eq. (16), results in the total power needed for the motor to be 2143 lbf·ft/s or 3.9 HP.

To determine the torque requirement for the motor, an initial sprocket diameter must be chosen to run through the calculations. The sprocket diameter chosen to use for this application is 6 inches. This sprocket size will also be used in determining the chain size. Once the final calculations are made, the sprocket size can be adjusted to achieve results that are sufficient for the design. The equation used to determine the required torque is shown in Eq. (17) below.

$$T = P/\omega \tag{17}$$

P is equal to the power, determined using Eq. (17), and ω is equal to the angular velocity, or velocity divided by the radius of the sprocket. The velocity of the machine, 6 inches per second is converted to 0.5 feet per second for this calculation. After dividing this value by a radius of the sprocket, which is equal to 0.25 feet, the resulting value of ω equals 2 radians per second. This value is equivalent to 19 rotations per minute. By substituting 2143 lbf·ft/s for P and 2 radians per second for ω, the required torque value needed for this application is 1072 lbf·ft.

3.4.2.1 Lift Motor Selection

High torque, and low RPM motors are common in industrial applications. For the carriage lift system, this type of motor is desired to eliminate the need for an external gear reduction through chains and sprockets to increase the torque applied to the sprocket and motor chain. The Nord AC gearmotor shown in Figure 31, has a gear reduction of 93.5:1 giving an overall torque of 1143 lb-ft at an angular velocity of 19 RPM.

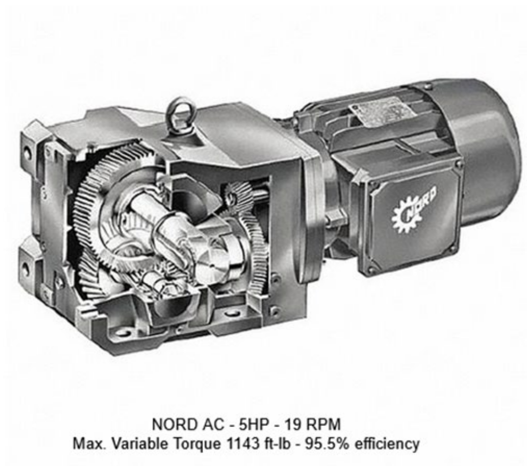


Figure 31: Motor selection for the lifting application [20]

3.4.3 *Motor Brake Calculations*

A motor brake is a safety feature that is utilized during lifting operation of the carriage assembly. It works by disengaging the brake when power is applied to the lifting motor and engaging the brake as soon as power is removed from the lifting motor. When power is removed during operation and the desired height is reached, the carriage should not fall under its own weight into a new position that is undesired. A motor brake prevents any unwanted drop in carriage height due to power loss at the lifting motor but is not a failsafe in the event of chain failure, as was previously discussed in section 3.3.4.

Motor brakes are sized according to how much torque they can resist. The lifting motor shaft is directly inserted into the motor brake, and in this case at a 1 to 1 gear ratio related to the load that is being driven. To calculate the amount of static torque that is required for the motor brake, Eq. (18) is used, where W is equal to the weight of the carriage, D is the diameter of the sprocket, and T is the static torque.

$$T = W * \frac{D}{2} \tag{18}$$

The weight of the carriage is approximately 5000 lbs, and the diameter of the sprocket is 6 inches. Substituting these values into Eq. (18) results in a static torque value of 1125 lbf·ft. This value will be used when selecting a motor brake for the assembly.

3.4.3.1 Motor Brake Selection

The motor brake must be able to handle the static torque when the carriage is at rest. The motor brake selected is the style SCEB Class S3 spring set brake, as seen in Figure 32, which disengages when voltage is applied and comes in several options for static torque loads. The brake size option 1004 has a nominal static torque load of 1400 lb-ft, which is adequate for this application. This motor brake is regularly integrated into VFD or Variable Frequency Drive systems, as will be later discussed in section 5.5.2.

Style SCEB, Class S3

Style SCEB, Class S3 Spring-Set Brake is end shaft mounted with base. Released when voltage is applied.

- 450-12,000 lb-ft static torque.
- See pages 44 and 45 for product selection.

Performance Data

MECHANICAL

Brake Size	Nominal Static Torque (lb-ft)	Wk^2 ($lb\text{-}ft^2$)	Approx. Shipping (wt-lbs)	Max. RPM
802	450	.36	86	3600
804	900	.53	97	3600
1004	1400	1.35	170	3600
1006	2100	1.85	180	3600
1204	3500	2.10	270	2400
1206	5000	4.35	280	2400
1406	7800	10.70	370	2000
1606	12000	19.00	660	1800

Figure 32: Motor brake selection for the lifting application [8]

3.4.4 *Chain and Sprocket Calculations*

Using a motor with a drive chain and sprocket assembly is a common method used in motion control systems. Roller chains and compatible sprockets provide a reliable method to precisely move objects from one place to another. Roller chains have significant advantages such as very little slippage or stretching across the chains, even with a heavy load and are incredibly efficient and long-wearing. Roller chains are exceptional for use in a high-stress environment as they will hold up under even the most difficult conditions such as high and low temperature or start-stop action usage. Roller chain sizes are standardized according to ANSI/ASME or ISO British standards, which are meant to provide interchangeability of design and dimensions for chains, sprockets, and attachments (30).

The tensile load and working load of the chain are key properties in determining the correct size chain for the application. In this design consideration, the working load is equal to the weight of the carriage, 5000 lbs. But, to ensure that the chain will not break under a weight that could be slightly higher, a minimum factor of safety of three is built into the design. This means that the chain selected should have an approximate allowable working load of 15,000lbs.

The tensile load, sometimes referred to as the ultimate tensile strength, UTS, is the static load required to break the chain. Tensile strength values in chains can be similar but can result in dissimilar working loads. There is no consistent relationship between a roller chain's working load capacity and its ultimate tensile strength. Load or tension applied to the chain in service should never exceed 1/6th of the UTS. A roller chain should never be loaded above 50% of UTS for even one cycle. Doing so will permanently damage the chain [13].

For the initial calculation when determining chain size, a sprocket with a six-inch diameter is chosen. A six-inch diameter sprocket has a circumference of 18.85 inches. From the lifting speed analysis from section 3.2.2, using a lifting velocity of 6 inches per second, means that the sprocket will rotate 1/3 of its circumference per second. Converting this to rotations per minute results in a sprocket RPM of 20. Next, a chain selection chart is used, shown in Figure 33 below.

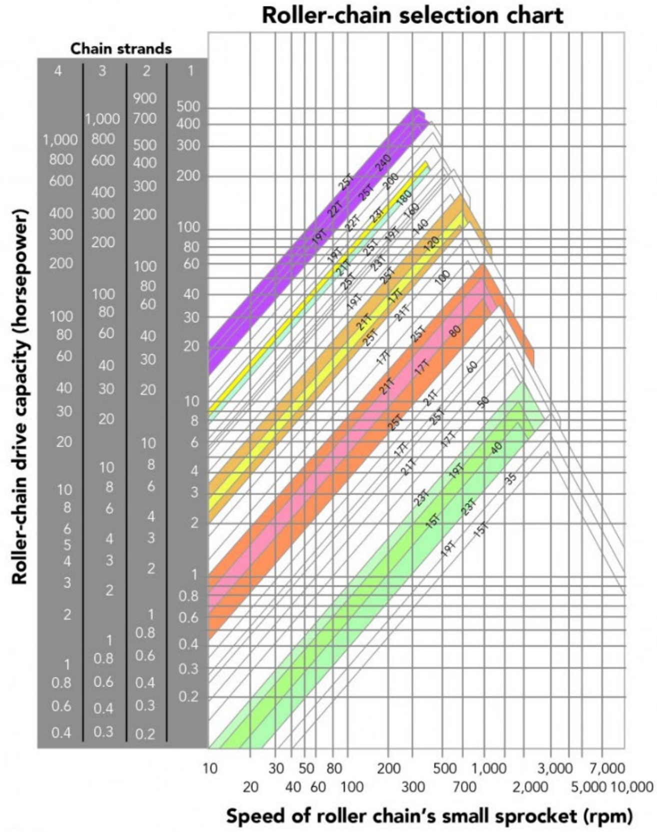


Figure 33: Roller chain selection chart. [26]

According to Figure 33, using a motor with approximately 4 HP, determined in section 3.4.2, and a speed of the sprocket at 20 RPM's, a chain size #160 was selected. Roller chain manufacturers have different tensile strengths and working loads for their chains. For these calculations, a roller chain from USA Roller Chain & Sprockets was used. It has a tensile strength of 62,200 lbs and a working load of 15,490 lbs. This specific chain will be sufficient for the working load of 15,000 lbs that was determined with a 3x safety factor earlier in the calculation. The six-inch diameter sprocket that was initially chosen will also be sufficient for this design. Chordal action, pitch, and other properties were not analyzed, as these. Based on the chain type required, the sprocket must match up to meet ANSI standards. It should be selected using the guidance from USA Roller Chain & Sprockets to meet ANSI standards.

3.4.4.1 Chain and Sprocket Selection

There are 14 different ASME/ANSI B29.1 standard roller chain. Using the horsepower of the motor and the RPM of the small sprocket, ANSI standard chain, size #160, was selected from the quick selection chart. In accordance with Figure 33. #160 chain has a tensile strength of 62,200lbs and a working load of 15,490lbs to achieve the desired safety factor of at least 3. Additional chain info such as the pitch, width, diameter, height, and thickness are shown in Figure 34 and Table 3.

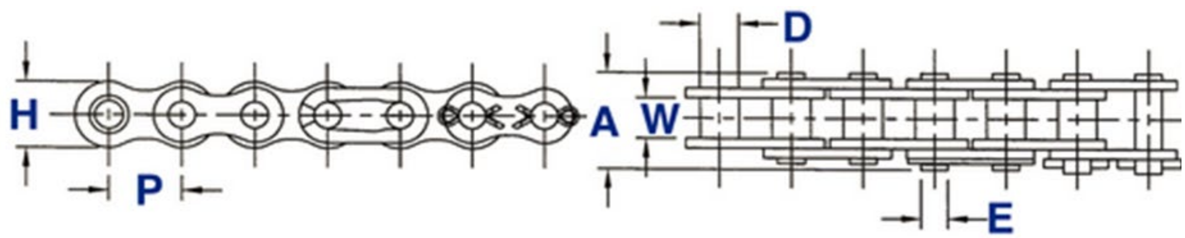


Figure 34: Chain Dimensions [9]

Table 3: Chain Specifications

Working Load	15,490 lbs
Pitch (P)	2.000 in
Roller Width (W)	1.250 in
Roller Diameter (D)	1.125 in
Plate Height (H)	1.898 in
Plate Thickness (T)	0.250 in
Pin (Diameter)	.0562 in
x Safety Factor	3.44
Overall Width (F)	2.524 in

The sprocket selected in 3.4.2, meets ANSI standards for the contact between roller chain and sprocket. This ensures that the maximum life of the chain and sprocket are achieved. The details for the sprocket are seen in Figure 35 and Table 4.

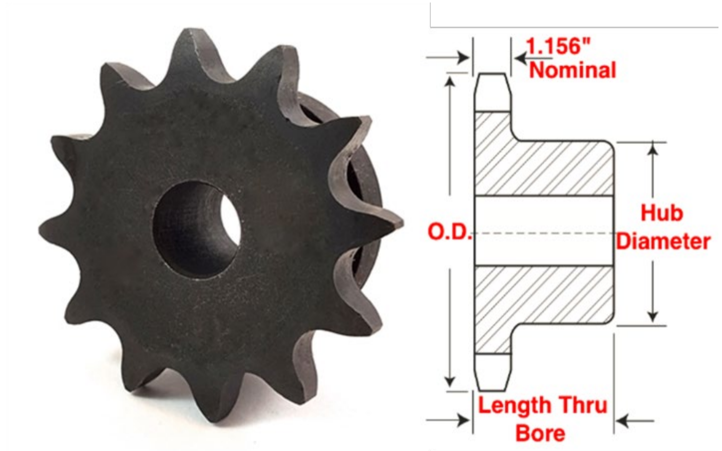


Figure 35: 160B9 Sprocket [9]

Table 4: Sprocket Selection & chain length for Lift System

Sprocket selection for all Sprockets in Lift	
Sprocket Size:	160B9
Nominal Width	1.156”
Tooth Count	9
Hub diameter	3-5/8”
Max bore size	2-1/8”
Outside Diameter	6.70”
Technical Info	
Motor Drive Gear Ratio	1:1
Drive Chain Length (approx..)	32”
# of links (approx..)	16
Hoist Gear Ratio	1:1
Hoist Chain Length (approx..)	438”
# of links (approx..)	219
Sprockets have been hardened and have a fully black oxide coating, increasing the resistance to wear and early corrosion from the environment.	

For the selection of the additional sprockets within the continuous chain loop of the lift system and separate chain drive from motor to lift shaft, this same method was used, resulting in identical sprockets throughout the lift system.

When computing the total number of chain links needed, some estimations must be made to consider that the sprocket distances during install could vary with our estimates. This is even more the case when considering the positioning of the sprockets within the hoist chain length, as indicated by the red arrows in Figure 36.

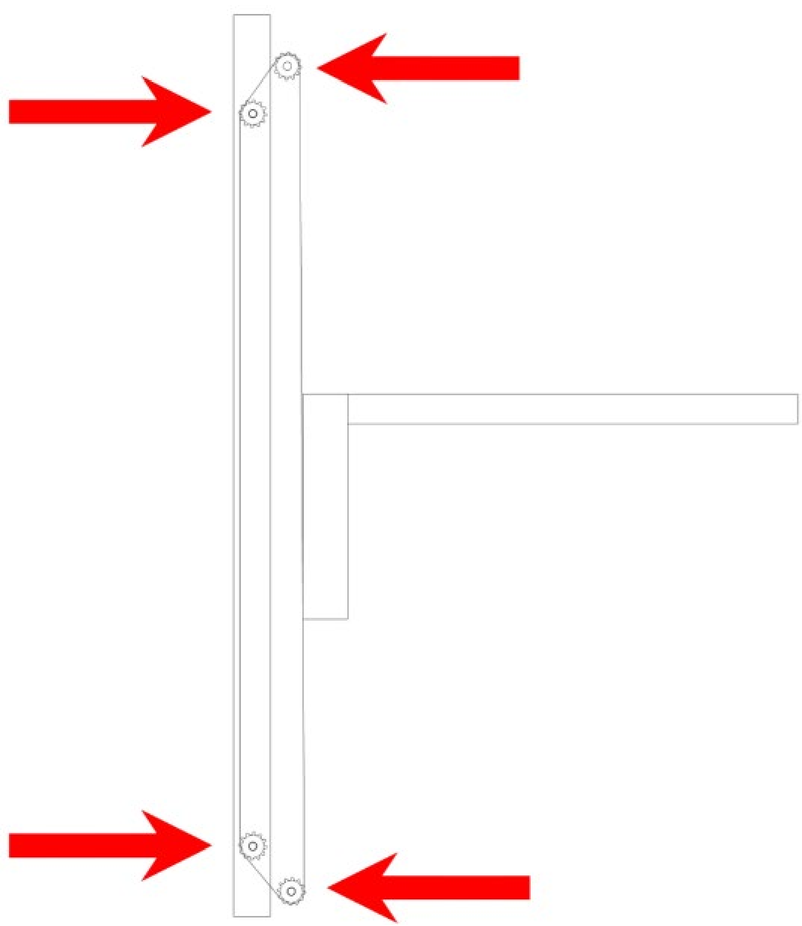


Figure 36: Hoist chain sprockets

These discrepancies do not however, affect the integrity of the lift system design. Consulting with the supplier or manufacturer of the chain and sprockets with precise center to center sprocket spacing dimensions will result in proper chain length calculations. These roughly calculated chain lengths can be seen in Table 4.

3.4.5 *Shaft Diameter Calculation*

One of the most critical points of the lifting system is the chain drive assembly. The shaft in this assembly will be bearing most of the load during lifting operations and will be subjected to bending and torsion simultaneously. The outside edge of the circular shaft will experience significant compressive and tensile stresses and the cylindrical circumference of the shaft will be in shear stress due to the torsion applied during lifting. The highest combined bending and torsional stresses will occur at the point on the shaft where the sprocket is located, denoted in blue in Figure 37.

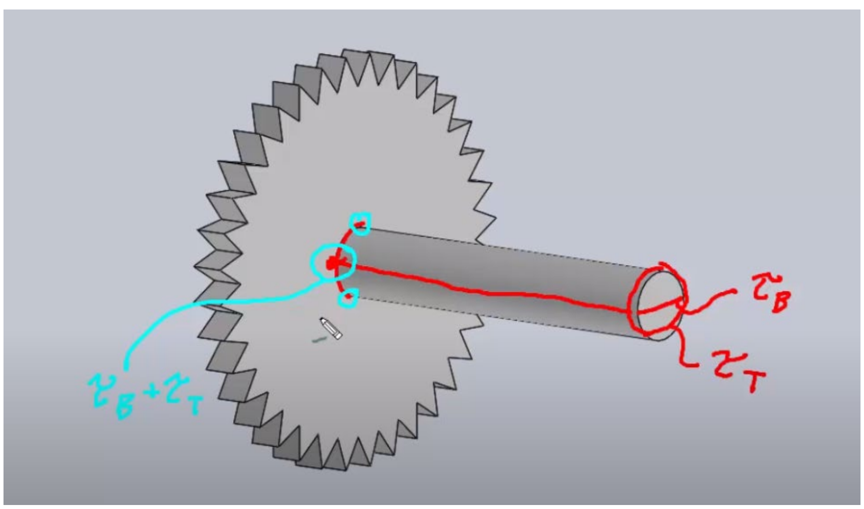


Figure 37: Sprocket and shaft bending and torsional stresses [15]

During rotation, the locations of the high stress areas are constantly being shifted around the shaft at the sprocket location. This area will be subjected to significant fatigue and the proper selection of shaft diameter is crucial to prevent failure. To find these crucial values, calculations will need to be conducted to determine bending stresses, torsional stresses, and then combined bending and torsional stresses. To begin calculations, known values for the machine need to be expressed. These values are shown in Table 5 below. The load of the machine, P, is given from the weight analysis in section 3.2.1. The torque, T, on the shaft is a value that can be calculated. The factor of safety, abbreviated F.S., is a design consideration determined by how critical the area of design is. In this case, since this part of the machine is the most crucial, a factor of safety of three is included in the design. This means that the diameter of the shaft that is determined will hold three times the amount of weight necessary.

Table 5: Given values for the shaft analysis.

P =	5000 lbs
T =	15000 lb-in
F.S.=	3

The torque that the shaft is subjected to is determined by the sprocket diameter attached to the shaft. It is going to be assumed that the sprocket drives the torque on the shaft. The sprocket chosen for the purpose of this application is a 6-inch diameter sprocket, shown in Figure 38. To determine the torque applied onto the shaft, the radius of the sprocket of 3 inches is multiplied by the weight of the machine, 5000 lbs. This results in a torque value of 15000 lb-in.

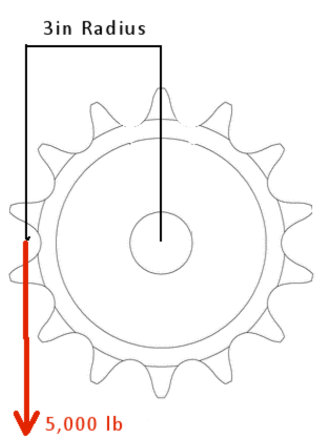


Figure 38: Sprocket torque and radius

As previously mentioned, the values that are of interest to find in this application are the principal stresses, ϑ_1 and ϑ_2, on the top and bottom of the shaft, the maximum shear stress, τ_{max}., and the combination of the principal and max shear stresses. These locations are where the gear is positioned on the shaft along with the center of the shaft at this location, shown in Figure 39.

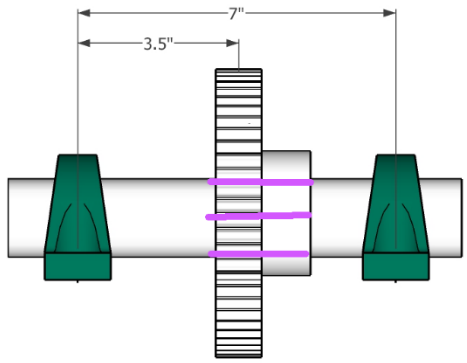


Figure 39: Stressed on shaft shown in pink [11]

To begin calculations to determine bending of the shaft, a shear and moment diagram is going to be used for this scenario. The first step is to create a free body diagram of the shaft, which is 7 inches in length, with a point load of 5000 lbs in the center, colored in red. This represents the weight of the carriage pulling down on the shaft, while the pillow block bearings that mount onto the lift assembly platform are represented by the 2500 lb reaction forces, as seen in Figure 40 below.

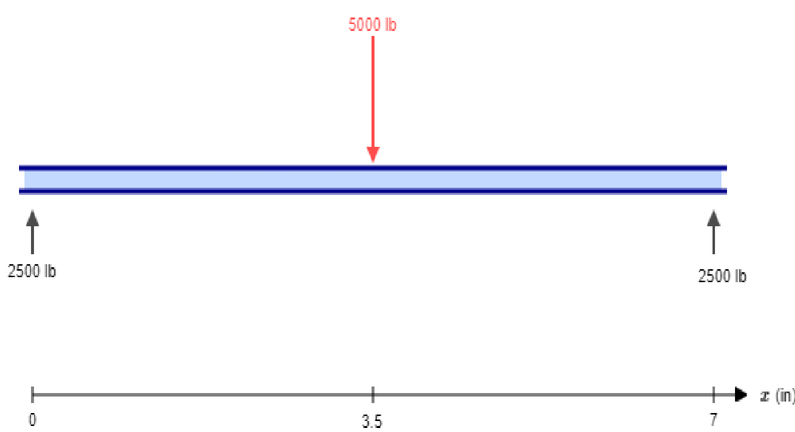


Figure 40: Free Body Diagram of a simply supported beam.

The reaction forces are equal on each side of the shaft since the point load is located directly center of the shaft, at 3.5 inches. From this free body diagram, a shear and moment diagram can be created, as seen in Figure 41 and Figure 42. The shear demand value is equal to the reaction force on one side of the shaft, which in this case is equal to 2500 lbs.

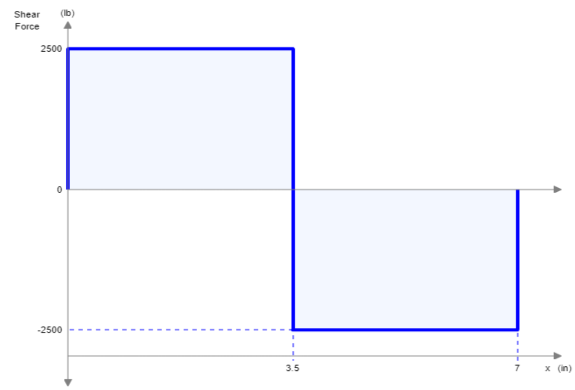


Figure 41: Shear diagram of the shaft.

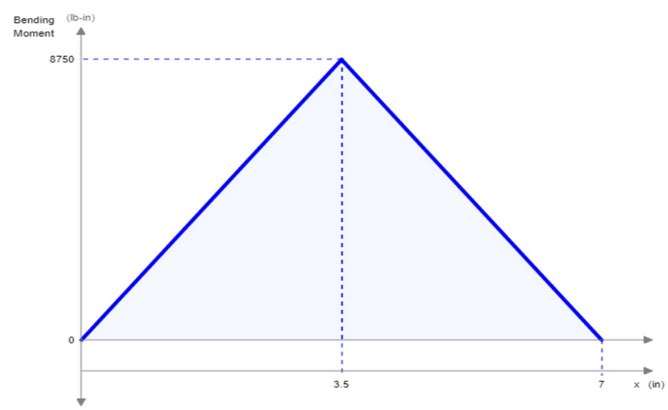


Figure 42: Bending moment diagram of the shaft.

The maximum moment for the shaft would occur at the sprocket location, directly center of the shaft, shown at the peak in Figure 42. This moment can be calculated by using Eq. (19), where F is equal to the reaction force of 2500 lbs and d is equal to the distance where there is maximum moment, 3.5 inches.

$$M = F * d \tag{19}$$

Solving for M results in a maximum moment value of 8748 lb-in. This value can now be used in all subsequent calculations for determining the proper diameter of the shaft.

An important step to determining the correct shaft diameter is to assume a diameter to begin working with. For this case, the diameter, D, chosen to use is 1.75 inches. This number was chosen because it is a common shaft diameter used in industrial motors. The next step is to calculate some of the values that will be used in later equations. The first value is the moment of inertia of a circular solid shaft, I, using Eq. (20).

$$I = \frac{\pi D^4}{64} \tag{20}$$

Substituting 1.75 inches for D results in a moment of inertia value of 0.46 in^4. This value is used to determine the normal stress from bending using Eq. (21) where, M is equal to the maximum moment of 8748 lb-in, c is equal to the radius of the shaft, 0.875 in., and I, is the moment of inertia calculated in Eq. (20). Substituting these values into Eq. (21) results in a bending stress value of 16,640 psi.

$$\vartheta_B = \frac{Mc}{I} \tag{21}$$

Next, the torsional shear stress, τ_T, needs to be calculated. The torsional shear stress is found using Eq. (22).

$$\tau_T = \frac{Tc}{J} \tag{22}$$

In Eq. (22), T is equal to the torque calculated, c is still equal to the radius of the shaft 0.875 in., and J is the polar moment of inertia, otherwise stated as the shaft's ability to resist torsion. To calculate J, the polar moment of inertia, Eq. (23) is used, substituting in the diameter of the shaft for D.

$$J = \frac{\pi D^4}{32} \tag{23}$$

It can also be noted that the polar moment of inertia for a circular shaft is two times the moment of inertia. Using Eq. (23), the value for J is calculated to be 0.92 in^4. From this value, the torsional shear stress τ_T, is determined to be 14,266 psi.

Next, Mohr's circle is used to show a geometric 2-D representation of the principal stresses associated with the shaft. The first step to draw Mohr's circle is to determine the center of the circle, C. The center of the circle is the average of the normal stress in one direction and then the normal stress in the perpendicular direction. This can be done using Eq. (24) below.

$$C = \frac{\vartheta_x + \vartheta_y}{2} \tag{24}$$

In this application for Mohr's circle, $\vartheta_B = \vartheta_x = 16{,}640$ psi. This is the normal stress in the x-direction. The stress in the y-direction, ϑ_y, is equal to zero because there is only a downward load being applied, and the torsional shear stress, $\tau_T = \tau_{xy} = 14{,}266$ psi. Substituting these values into Eq. (24) determines that the center of the circle, C, is located at 8,320 psi. The radius of the circle is found using Eq. (25).

$$R = \sqrt{\left(\frac{\vartheta_x - \vartheta_y}{2}\right)^2 + {\tau_{xy}}^2} \tag{25}$$

Substituting the values for ϑ_x, ϑ_y, and τ_{xy} determines that that the radius, R, of Mohr's circle will be 16,515 psi. This value will also be the maximum shear stress that the shaft will be experiencing during loading.

The next step in constructing Mohr's circle is to determine principal stresses, ϑ_1 and ϑ_2. These values are calculated by using Eq. (26) and Eq. (27) below.

$$\vartheta_1 = C + R \tag{26}$$

$$\vartheta_2 = C - R \tag{27}$$

Substituting the values for both C and R determines that $\vartheta_1 = 24{,}835$ psi and $\vartheta_2 = -8{,}915$ psi. Now that these values have been determined, Mohr's circle can be constructed and is shown in Figure 43.

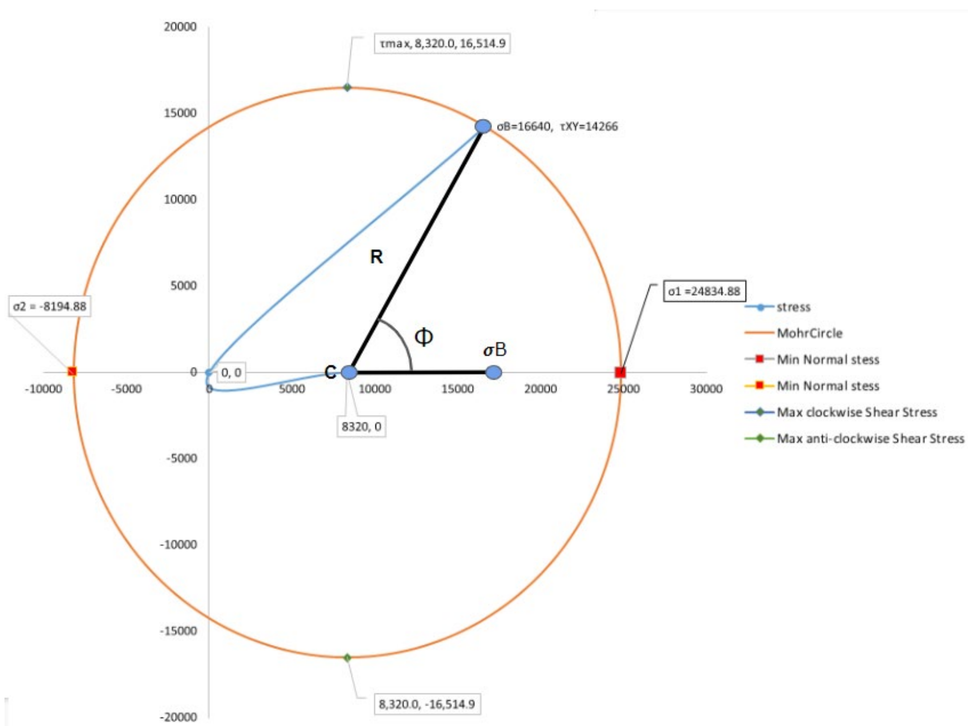


Figure 43: Mohr's circle for maximum bending and torsional stress.

The angle $\emptyset$, in between the radius of the circle R, and the principal stress ϑ_B, is calculated by taking the inverse tangent of τ_{xy} over $\vartheta_B - C$ shown in Eq. (28).

$$\emptyset = \tan^{-1} \frac{\tau_{xy}}{\vartheta_B - C} \tag{28}$$

The value for $\emptyset$ is calculated to be 59.74° but, when using Mohr's circle, $\emptyset$ is equal to 2θ. This angle is representative of the orientation of the principal normal stresses and principal shear stresses in the actual physical space. Therefore, θ is equal to half of $\emptyset$, which equals 29.87°. From the x-direction, the first principal stress is at an angle of 29.87° in physical space on the shaft [17].

The next step is to use the same methods to determine only the shear stresses from both torsion and bending. For a circular shaft, the shear stress from bending can be determined using Eq. (29), where V is equal to the shear force of 2500 lbs, and A is equal to the cross-sectional area of the shaft which is equal to 2.41 in^2.

$$\tau_B = \frac{4V}{3A} \tag{29}$$

Substituting these values, the bending shear stress τ_B is equal to 1,383 psi. When determining only the shear stresses from both torsion and bending, the stresses in the normal direction for ϑ_x and ϑ_y are equal to zero, and $\tau_{xy} = \tau_T + \tau_B$, which results in a value of 15,637 psi. This value is now compared with the combined bending stress/torsional shear stress. Comparing the two values determines which stresses will have more of an impact on the shaft. The combined bending stress/torsional shear stress has a value of 30,906 psi, whereas only the shear stresses have a value of 15,637 psi. Since the combined bending stress/torsional shear stress is the higher value, this value will be used to determine whether the diameter of the shaft needs to be increased or if it can be decreased. After multiplying the combined bending stress/torsional shear stress value by the factor of safety of three, the 1.75 in diameter shaft will need to be made of a material that is capable of withstanding 92,718 psi.

3.4.5.1 Shaft Selection

Researching material properties of certain types of steel that could be used for this application using www.matweb.com, it was determined that AISI 1045 steel would be the most appropriate material. It is a medium carbon steel that has a high tensile yield strength and a high modulus of elasticity. This material is often used in motor shafts, gears, and axles that are subjected to cyclic motions. The yield strength of the 1045 steel is approximately 65,300 psi. Because the yield strength of the material is lower than the actual applied stresses, design optimization must occur. To do this, a spreadsheet in excel was created using the calculations previously discussed to quickly input the chosen diameter of the shaft to determine the stress values. When the shaft diameter was increased to 2 inches, the calculated applied stresses on the shaft were approximately 62,000 psi which is lower than the yield strength of the material. It is decided that the 2-inch diameter shaft will be sufficient for this application [28].

3.4.6 *Safety Latch Calculations*

When determining the energy needed to absorb the impact when the safety latch engages, it is necessary to calculate the total energy, E_T, using Eq. (30) where $E_k$is the potential energy and E_w is the work energy.

$$E_T = E_k + E_w \tag{30}$$

Table 6 lists the values needed to calculate the total energy of an impact. With these values it will be possible to determine the size of an absorber that will be needed to reduce chances of machine damage. Using the values from Table 6, the potential energy can be determined using the total weight of the machine multiplied by the height, h, between each sawtooth in the sawtooth track or $E_k = wh = 5000 * 3 = 15000\ in\ lb$. To get the energy based off the work being done by the absorber, the stroke length, s which is the displacement of the spring or shock absorber from its at rest state, to when it is fully compressed is multiplied by the weight or $E_w = ws = 5000 * 2 = 10000\ in\ lb$.

Table 6: Values to determine the energy

Variable	**Value**
Weight =	5000 lb
Height =	3 in
# of Cycles =	1
Stroke length =	2 in

Adding these energy values from these calculations together, the total energy using equation (30) is $E_T = 25{,}000\ in\ lb$. When sizing these components, heat is usually considered from the conversion of kinetic energy to thermal energy based on the number of cycles the absorber must withstand. In this case the number of cycles does not have any influence since the safety latch will only engage if a chain failure event occurs. In Figure 44 ,the Enidine HD 1.5 x 2 is recommended to be capable of absorbing 27,000 in-lb and any similar type of absorber with the same energy absorption capability could be used if it has a stroke length, s, of 3 inches or less.

Figure 44 : Enidine HD 1.5 x 2 Shock Absorber [13]

3.4.6.1 Bolt Calculations

The bolt size chosen to mount the safety latch onto the latch mounting tabs is a ¾” FF1554 Grade 105 bolt manufactured by Portland Bolt. The specifications for this type of bolt are shown in Figure 45 below.

Mechanical Requirements

Grade	Identification	Size, Inches	Tensile, ksi	Yield, ksi min	Yield, MPa min	Elong. % min	Reduction of Area %, min
105	PB AB105	1/2 – 3	125 – 150	105	724	15	45

[1]Color coding (red) is required while permanent stamping with manufacturer's identification (S2) and grade designation (S3) are supplementary requirements.

Nuts and Washers

Grade	Identification	Size, Inches	Recommended A563 Nut				Washer
			Plain Finish		Hot-Dip or Mechanical Zinc Coated		
			Grade	Style	Grade	Style	
105	PB AB105	1/2 – 1-1/2	D[1]	Heavy Hex	DH[2]	Heavy Hex	F436
		1-5/8 – 3	DH[2]	Heavy Hex	DH[2]	Heavy Hex	

Figure 45: Bolt specifications for F1554 Grade 105 [14]

When analyzing a bolt used in this application, the property of most concern is the shear strength of the bolt. Shear strength is defined as the maximum load that can be supported prior to fracture, when applied at a right angle to the fastener's axis. A load occurring in one transverse plane is known as single shear. Double shear is a load applied in two planes where the fastener could be cut into three pieces [15]. Figure 46 illustrates the forces acting on the bolt. In this case, since the bolt is supported by two mounting tabs each carrying half of the load, and the latch attached to the bolt opposing the full load, double shear strength needs to be calculated.

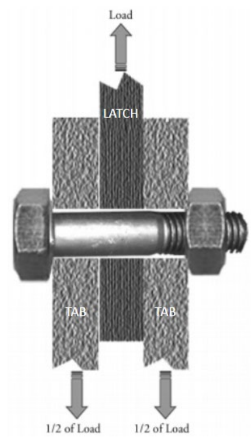


Figure 46: Forces acting on the bolt [15]

To determine the shear strength of the fastener, the total cross-sectional area of the shear plane is important. There are two possibilities for applied shear load (as illustrated below in Figure 47). One possibility is that the shear plane occurs in the threaded portion of the bolt. Since shear strength is directly related to the net sectional area (i.e. the amount of solid bolt material in the diameter), a narrower area will result in lower bolt shear strength [15].

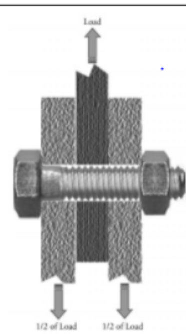
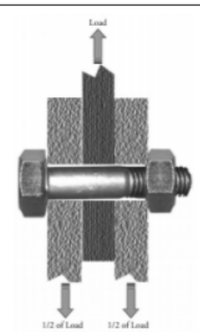
Double Shear Through Threads (1/2-13 SAE J429 Grade 8)	**Double Shear Through Body** (1/2-13 SAE J429 Grade 8)
	
1/2-13 Thread Root Area: 0.126 sq-in	Minimum Body Area 0.191 sq-in
60% of Tensile Strength: 90,000 PSI	60% of Tensile Strength: 90,000 PSI
Double Shear = 2 x 0.126 sq-in x 90,000 PSI	Double Shear = 2 x 0.191 sq-in x 90,000 PSI
Double Shear = **22,680 lbs.**	Double Shear = **34,380 lbs.**

Figure 47: Shear through the threads or through the body of a bolt [15]

Using the material properties shown in Figure 45, the shear strength of the bolt can be calculated by using Eq. (31), where τ_2 is the double shear strength, A is equal to the cross sectional area of the body of the bolt, and UTS is the tensile strength of the material.

$$\tau_2 = 2 * A * (0.6 * UTS) \quad (31)$$

Using a ¾" diameter bolt yields a cross sectional area of 0.442 in^2. The tensile strength of the material is 125,000 psi. Substituting these values into Eq. (31), results in a shear strength value of 66,300 lbf. This shear strength value is the minimum breaking strength of the bolt.

3.4.6.2 Finite Element Analysis of Fabricated Components

To verify that the design for the safety latch mounting tabs is sufficient to support the weight of the carriage at 5000 lbs, a finite element analysis was done in Solidworks. Each component element for the safety latch was created using Solidworks and analyzed separately to determine the stresses, strains, and displacement. Figure 48 shows the results from the analysis of the mounting tabs attached to the latch slider. In the figures, each of the analysis for stresses, strains, and displacements are shown. From the analysis an exaggerated image is shown with a small deformation of 0.0197 mm. The von Mises stresses are in the range of 5,867 psi to 14,080 psi, which is under the yield strength of 36,260 psi for A36 steel, indicating that it will not fracture at any point during loading.

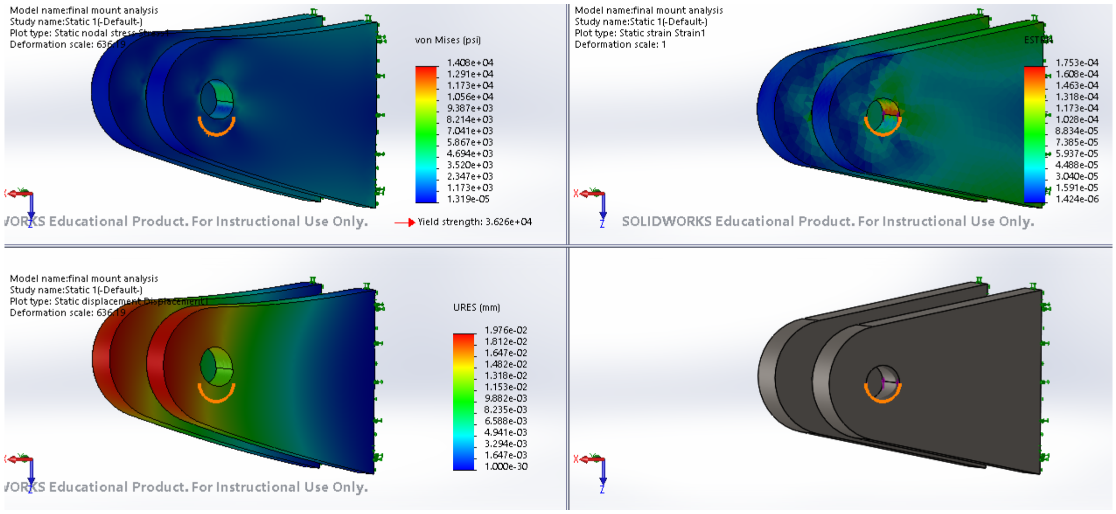


Figure 48: FEA of mounting tabs

In the next analysis shown in Figure 49, the sawtooth track was tested to determine if when loading occurs, will the sawtooth support the weight. At the edge of the tooth, it shows deformations of about 0.01009 mm. Due to the nature of the safety latch system, the sawtooth must only carry the weight, and deformations are not a concern. Even if they were, 0.01009 mm is acceptable. Again, like the other analysis, the stresses of 3,745 psi are well below the yield strength of the material at 36,260 psi.

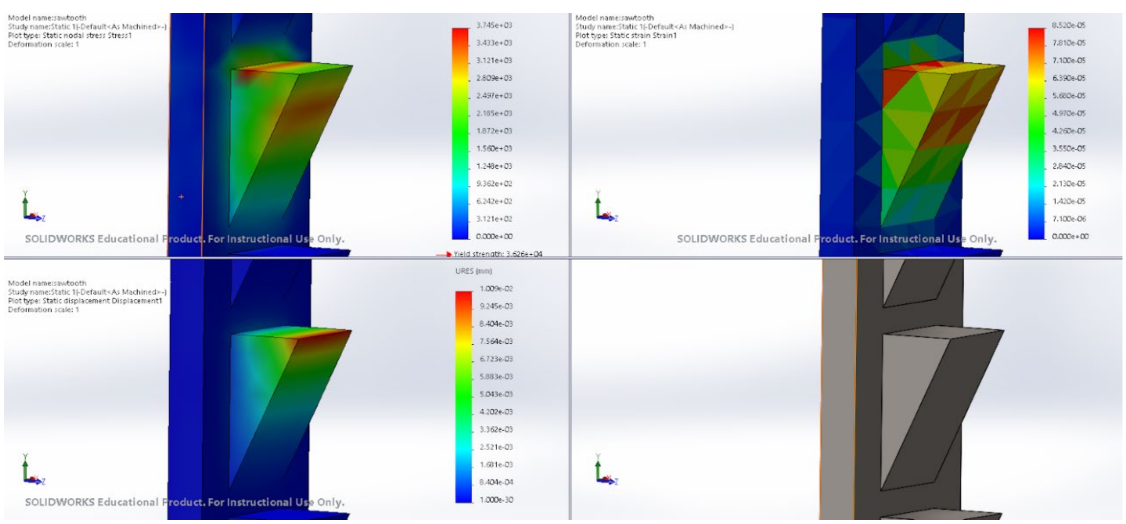


Figure 49: FEA of the Sawtooth catch

In Figure 50, the final analysis test was conducted on the safety latch to determine if it would be able to sustain the loading. Similar to the previous two analyses conducted, the latch will produce similar results as far as little to no deformation with a value of only 0.005639 mm. The stresses are 9,974 psi, well below the 36,260 psi yield strength of the material.

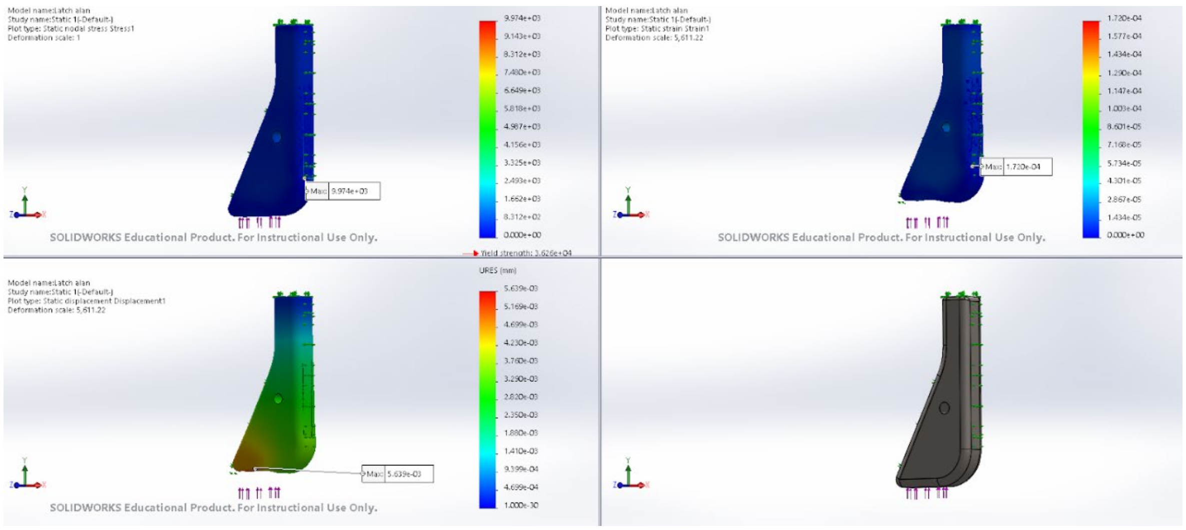


Figure 50: FEA of the Safety Latch

After running the study in Solidworks, it was determined that the design for each of these components would be sufficient for the amount of loading that would be placed on them. In all cases, deformations remained low, allowing for some flexibility in the design [22].

3.5 *Final Design of Lift System*

The final design for the lift system can be seen in Figure 51 below. The selected motor, chain and sprockets have been added to the new center column. It can be concluded that after calculating the torque required to lift the jib assembly into position for welding, the motor, sprockets, shafts, motor brake, and chain in combination with the VFD, are acceptable components. The existing machine was supported by the two outer columns, so the addition of a third column, will only help to increase the overall stiffness in the columns While also helping to stabilize the movements vertically from the addition of the 3rd sled. The addition of a safety latch system, consisting of the sawtooth track, latch, spring, shock absorber, and sliding plate mount, will reduce the chances that a catastrophic failure of the chain will lead to the carriage falling towards the ground, possibly injuring any person below the machine.

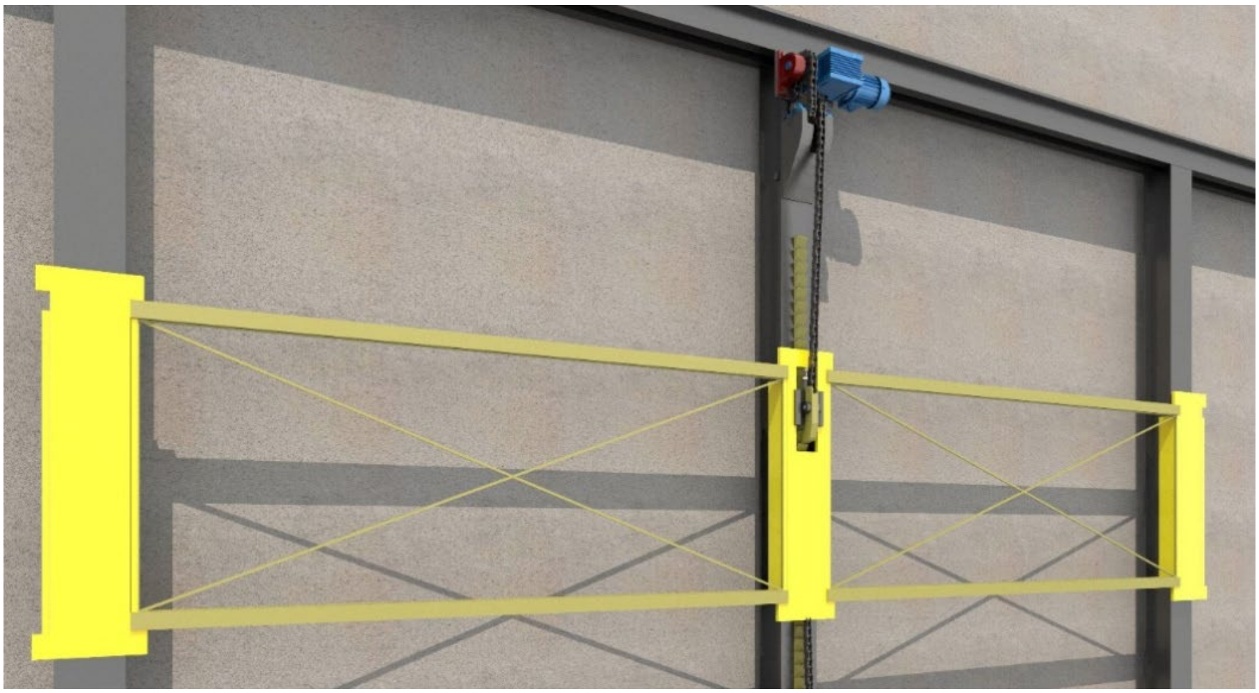


Figure 51: Final Design of Lift System

4 Machine Jib Rotation Improvements

In this section, a jib assembly rotation analysis will be conducted, and then suggested improvements to the existing machine will be discussed. First, the existing conditions of the jib assembly are provided along with safety concerns that need to be addressed with the proposed design. Next, design boundary conditions will be discussed and then several design options will be evaluated based on client needs and applicability of each to determine the best design solution. Then, calculations will be performed to determine component sizing. And lastly, a conclusion reporting the final design decision and justifications will be presented.

4.1 Existing Conditions

4.1.1 ***Manual Movement***

The existing machine is capable of rotating a pair of 14-foot parallel jibs that are connected with a 24' long weld track, forming a parallelogram 4-bar mechanism [25]. The two jibs are connected by a welded track that supports the weld head, allowing for the movement of both jibs. This synchronous pivoting motion allows the jib assembly to be stored along a wall when not being used, or when a tank needs to be positioned for welding. To move the jib assembly, the operator must manually grab the end of the jibs and pull the jibs into position, shown in Figure 52.When the entire machine is positioned higher than arms reach, a rope tied to the end of the jib assembly allows for the operator to pull and swing the jib assembly into position from the ground.

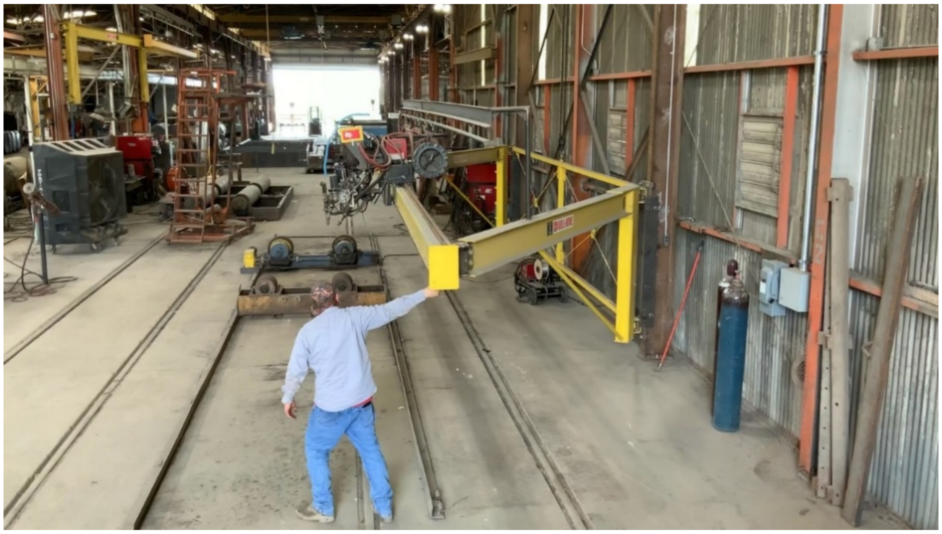


Figure 52: Pivoting motion of the jib assembly

4.1.2 *Safety Concerns*

There are three major safety concerns relating to the jib assembly rotation: over-rotation of the jib assembly, the rotational speed of the assembly, and as stated in the previous section, the position of the operator when the jib assembly is in motion. Rotational movement speeds are going to be factored into the design of the overall jib motor assembly. Over-rotation of the jib assembly itself will be addressed with limiting switches, discussed in section 5.5.8, along with safety concerns relating to the position of the operator.

4.2 Design Boundary Conditions

In this section, design boundary conditions are discussed to provide a basis for calculations. The first design constraint is the weight of the machine. The second design constraint is the speed at which the jib assembly rotates, angularly. To elaborate on these boundary conditions, a brief weight analysis regarding only the jib assembly will be discussed, the mass moment of inertia related to the rotational movement will be introduced, and an angular velocity analysis will be conducted.

4.2.1 ***Weight Analysis***

To calculate the torque needed to rotate the jib assembly, the mass moment of inertia or the measure of the jib assembly's resistance to change in a rotational direction is needed. This concept will be discussed in further detail in section 4.4.1. For this calculation, the weight of the jib assembly components are needed. The weight of the jib assembly consists of the weight of the two jibs, the weld head, and the weld track only. The weight of the full carriage is not considered for this analysis. For calculations, the weight of the jib assembly equals 2500 lbs.

4.2.2 ***Moment of Inertia***

Moment of inertia in a rotational motion is equivalent to mass in linear motion. In the same way that the force needed to give an object, a certain acceleration depends on its mass, the torque needed to give a rotating object a certain angular acceleration depends on its moment of inertia [19]. To design a method that will eliminate the need for the operator to manually rotate the connected jib assembly, the mass moment of inertia is needed. This calculation will be used to determine motor sizing and torque requirements needed to adequately rotate the jib assembly from one side to the other. This value depends on the mass of the jib assembly, its shape, and the axis of rotation. For this calculation, the axis of rotation will be located on the pivot joint where the jib assembly is connected to the carriage, denoted by the red arrow in Figure 53 below. The actual calculation of the moment of inertia will be discussed in section 4.4.

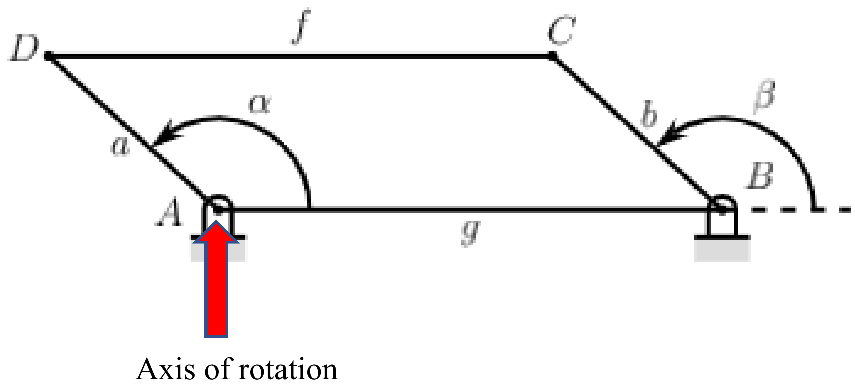


Figure 53: Axis of rotation for the moment of inertia analysis

4.2.3 ***Rotational Speed Analysis***

To appropriately select a motor for the rotation of the jib, the velocity at which the jib is required to rotate was first assessed. When the jibs are positioned for storage or rotated out into the correct position needed for welding, the velocity at which this motion occurs depends heavily on several factors. The first one is the moment of inertia of the jib assembly. The second is any friction in the pins that mount the jib assembly onto the carriage. The third is the amount of force being applied by the technician who is manually moving the assembly. Based on video reference, when operators manually move the jibs out and into position, a reasonable angular velocity, ω, was estimated to be 3 $\frac{deg}{s}$ or 0.5 RPM. This angular velocity was chosen to use throughout the analysis because it mimics the existing conditions and allows the jibs to move at a safe speed. More importantly, however, the acceleration, α, of 3 $\frac{deg}{s^2}$ is used to determine how much torque is required to reach that speed within 1 second. A piecewise function for the movement of the jib is described as follows:

For angular velocity: $$\omega_{jib} = \{0 < t < 1: 3t) \tag{32}$$

$$\omega_{jib} = \{1 < t < 35: 3) \tag{33}$$

$$\omega_{jib} = \{35 < t < 36: 108 - 0.5t) \tag{34}$$

For acceleration: $$\alpha_{jib} = \{0 < t < 1: 3) \tag{35}$$

$$\alpha_{jib} = \{1 < t < 35: 0) \tag{36}$$

$$\alpha_{jib} = \{35 < t < 36: -3) \tag{37}$$

In these equations, t is the time to reach the position between each interval, for a total of 36 seconds from start to finish. A deceleration value of $-3\ \frac{deg}{s^2}$ is used for the last time interval in the piecewise function, from 35 seconds to 36 seconds. The deceleration value is used to program the VFD which will be discussed in more detail in section 5.5.2. Again, these values are based on the current movement speeds of the existing machine.

4.3 Design Options

Given the existing movement requirements for the jib, it was necessary to determine some design options to accommodate the client's requests. The first design option was to utilize a hydraulic system to rotate the jib assembly left and right, seen in Figure 54 below. This option would allow for a maximum range of motion, large amounts of power, variable speed control, and automatic overload protection.

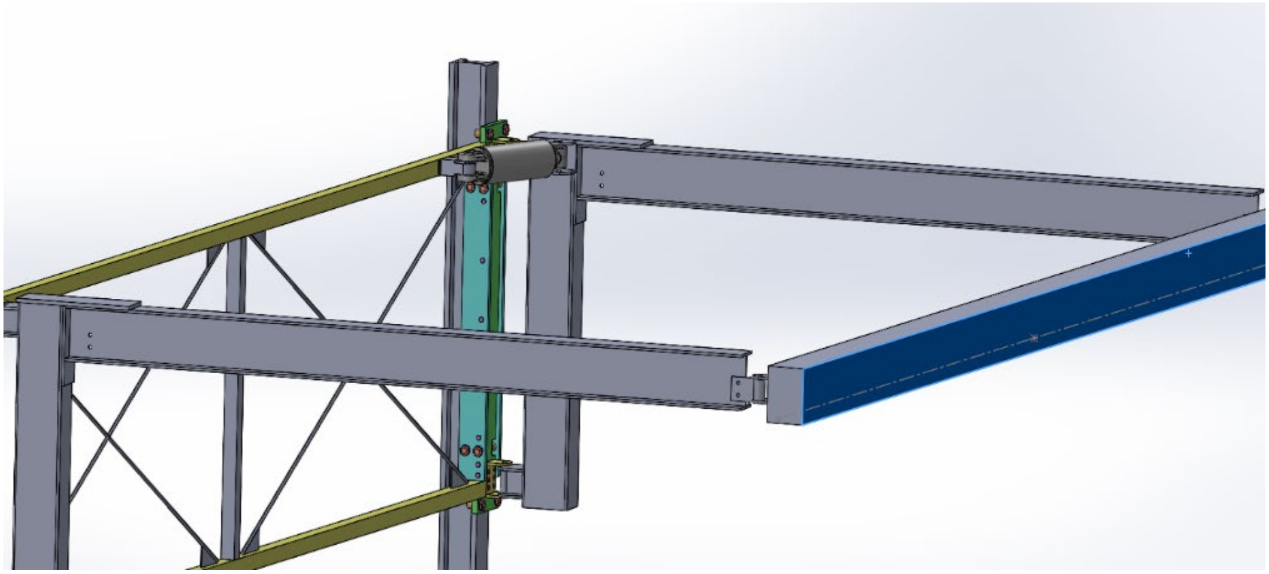


Figure 54: Option 1 - Hydraulic System

The second option was to use a rack and pinion system to move the weld head across the carriage, eliminating the jib assembly altogether. This configuration can be seen in Figure 55 below.

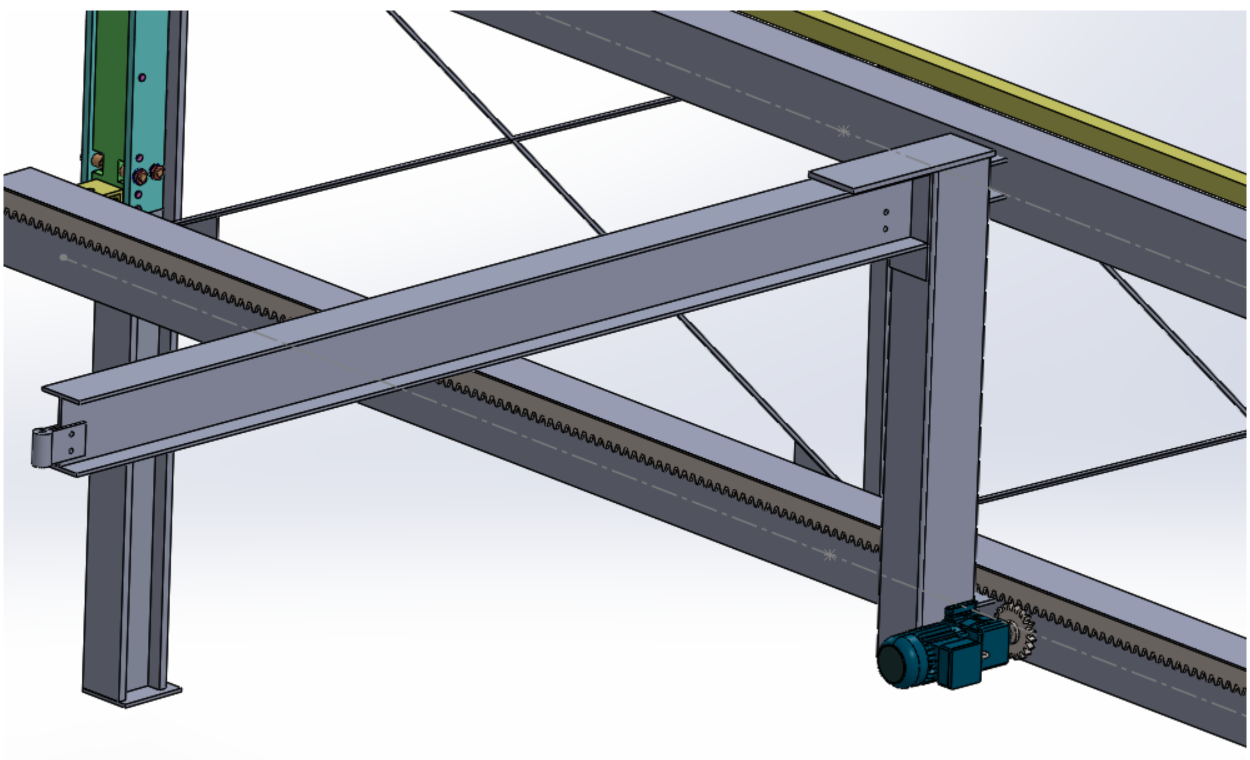


Figure 55: Option2 - Rack and Pinion

The third design option chosen to consider was a chain drive with a sprocket and motor. This option would allow for a maximum rotation distance, high torque output, and speed reduction/torque increases (via gear ratios, if necessary), along with minimal additional weight being added to the overall machine. This design option can be seen in Figure 56.

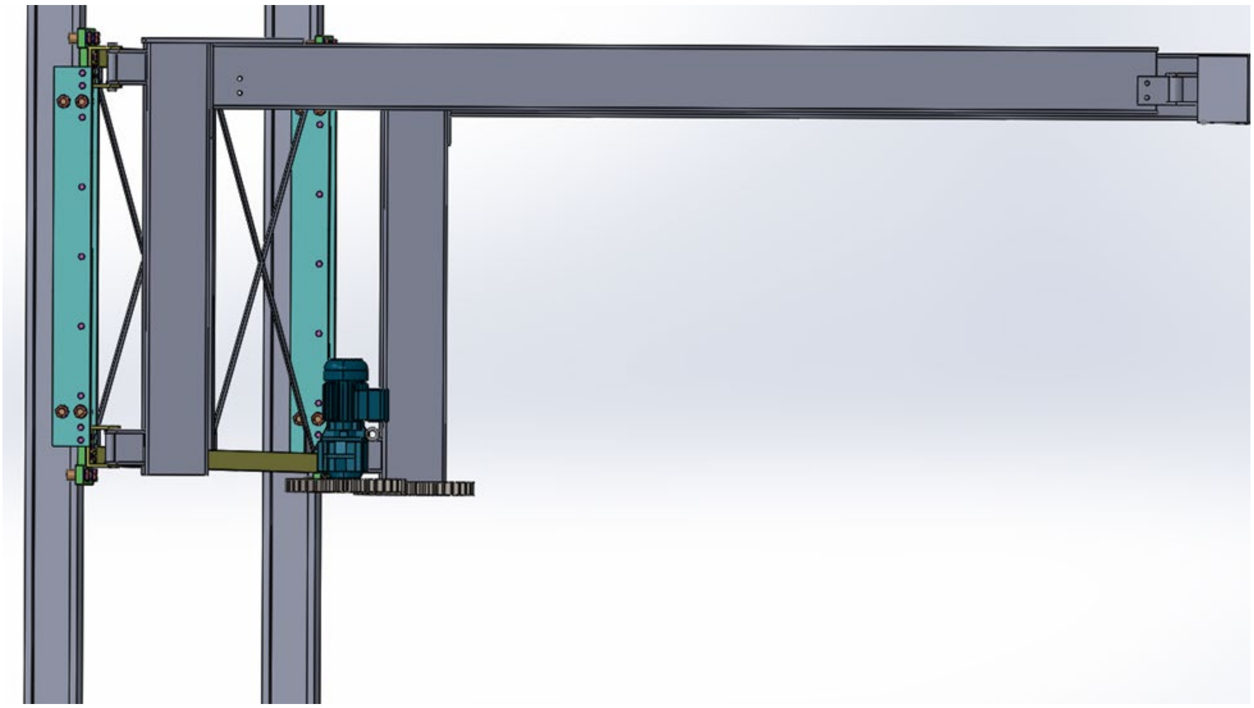


Figure 56: Option 3 - Chain drive with motor and sprocket

After extensive research on the three options, the rack and pinion system was quickly eliminated since it would not provide the type of rotational movement needed to allow the jibs to place the welder in the desired position. The hydraulic system was highly considered and would allow for the jibs to rotate as requested. But, since the shop currently does not have a hydraulics system installed, this option was eliminated as well. The hydraulics system would require an additional start-up cost and an additional amount of maintenance to the system. Since we were able to eliminate two options due to not meeting operational requirements, the design option chosen was to utilize a motor with a chain and sprocket system, like option 3. It provides the client with a system they were already familiar with using and minimal additional training would be needed to maintain and operate it.

4.4 Calculations

In this section, calculations are performed and discussed relating to the moment of inertia of the jib assembly and torque requirements for motor selection. Chain and sprocket selection, to be used in the chain drive for the jib assembly will also be discussed.

4.4.1 *Moment of Inertia*

To determine the moment of inertia for motor selection, Solidworks was used to simulate the existing machine and obtain values used in calculations. However, to validate the reliability of the values taken from Solidworks, a sample calculation was done for a problem that a solution could be solved for manually, and then both findings were compared.

4.4.2 *Moment of Inertia Validation*

The problem chosen to validate the reliability of the moment of inertia calculation in Solidworks is a simple sphere and rod assembly shown in Figure 57.

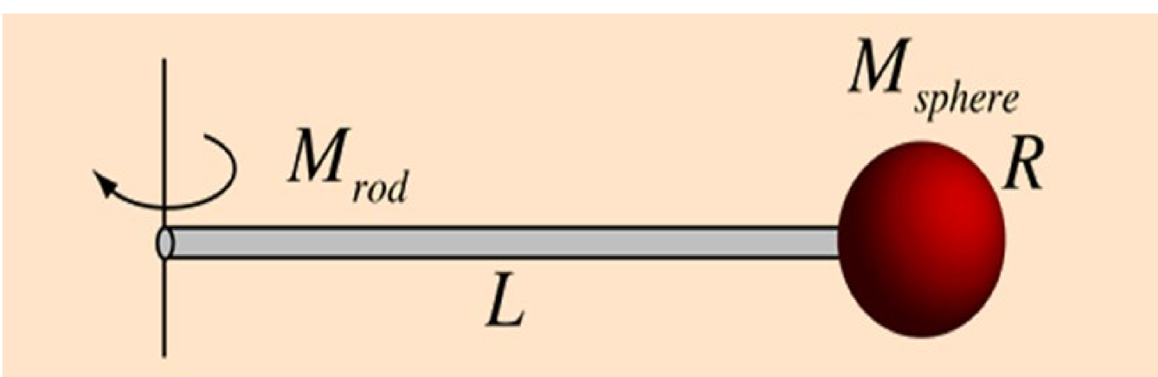


Figure 57: Rod and Sphere assembly [18]

The moment of inertia of a sphere was calculated and then the moment of inertia of a thin rod connected to the sphere was calculated. The formula used to calculate the moment of inertia for the sphere is as follows in Eq. (38), where m, is the mass of the object and R, is the radius of the sphere.

$$I_{sphere} = \frac{2}{5} m_{sphere} R^2 \tag{38}$$

The formula used for calculating the moment of inertia for the rod is as follows in Eq. (39) where m, is the mass of the object and L, is the length of the rod.

$$I_{rod} = \frac{1}{3} m_{rod} L^2 \tag{39}$$

Using the parallel axis theorem, which states that the moment of inertia of an object about an axis through its center of mass is the minimum moment of inertia for an axis in that direction in space, the moment of inertia about an axis parallel to that axis through the center of mass is given by Eq. (40) [18]. The values substituted into these equations are shown in Table 7 below.

$$Parallel\ Axis\ Contribution = M_{sphere}(L + R)^2 \tag{40}$$

Table 7: Dimensions for the rod and sphere

m_{sphere}	1.0 kg
R	0.2 m
L	0.5 m
m_{rod}	2.0 kg

Using these values into equations Eq. (38), Eq. (39), and Eq. (40) yields the following values:

$$I_{sphere} = 0.016\ kg\ \cdot m^2$$

$$I_{rod} = 0.167\ kg\ \cdot m^2$$

$$Parallel\ Axis\ Contribution = 0.489\ kg\ \cdot m^2$$

Adding these three values together results in a total moment of inertia of 0.672 kg·m2. Once this value was obtained, the rod and sphere assembly was then created in Solidworks as seen in Figure 58, with the same dimensions and masses. The mass moment of inertia analysis was used in Solidworks and returned a value of 0.672668 kg·m2. The data directly pulled from the moment of inertia analysis in Solidworks is shown in Figure 59.

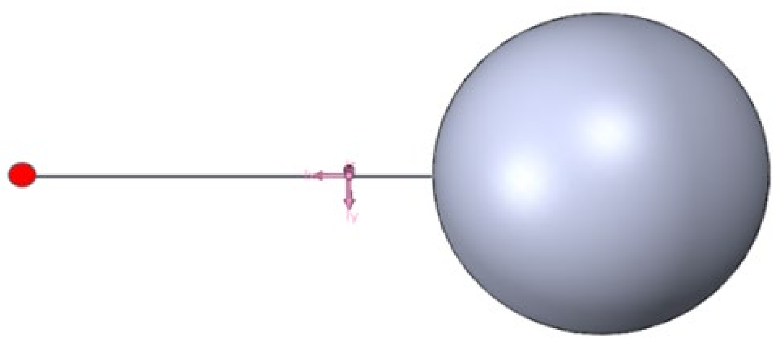


Figure 58: Solidworks model of the rod and sphere

Moments of inertia: (kilograms * square meters)
Taken at the output coordinate system.

Ixx = 0.672678	Ixy = 0.000000	Ixz = 0.000000
Iyx = 0.000000	Iyy = 0.016012	Iyz = 0.000000
Izx = 0.000000	Izy = 0.000000	Izz = 0.672668

Figure 59: Mass moment of inertia analysis from Solidworks

Knowing that the Solidworks values were reliable and produced identical values, we proceeded to use the software to calculate the moment of inertia value of the jib assembly as shown in Figure 60 . The CAD drawings in this figure show two extreme positionings of the weld head along the track connecting the two jibs. The red block in Figure 60 denotes the weld head and the red circle represents the axis where the moment of inertia will be calculated from.

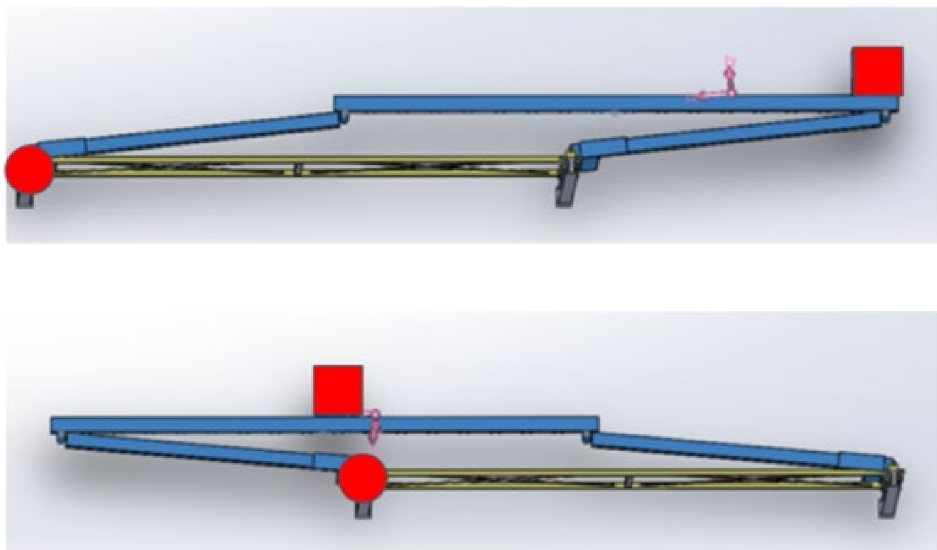


Figure 60: Extreme positioning of the weld head from the axis of rotation

Using these two positions, the maximum and the minimum moment of inertia were calculated using the same mass moment of inertia analysis tool in Solidworks. The maximum moment of inertia value when the weld head is farthest away from the origin is 61709 kg m2. The minimum moment of inertia value when the weld head is closest to the origin is 17729 kg m2. These values can now be used in the jib motor calculations.

4.4.3 *Jib Motor Calculations*

Using the jib rotational speed analysis equations and moment of inertia data from Solidworks, the maximum and minimum torque requirements were determined. The following equations were used to calculate the maximum and minimum torque:

Max Torque, τ_{max} $$\tau_{max} = \alpha_{max} * I_{assembly\_max} \tag{41}$$

Min Torque, τ_{min} $$\tau_{min} = \alpha_{max} * I_{assembly\_min} \tag{42}$$

In Eq. (41) and Eq. (42), the maximum acceleration value, α_{max} was used to calculate the torque required for jib movement. Multiple scenarios exist related to the moment of inertia, I, dependent upon the position of the weld head that moves along the 24-foot track, but the worst-case scenario min and max position moment of inertia calculations are as follows in Table 8.

Table 8: Min and max moment of inertia

α_{max}	$3 \frac{deg}{s^2}$
$I_{assembly\_max}$	61709 kg m2
$I_{assembly\_min}$	17729 kg m2

Converting degrees per second to radians per second by multiplying by 0.017453 and then substituting these values into Eq. (41) and Eq. (42) resulted in a maximum torque of 3246 N*m and a minimum torque of 932 N*m. Converting this value to horsepower yields a maximum horsepower needed to be 4.35 HP and a minimum horsepower needed to be 1.24 HP. These values represent the maximum and minimum horsepower needed when the weld head is farthest away from the origin and closest to the origin. However, on the existing machine, the weld head is positioned primarily at the far right (or left when viewed from the columns) of the weld track during jib rotation and remains stationary during this motion. An example of this can be seen in Figure 61 below. The red square represents the weld head position on the track and the red circle represents the axis of rotation, also referred to as the origin.

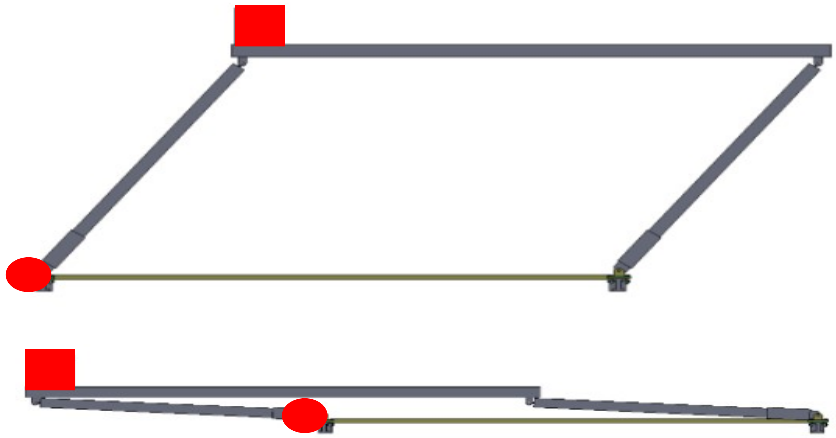


Figure 61: Realistic weld head positioning during movement

Using the more realistic positioning of the weld head assembly during the entire operation, a more accurate min and max range of the horsepower needs was determined. Operators of the machine should be instructed to use this position when rotating the jibs. From the Solidworks moment of inertia analysis tool, the values in Table 9 were obtained. Using the same equations (41) and (42) resulted in a maximum horsepower requirement of 3.5 HP and a minimum horsepower requirement of 1.24 HP.

Table 9: Min and max realistic moment of inertia

α_{max}	$3\ \frac{deg}{s^2}$
$I_{assembly\_max}$	50,000 kg m2
$I_{assembly\_min}$	17800 kg m2

4.4.3.1 Jib Motor Selection

The torque requirements to accelerate the jib assembly at a rate of 3 degrees per second with a 1:1 gear ratio means, that without friction, between 3.5 HP and 1.24 HP are required. In order to increase the service life of the motor, account for motor inefficiencies, and overcome unknown friction forces, a 5 HP motor is more suitable for this application. A 5 HP motor exceeds the 3.5 maximum horsepower by a factor of 1.42, but this factor can be doubled to 2.84 by utilizing a 2:1 gear ratio from the motor to the jib sprocket, effectively doubling the torque of the 5 HP motor. For simplicity of the overall design, since the horsepower needed for both the lifting movement and the jib rotation is the same, it was decided to use identical motors for both assemblies. Doing so could potentially alleviate maintenance issues that would arise from having two different types of motors to order, install, and maintain.

Given the motor calculation from section 4.4.3, the maximum angular velocity of the motor that was selected is 19 RPM. However, at this rate, this will cause the jib to rotate too fast, and not allow it to be safely controlled. To overcome this, programable settings using the variable frequency motor drive (VFD) required by the client, discussed later in section 5.2.2, can reduce the angular velocity of the motor shaft and sprocket, with an additional reduction in angular velocity from a 2:1 gear ratio to the jib sprocket. Utilizing this combination of VFD controls and gear reduction, a final RPM of the jib assembly can be reduced to 3 deg/s or 0.5 RPM. Using the variable frequency drive, the system can be fine-tuned with increased torque to account for any loss of rotation due to friction or wear on the rotating components of the machine over time.

If an inexpensive motor is desired that has lower torque, redesigning the chain and sprocket set up to increase the overall gear ratio would be required. Currently, the 2:1 gear ratio of the sprockets doubles the torque being applied to the jib as the small sprocket moves 2 times for every 1 time that the large sprocket rotates. As the gear ratio is increased, the angular velocity of the jib assembly decreases, but torque increases. This flexibility in design should help with the long term maintenance of the system in the event components must be replaced. To reduce the amount of strain on each motor assembly, proper maintenance procedures will be discussed in section 9.10.

4.4.4 *Chain and Sprocket Calculations*

To determine the chain and sprocket combination for the jib assembly movement, the process from section 3.4.4 was applied.

4.4.4.1 Chain and Sprocket Selection

There are 14 different ASME/ANSI B29.1 standard roller chain sizes. Using the horsepower of the motor and the RPM of the small sprocket, ANSI standard chain, size #120, was selected from the quick selection chart from section 3.4.4. Matching carbon steel sprockets that are hardened via shot peening can be selected. Two sprockets are required, where the small sprocket has half the number of teeth as the large sprocket. According to ANSI B29.1 standards, the manufacturer's recommendation for sprockets on a #120 roller chain is the 120B10 and 120B20, as seen below in Table 10.

Table 10: Manufacturer Recommended Sprocket Sizing

Small Sprocket	
Small Sprocket Size:	120B10
Tooth Count	10
Outside Diameter	5.52”
Large Sprocket	
Large Sprocket Size	120B20
Tooth Count	20
Outside Diameter	10.37”
Technical Info	
Gear Ratio	2:1
Chain Length	90”
# of links	60
Sprockets have been hardened and have a fully black oxide coating, increasing the resistance to wear and early corrosion from the environment.	

4.5 Final Design of the Jib Rotation Improvement

The final design for the motor assembly can be seen in Figure 62 below. The selected motor, chain, and sprockets have been added to the design. It can be concluded that after calculating the torque required to move the jib assembly into position for welding, the motor, sprockets, and chain in combination with the VFD, are acceptable components for rotating the jib assembly safely. The validation done through hand calculations and Solidworks increases the confidence that the solutions mentioned for the jib assembly will be viable. An additional safety measure will be recommended to reduce the chances of accidental contact with the chain assembly. A chainguard is recommended to be placed over both sprockets and the roller chain connecting the two sprockets. This guard should completely cover the chain and sprocket assembly, to prevent accidental injury. Not only is this improvement a beneficial addition from the standpoint of safety, but it is also helpful in reducing the amount of dust and abrasive airborne contaminants that could prematurely end the life of the sprocket and roller chain. As the chain contacts the sprockets, any contaminants that are introduced have the potential to cause wear and tear on the equipment that would be unexpected.

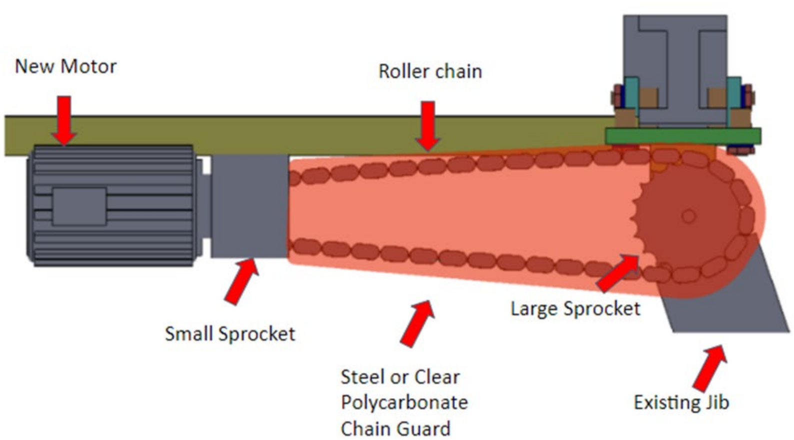


Figure 62: Final Jib rotation assembly design

5 Remote Control

In this section, the addition of a remote-control system to the existing machine will be discussed. First, the existing conditions of the machine are provided along with safety concerns that need to be addressed with the proposed design. Next, several design boundary conditions set by the client will be explained, and based on these constraints, two design options will be evaluated on the applicability of each to determine the best design solution. Then, components and wiring that were chosen for the design will be shown along with explanations of how these components will work together. Lastly, a conclusion will be presented, reporting the validation of the design.

5.1 Existing Conditions

5.1.1 ***Manual Movement***

The first functional movement of the weld head manipulator involves manually rotating the jib assembly in the horizontal direction by an operator as discussed in section 4. Moving the manipulator's jib assembly requires that the operator be physically capable of pushing or pulling the assembly to the desired operating position. The pair of jibs combined with the weld track, the weld head, and all the tubing/wiring necessary for the weld head operation weighs approximately 2500 lbs. The jib assembly can be seen again in Figure 63 below. The operator must be capable of rotating the heavy system from the ground but, while doing so, the operator is in a dangerous position underneath the machine. If the machine were to have a serious failure such as chain failure, the position of the operator beneath the machine could result in serious injuries.

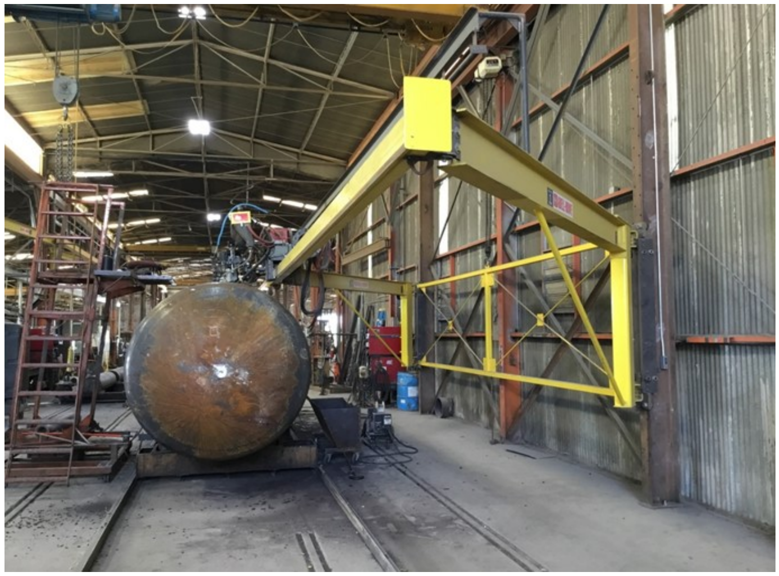


Figure 63: Jib Assembly

The second movement uses a hoist to lift and lower the manipulator in the vertical direction as discussed in section 3. To move the weld head manipulator in the vertical direction, the operator must use the current hoist system and a push-button control which is currently located on the column beneath the entire machine. While doing so, the operator must also manually insert safety pins placed in pinholes within the two existing outer columns as discussed in section3.1. To address these concerns, a remote-control system is proposed to be added to the machine to eliminate the need for manual movement.

5.1.2 ***Safety Concerns***

As stated in the previous section, the operator is responsible for moving the jib assembly in the horizontal direction, from left to right, to get the weld head in the desired position for welding. The main safety concern from the client with the existing setup is that the operator must be positioned beneath the machine to move it to its operating position. This puts the operator in a dangerous position in the event of chain failure from the hoist. The second safety concern from the client is to have a way to stop the movement of the machine in the event of an emergency. These safety concerns will be addressed and considered in the chosen design of the remote-control system.

5.2 Design Boundary Conditions

The design boundary conditions for the remote-control addition are set by the client. There are three constraints to the boundary conditions. The first constraint is having a system that is easily operated, requires little training and, maintenance on the system can be performed by current maintenance employees. This will eliminate the need to use outside resources for repairs. The second constraint is to be able to use a variable frequency drive (VFD) to control the motor and chain assemblies. The third constraint is to have two remote controls that operate the same functions of the machine but from two separate locations. These remotes cannot be operated simultaneously.

5.2.1 *Ease of Operation and Maintenance*

In accordance with client requests, the remote-control system must be easy to use and require little training to learn how to operate. The proposed system should be similar to the controls currently used in the shop that rotates rollers for the tanks being welded. This would result in a smooth transition from the current operations to the new machine operations. This requirement is important to minimize the disruptions to production schedules currently in place. It is also desired to be able to maintain the system with the current on-staff maintenance employees rather than having to use resources from outside the company.

5.2.2 *Variable Frequency Drive*

The second requirement requested by the client is that the remote-control system must utilize a VFD to control both the vertical lift movement and the horizontal movement of the jib assembly. Taking advantage of a VFD for the improvement to the weld head manipulator will allow the machine motors to be controlled with programmable settings within the VFD. The client is currently using VFD systems within the shop to operate other machinery, so there is the current knowledge of maintaining and programming by on-staff personnel. This eliminates the need to use resources outside of the company to maintain, troubleshoot, and repair the system.

5.2.3 *Remote Locations*

The third requirement from the client is that the controls for the manipulator movement be placed in two separate locations within the work area, with identical remote-control setups. One remote will need to be located on what will be referred to as the "ground" and another location that will be referred to as the "sky". The remote-control located on the ground can be used anywhere in the vicinity of the machine from ground level. It will be suspended from the ceiling, allowing for the operator to move around the machine to observe movement. The remote control for the sky will be located atop what the operators call the bird's nest. This is essentially a rolling ladder platform that the operator stands or sits on while operating the weld head above the tanks when welding, as seen in Figure 64 denoted by the red arrow.

The movement of the manipulator system will need to be controlled non-simultaneously from either one of the locations. To accomplish this, a switch will have to be utilized to allow the operator to choose which remote location to activate. Having the identical remote-control located atop the bird's nest enables the operator to maneuver the machine without having to come down from the platform. This could potentially save many man-hours during the manufacturing process.

Figure 64: Bird's Nest

5.3 Calculations

There were no calculations used to determine the components of the remote-control system due to the generally low level of complexity. The compatibility of all the components is the main concern when designing, and it is not expected that improvised wiring methods, requiring complex electrical theoretical background knowledge, will be used. All the components must be rated for the same maximum input voltages and amperages and must be capable of handling the power needs of the motors. The input voltages currently used in the shop are 240 volts. This is a concern related to safety because a mismatch in equipment components could lead to overloading the circuits, causing unexpected shutdowns or damage. VFDs offer programable input controls utilizing low voltage digital or analog communication wires. VFD selection will need to take into consideration whether analog or digital control methods are to be implemented.

5.4 Design Options

When considering options for remotely controlling the motor and chain assemblies using the paired VFD, there were two options considered: a wireless remote-control or a direct-wired remote-control. There are advantages and disadvantages of each setup shown in Table 11 below.

Table 11: Advantages & disadvantages of two types of remote-control systems

Design Option	Advantages	Disadvantages
Wireless Remote Control	• Free-range of motion • Ideal for this application • Wide range of control options	• Expensive • No variable speed • Requires more training • Third-party maintenance • Relies on a battery/transmitter
Direct Wired Remote Control	• Cheap • Easy installation • In house maintenance • Little to no training	• Limited range of motion • Components will have to be replaced periodically due to wear and tear

The first design option, a wireless remote-control box that is commonly used for large crane control is shown in Figure 65. This wireless remote control is ideal for this type of application because it can be used anywhere within the vicinity of its transmitter. This would give the operator a free range of motion within the shop to control the machine. This type of remote is also capable of a wide range of control options depending on the application.

Figure 65: Wireless Remote Control Used for Crane Operation [1]

One disadvantage of this remote control is that when the battery is low, the power to the machine is cut off. It would rely on the operator to maintain the charge of the battery in the remote control and, if the remote control was to malfunction, the machine could only be controlled with the drive itself. A second disadvantage is that if it were to malfunction, the maintenance on it would require outsourcing it to a specialized repair shop. A third disadvantage is that the types of remote-controls that have the functionality required are expensive and many of them do not have variable speed controls.

The second design option is to wire the remote directly to the drive. The two main advantages of this option are that it can be easily maintained in-house and is operated by only two controls: a push button to control the direction of the motor and a dial to control the speed of the motor. The disadvantage of this option is that since it is directly wired to the drive, there is a limited distance where the operator can be located when using this control. After considering the different design options, it was determined that the best design decision for the client would be to choose the direct-wired remote-control system since the wireless remote-control option does not allow for a variable speed option, requires third-party maintenance, and would require more training than desired by the client.

5.5 Final Design Components

The following sections discuss components that have been chosen for the final design of the remote-control system. These include a variable frequency drive (VFD), switches, relays, remote-controls, and an electrical panel. At the end of this section, a wiring schematic will be provided, detailing the connections of all the components needed to operate the drive externally from different locations.

5.5.1 *Motor Drive Selection*

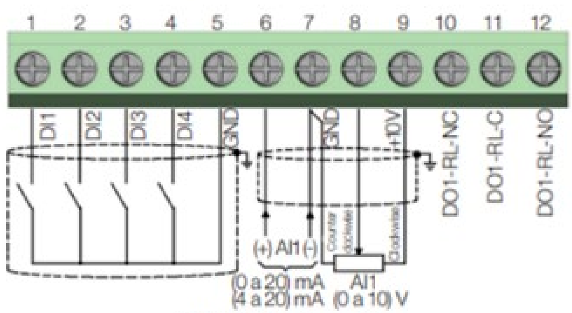
To control motors in a manufacturing setting where high torque and low speeds are required, variable frequency drives (VFD) are most often used. They provide precise control of the voltage to hertz ratio for a motor and allow for local control on the drive control panel or remote control from a different location. The drive chosen to use for both the carriage lift and the jib rotation is the WEG Electric CFW300 series AC micro variable frequency drive, manufactured by WEG Industries shown in Figure 66.

Figure 66: WEG variable frequency drive [3]

This type of drive is used in electro-mechanical drive systems to control AC motors. It controls the speed and torque by varying the frequency and voltage. This drive has a maximum horsepower of 5 HP and an input voltage of 200-240 Volts. This type of drive will be able to run and control the motors that were chosen for the lifting operation and the jib rotation operation. Two separate drives and two separate motors will be used but will be controlled by one remote control.

5.5.2 *VFD External Controls*

One of the features of the drive mentioned previously is the ability to remotely control from a location other than the panel on the front of the drive. To do this, the VFD programming options are changed according to manufacturer instructions to allow for external controls. This installation method allows for push-button up/forward and down/reverse controls, as well as variable speed through Al1 (pin 8) from a potentiometer, as seen in Figure 67.



(a) NPN Configuration

Connector		Description (*)
1	DI1	Digital input 1
2	DI2	Digital input 2
3	DI3	Digital input 3
4	DI4	Digital input 4
5	GND	Reference 0 V
6	AI1	Analog input 1 (Current)
7	GND	Reference 0 V
8	AI1	Analog input 1 (Tension)
9	+10 V	Reference +10 Vdc for potentiometer
10	DO1-RL-NC	Digital output 1 (NC contact of relay 1)
11	DO1-RL-C	Digital output 1 (Common point of relay 1)
12	DO1-RL-NO	Digital output 1 (NO contact of relay 1)

(*) For further information, refer to the detailed specification in Section 8.2 ELECTRONICS/GENERAL DATA on page 32.

Figure 67: VFD Control wire inputs [6]

Digital input pins 1 and 2 are programmable to operate the forward/up and reverse/down operations through the VFD. An additional feature that is commonly implemented using the input wires at this connection point of the VFD is an interrupt to pin 5, ground, to stop all motor movement instantly. As explained later, in section 05, this is useful for an emergency stop push button.

5.5.3 *Ground/Sky Selectable Switch*

A client request for the external control of the VFD with two remote controls is that there cannot be simultaneous operation from both remotes. The intended locations of each remote are as follows: One remote will be located at, what is referred to, as the "Sky location" while the other remote will be located at, what is referred to, as the "Ground location". To achieve isolation of the sky and ground controls, a rotary selector switch will be mounted near the ground controls which allows users to select between the operation of the ground remote and sky remote [30]. This type of switch is shown in Figure 68 below.

Figure 68: Rotary Switch [30]

Due to the complexity of wiring involved in externally controlling the VFD using input wires described in section 5.5.2, the selector switch will supply low voltage 12VDC power to relays, as described in sections 5.5.4 and 5.5.7.

5.5.4 *Electric Panel*

The selectable switch in section 5.5.3 should be mounted into its enclosure, or into the electrical panel that will house the wiring and relays that are needed to operate the system. In addition to housing the wiring and relays for the operation of the system, an additional 110v standard power outlet is to be installed as seen in Figure 69.

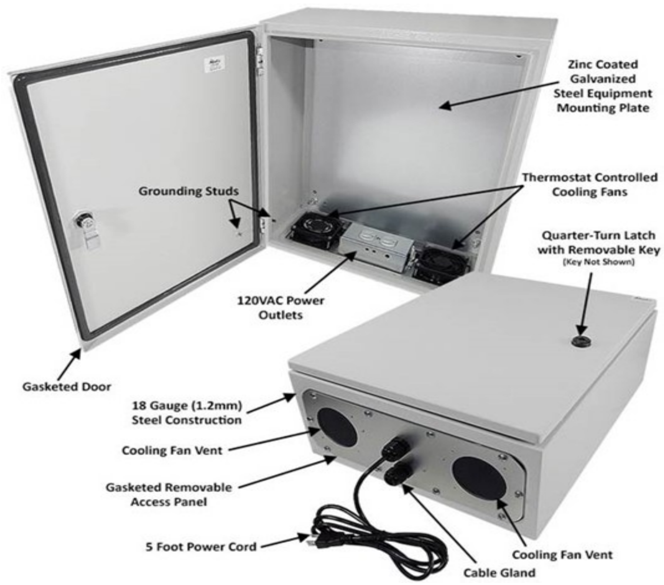


Figure 69: Electrical Panel [28]

Through an AC/DC power converter, 12VDC power can be supplied to the relays, which are needed to drive the isolation of the sky and ground controls. Additional relays would be used to control the selection of the jib VFD and the lift VFD, as mentioned in section 5.2.2, from one remote control. For clarification, the ground and sky remote controls will each be able to control the lift VFD and jib VFD, and the selectable switch at the electric panel selects which remote is active. This electric panel will not replace the panel that is required to be installed by a licensed electrician for powering the VFDs.

5.5.5 *Remote-Control Design for the Ground Location and Sky Location*

As stated earlier, the weld head manipulator is required to be operated from both the ground and the sky locations. This will require two identical remote-control housings that have the functionality of push-button controls, speed control potentiometers, e-stops, and rocker switches. An example of the remote-control design can be seen in Figure 70 below.

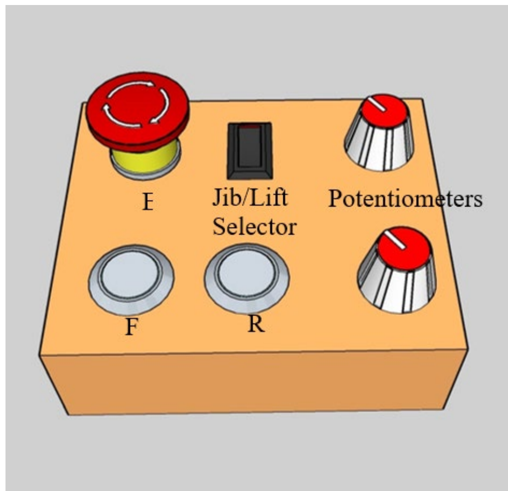


Figure 70: Remote Control Interface

To operate the lifting motion of the manipulator, the rocker switch will need to be pushed into the lift position to engage the VFD that controls the lifting motor. The rocker switch is a heavy-duty electrical switch that has two positions, on or off. An example of the rocker switch that is used is seen in Figure 71.

Figure 71: Rocker Switch [2]

Once the rocker switch has been put in the lift position, the yellow pushbuttons will control the direction of the motor. The button with the up arrow will lift the carriage vertically to the desired height. The button with the down arrow will lower the carriage back down. The type of pushbuttons chosen to utilize for this application is a momentary push button shown in Figure 72, which has spring return functionality within the push button that allows the profile to return to the original position. It also is ideal for automotive industries, factories, and automation lines.

Figure 72: Push Button Control [27]

If the operator wishes to speed up or slow down the movement of the lifting motion, the speed potentiometer can be turned towards the right to increase the speed or turned towards the left to decrease the speed. The position of the wiper located inside of the potentiometer along a resistive track determines the output voltage between 0 and 10v. An example of the wiper and resistive track determines the output voltage between 0 and 10v. An example of the wiper and resistive track can be seen in Figure 73 below. The outer pins labeled GRD and +Vcc are used to connect the ground wire and the input voltage wire. The middle pin labeled output will produce the output voltage. When the potentiometer is wired in this configuration, the circuit in Figure 74 is created. the input voltage wire. The middle pin labeled output will produce the output voltage. When the potentiometer is wired in this configuration, the circuit in Figure 74 is created.

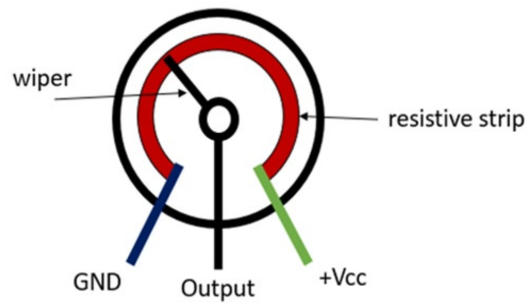


Figure 73: Potentiometer resistive strip [23]

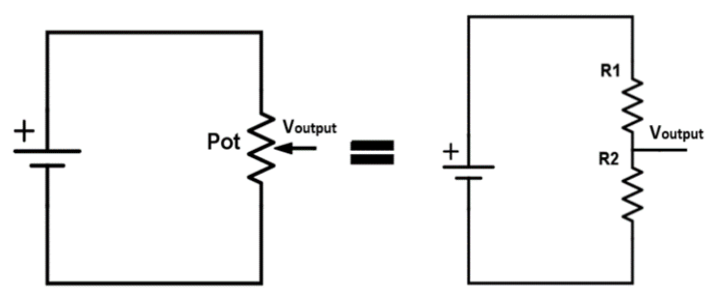


Figure 74: Potentiometer Schematic [23]

Ohms Law is used to determine the output voltage by multiplying the input voltage by the ratio of the resistances as shown in Eq. (43), where V_{out} is equal to the voltage coming out of the potentiometer, V_{cc} is the voltage coming into the potentiometer, R_1 is the resistance on the left side of the wiper and R_2 is the resistance on the right side of the wiper.

$$V_{out} = V_{cc} * \frac{R_2}{R_1 + R_2} \tag{43}$$

To operate the jib rotation motion of the manipulator, the rocker switch will need to be pushed to the jib position to engage the VFD that controls the jib assembly rotation. Once the rocker switch is put into the jib position, the yellow pushbuttons will control the direction of the motor. The button with the up arrow will rotate the jib assembly to the right to the desired position horizontally. The button with the down arrow will rotate the jib assembly back to the left. If the operator wishes to speed up or slow down the movement of the rotational motion, the speed potentiometer can be turned towards the right to increase the speed or turned towards the left to decrease the speed.

To address the client's safety concern of having a way to stop the motors in the event of an emergency, an emergency-stop button seen in Figure 75, will be installed on both the sky remote-control and the ground remote-control. This is a typical method that is used by automation professionals to stop motors instantly with a trigger. The emergency-stop buttons on the remote-controls will be installed according to ISO 13849-1, which specifies requirements for the design and implementation of safety-related control systems of machinery (International Organization for Standardization, 2015).

Figure 75: Emergency Stop Button [31]

5.5.6 *Remote-Control Schematic Diagram*

The wiring for every component within the system must be carefully arranged to maximize the safety of its users. A wiring schematic diagram for the complete remote-control system can be seen in Figure 76.

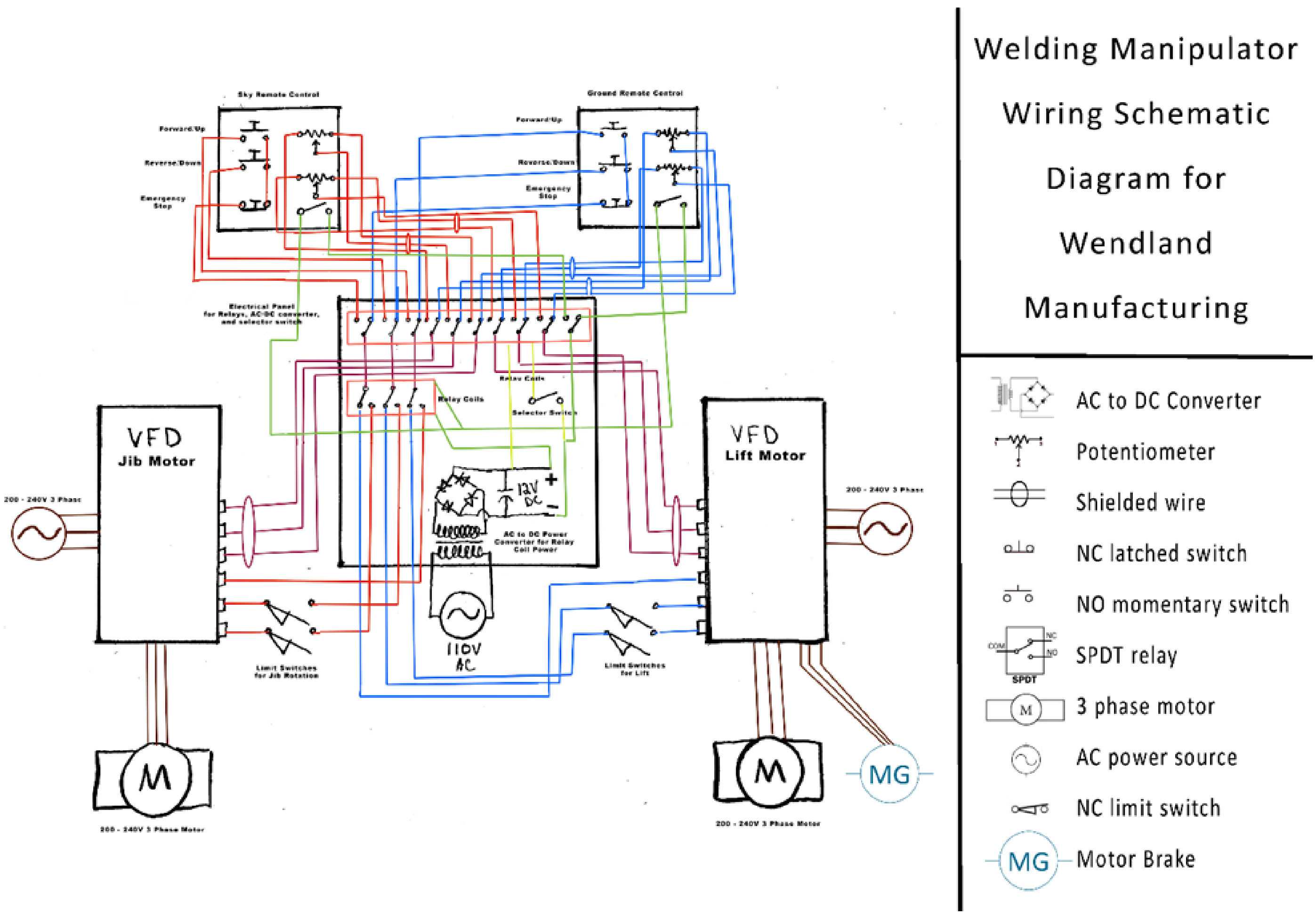


Figure 76: Wiring Diagram for Weld Head Manipulator

5.5.7 *Relays and switches*

Operating the machine must only be possible with one remote at a time. Using a series of relays, this can be accomplished. Because there are two remotes, single pole double throw (SPDT) relays are optimal. SPDT relays have a single input and 2 outputs. When unenergized, a relay connects the input to the output one, also known as the normally closed pin. When energized, an electromagnetic coil moves a contactor to connect the input to output two, or the normally open pin. Energizing a relay can be accomplished using low voltage, direct current (DC) electricity, obtained with an AC to DC power converter inside of the electric panel [32]. An example of the relays that are going to be used is shown in Figure 77.

Figure 77: SPDT Relays [32]

Activating the rotary selector switch at the electric panel mentioned in section 5.5.3, allows 12VDC to flow into the relay's electromagnetic input pins, thus switching the control from the ground to the sky remote. As only one remote can be used at a time, this disables the use of the other remote in the system. Additional usage of relays comes into play when the user wants to operate the lift motor or the jib motor. From each remote, a rocker switch is used to power a separate pack of 3 relays, switching the input signal wires from the jib motor VFD to the lift motor VFD. The system is complex, however, utilizing standard 12VDC relays found in automotive applications, provides a reliable switching method that can easily withstand repeated switching, as demonstrated in automotive electrical circuits in most vehicles on the road today.

5.5.8 *Limit switches*

As seen in the wiring diagram, Figure 76, limit switches are located in between the electrical panel relays, and the VFDs. Each set of limit switches disconnect the common wire at the VFD that runs the up/forward and down/reverse controls, preventing operation in the direction that was triggered. An example of a momentary hinged roller switch can be seen in Figure 78 on the left, however, a custom limit switch enclosure can be made to protect the switches and have adjustable limit positions to account for chain stretch as seen on the right.

Figure 78: Limit Switch and Limit Switch Enclosure [33]

These enclosures allow for operation within a contained area, usually with a threaded rod inside, and 2 limit switches on each end of the threaded rod. The rotation of the threaded rod drives a contact up and down the rod, eventually tripping the limit switches at either end of the threaded rod [33]. The rotation of the threaded rod is directly synchronized with the drive motor via a chain drive so that each revolution of the motor can be related to the rotation of the threaded rod. For example, if there is a chain drive gear ratio from the drive motor to the limit switch enclosure of 10 to 1, then for every 10 revolutions of the drive motor, the threaded rod will rotate 1 time. If the threaded rod has 20 thread rotations per inch of rod length, then it will take 20-rod rotations for the contactor to travel 1 inch along the rod length. The 10 to 1 ratio combined with the 20 thread per inch ratio means that for every 200 drive motor revolutions, the contactor will travel 1 inch of rod length. With this information known, limit switch positions can be precisely calibrated to allow for clearances at the top and bottom of the machine that is being lifted, on in the case of rotation, prevention of rotation past a certain point based on the number of motor revolutions.

When a limit switch is engaged due to the user operating the machine up to the limit that was calibrated, movement in the opposite direction remains functional, thus allowing the operator to resume movement away from the limit position without disruption. It is worth recognizing that the emergency stop that is located on each remote, in effect, acts as a limit switch by breaking the common wire that feeds the forward/up and reverse/down wires. The emergency stop buttons located on both remotes will be active 100% of the time of operation to the machine. This allows for multiple operators, at separate locations, to stop the machine at any moment. If one operator sees a harmful situation unfolding and does not have time to alert the other operator, they can still stop the machine from the E-Stop buttons on an otherwise inactive remote. In addition to stopping the machine during a dangerous situation, the E-Stop can also be engaged in the event of a button malfunction within the remote, allowing the operator to stop the machine [31].

5.5.9 ***Power supply***

Each VFD is powered by a 220VAC power source, but the power to the relays within the electric panel requires 12VDC power. 12VDC is widely available through AC/DC power converters such as laptop charges, alarm clock power supplies, automotive battery chargers, some cellphone chargers, and other electronic devices in the home. For more permanent applications, power supplies can be hardwired into a 110VAC source and converted to 12VDC. As seen in

Figure 79, this power source is visible in the electrical panel, represented by an inductor and rectifying diode circuit. A low amperage supply that would be appropriate is seen in Figure 79. This power supply produces 4.2amps max of current at 12VDC while taking 110VAC power as a source.

Figure 79: AC/DC Power Supply 12VDC@4.2A [34]

5.6 Conclusion

To provide solutions for all the requests made by the client for the remote-controlled setup, it was ideal to use a similar control system through a VFD, already implemented within the company. This allows the client to continue workflow with little to no loss of production time due to extra training. It provides client satisfaction that technical issues can be handled by on-staff personnel. The remote setup offers a simple but effective use of space while also being able to control the entire unit. The VFD system allows the client to configure the controls in accordance with their needs while controlling the machine from two separate locations. It addresses the safety concerns to remove operators from dangerous positions with the capability of completely shutting down the machine in emergencies.

6 Cost for Proposed Improvements

The cost for the proposed improvements to the welding manipulator is shown in Table 12 below. These costs are just an estimate and could change depending on the price of steel or inflation of parts prices. To account for taxes and shipping costs, the total cost of the improvements should be rounded up to approximately $16,000.

Table 12: Cost analysis of improvements to the welding manipulator

Item	Cost per unit	Quantity	Total Cost
Control panel	$595.95	1	$595.95
Push button enclosure	$64.00	2	$128.00
Motor	$2,641.80	2	$5,283.60
Motor brake	$911.92	1	$911.92
Roller chain (10 ft)	$484.00	4	$1,936.00
120B20 sprocket with stock bore	$161.86	1	$161.86
120B10 sprocket with stock bore	$71.56	8	$572.48
A36 10 x 30 C - Channel (20 ft)	$702.00	2	$1,404.00
A36 5/16" Steel Plate (20ft)	$800.00	1	$800.00
Variable frequency drive	$334.00	2	$668.00
Limit switch enclosure	$42.00	1	$42.00
Power supply	$100.00	1	$100.00
Buttons, switches, relays, limit switches	$300.00	1	$300.00
3 phase wiring (100 ft)	$100.00	1	$100.00
Drag chain	$20.00	7	$140.00
Shock Absorber	$99.00	2	$198.00
Pillow block bearings	$100.00	2	$200.00
Safety latch spring	$40.00	1	$40.00
Chain guard	$308.00	1	$308.00
Total	**$7,876.09**	**41**	**$13,889.81**

7 Future Improvements

The design concepts presented to Wendland Manufacturing for improving the current welding manipulator include new systems that can easily be incorporated into the existing machine. These concept solutions provide operational improvements and expanded safety measures that will contribute to the overall advancement of Wendland Manufacturing in the pressure vessel industry. Some additional improvements could be incorporated into the machine that would expand the safety measures proposed. The first idea is to install a proximity warning system that would detect the presence of workers in the vicinity of the machine. The proximity sensor would not allow the machine to be operated until the worker was clear of the area. The second idea is to install movement indication lights to visually show all workers in the area that the machine is in motion. These lights could correspond with the movements of each part of the machine with different colors. An expansion on this idea is to also incorporate a delay so that when the operator pushes one of the control buttons on the remote, the corresponding light would illuminate, but the motor would not immediately begin running. This would allow for anyone that was in the vicinity of the machine time to move. The delay could be set to the desired interval, shortening, or lengthening the amount of time between the initial button push on the remote control from the operator and the signal for the motor to engage. Along with this delay, audible beeping sounds could be added that would also give a warning to workers that the machine is about to be put into motion.

8 Prototyping

8.1 Introduction

To demonstrate the improvement concept design for the full-scale welding manipulator located at Wendland, a small, not-to-scale prototype was constructed. There are five main functional areas of the prototype. Each of these areas is color-coded in Figure 80. The first functional mechanism of the prototype is the lifting mechanism that raises and lowers the carriage along the length of the vertical columns. In Figure 80 below, this assembly is denoted in red. The second functional mechanism of the prototype is the jib rotation mechanism, denoted in green. The third mechanism included in the prototype is the safety latch concept, denoted in yellow in the center of the carriage. The fourth mechanism is an example of a limit switch enclosure, denoted in orange, which is similar to what is used currently in the garage door industry. The final mechanism demonstrated in the prototype is the VFD that controls the limit switch enclosure also denoted in orange. The lifting and jib rotation mechanisms are controlled by one remote and the VFD/limit switch enclosure is controlled by a separate remote. Due to difficulty in finding small three-phase AC motors to use for lifting and rotating the machine through the VFD, there are two separate demonstrations on the prototype. The first demonstration is the lifting and jib rotation assemblies shown in green and red using DC motors. The second demonstration is the VFD and limit switch enclosure shown in orange, using a three-phase 1750 RPM motor. The construction of the prototype will be discussed in the following sections.

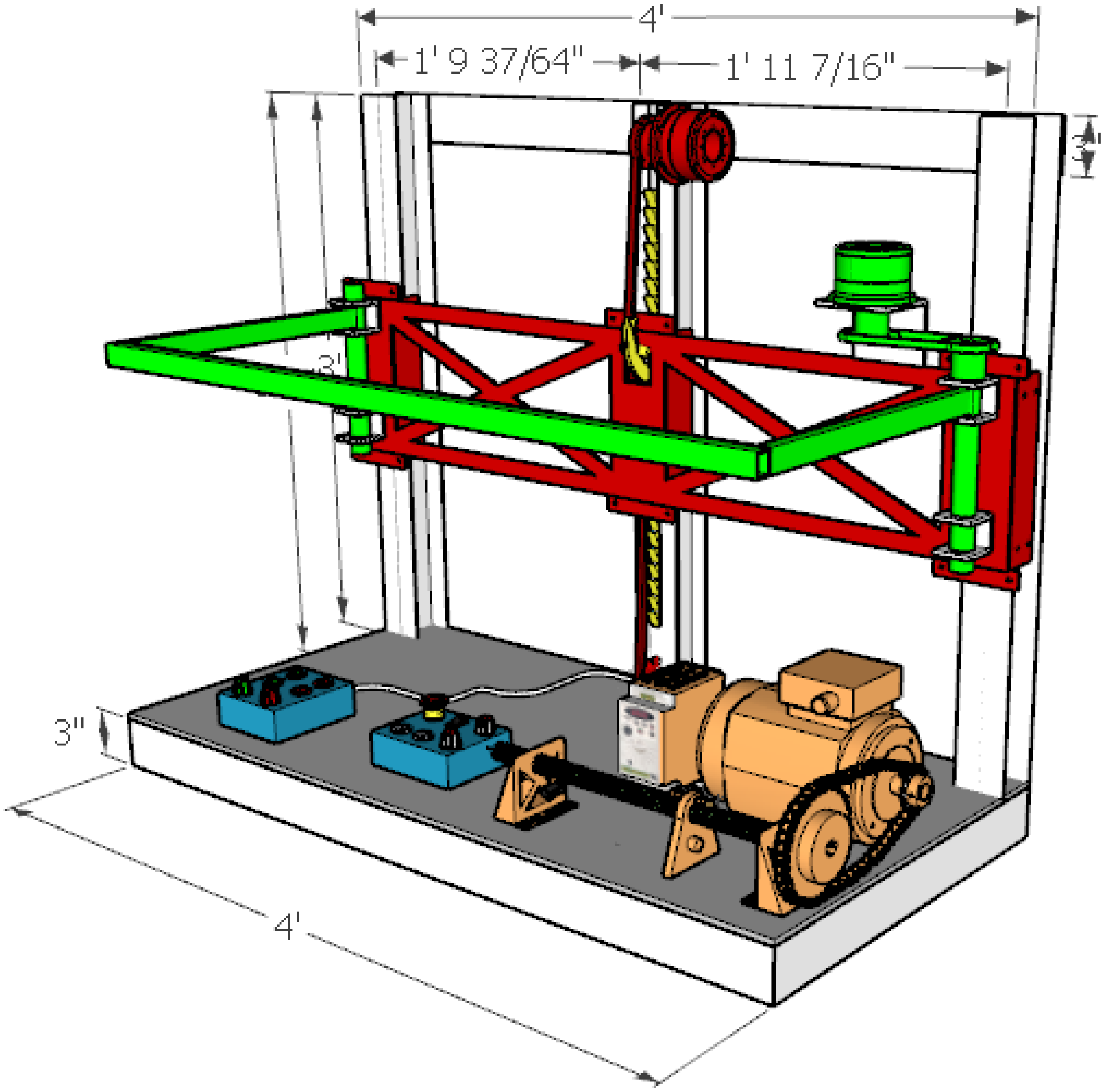


Figure 80: Full prototype model where the red denotes the lifting mechanism, green denotes the jib rotation mechanism, yellow denotes the safety latch mechanism, and orange denotes the limit switch enclosure and the VFD

8.2 The Base

The base of the prototype is the primary frame that provides a mounting surface for all the components that are being demonstrated, shown in Figure 81. It is constructed of 3” x 3” steel square hollow tubing, welded together to form the bottom frame. The footprint of the bottom frame is 2 feet wide by 4 feet long.

Figure 81: Completed base of the prototype

Along the backside of the bottom frame, two custom-made 3-foot-tall, I-beam columns are welded to the top-side corners of the frame, made of 10-gauge steel. The center column, shown in Figure 82, is constructed of 10-gauge steel, bent into shapes that resemble c-channels, and then joined together by a front plate. This built-up column is similar to what would be used on the full-scale model. On top of the base, a plywood sheet is added to provide a mounting surface for the VFD and limit switch enclosure demonstration. Four casters were then installed along the bottom side of the base to allow for easy maneuvering when moving from one location to another.

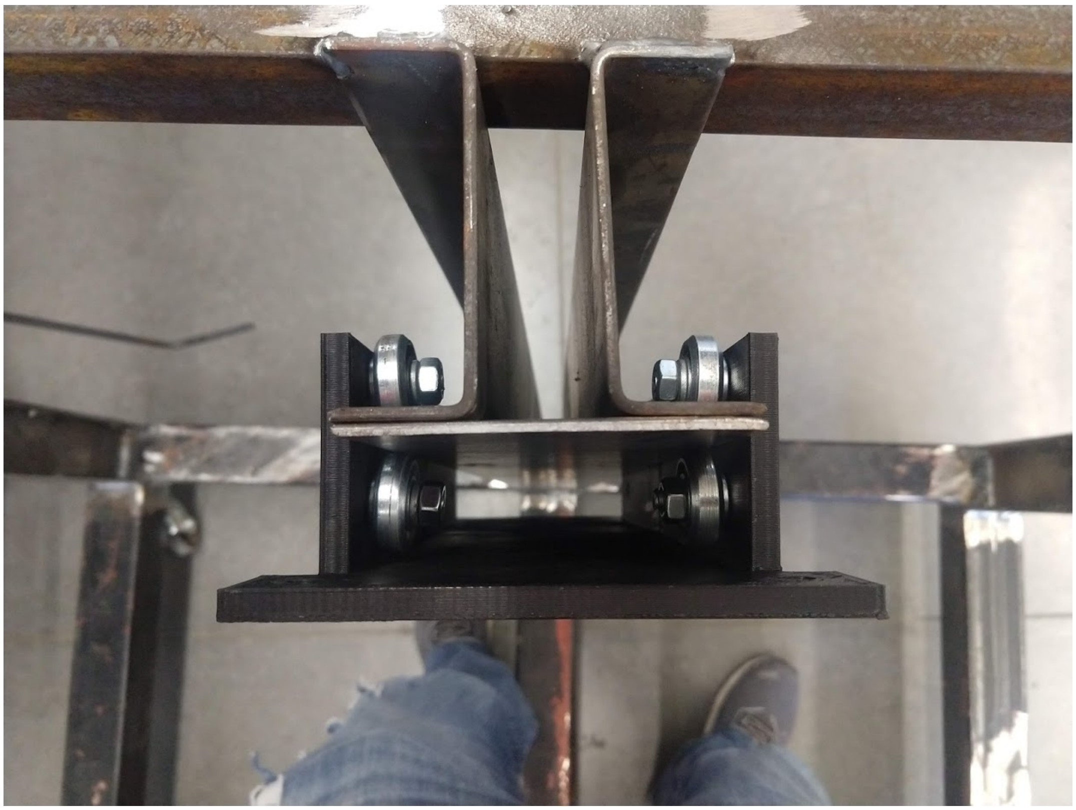


Figure 82: Cross-section of the center built-up column

8.3 The Carriage and Sleds

The next step in the construction of the prototype was to fabricate the carriage and the sleds that guide the machine vertically up and down the columns. The carriage serves as a rigid frame for the machine while the sleds help to keep the machine aligned horizontally. To create a sturdy frame for the carriage, ¼" steel plate was used and cut using a CNC burn table. This fabrication method provides straight smooth cuts on the metal. The carriage cutout can be seen in Figure 83 below.

Figure 83: CNC cut carriage frame

Next, a rapid prototype was created using a 3D printer for the sleds. This allowed for modifications to be made to the sleds before committing to fabricating a metal version. The sleds that were designed mimic the existing sleds on the full-scale model and are a vital part of the carriage assembly. Each sled has a total of 12 bearings installed, as shown in Figure 82 and Figure 84, which are in contact with the column flange.

Figure 84: Prototype sled design

These bearings help stabilize the sleds by having 4 tapered bearings in contact with the front of the column, 4 tapered bearings in contact with the rear of the front flange, and 4 tapered bearings that ride along the side of the front flange. Once the sleds were successfully tested, the fabrication of metal sleds began. The metal sleds consist of three pieces of metal plate that were welded together with holes drilled for the bearing placement, similar to the 3D printed sleds. The front plates of the sleds were then welded to the backside of the carriage to complete the carriage assembly, seen in red in Figure 85.

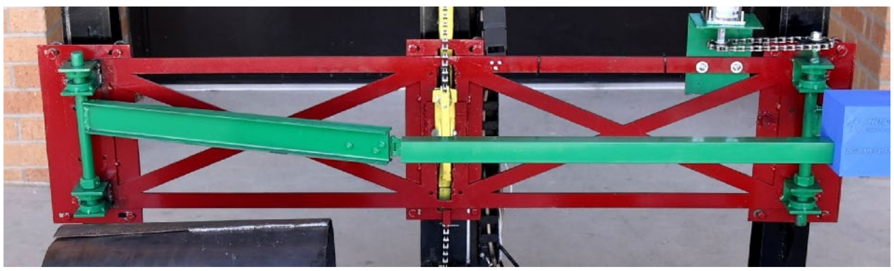


Figure 85: Carriage and sled assembly

8.4 The Jibs

To demonstrate the concept of the jib rotation improvements, several smaller components were fabricated to be able to mimic the full-scale model. The first component constructed was the jibs themselves. Each jib is composed of two 2” c-channel sections that were tack welded together back-to-back to form a custom I-beam section. The c-channel sections used to create the jibs can be seen in Figure 86.

Figure 86: C-channel sections used to fabricate the jibs

Once the jibs were tack welded together, a piece of 1045 steel rod was welded to one end of each jib, shown in Figure 87, denoted by the red arrow. This rod serves as a pivot point for rotation from the motor. Figure 87 also shows the u-shaped mounting bracket (shown by the green arrow) that was fabricated from a bent plate. Bearings (shown by the yellow arrow) were then tack welded inside the u-shaped mounting brackets. The blue arrow in, Figure 87, is a custom collar and set screw that is attached to the jib assembly pivot rod. The collar is made from a nut that had the threads removed and then a hole for a set screw drilled through. When the set screw is tightened on the pivot rod, the jib assembly cannot move in the downward direction through the u-shaped mounts.

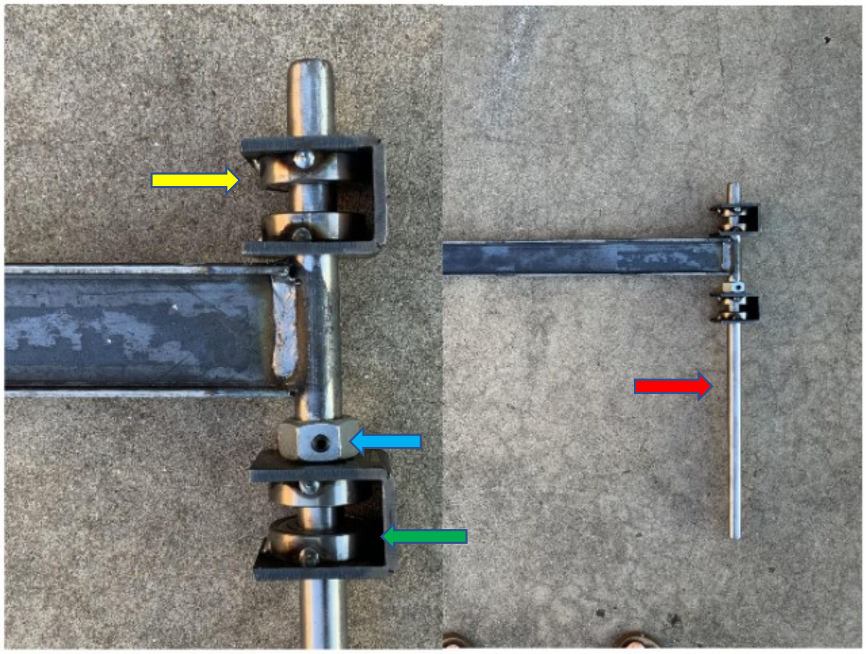


Figure 87: Jib Pivoting Assembly

The next component of the jib assembly is the weld head track that is connected by a hinge to the other end of both the jibs seen in Figure 88. The track is made from a 1-1/4" piece of square tubing. For purposes of the prototype, this weld head track is only used to connect the ends of the jib to produce simultaneous movement.

Figure 88: Weld head track connected to the jibs

After assembling the jibs, mounting brackets, bearings, and the weld track, the entire assembly was then welded to the face of the carriage seen in Figure 89. This completed the fabrication of the main components of the machine. The next step was to attach systems of motors, chains, and sprockets for the lifting function and the jib rotation function.

Figure 89: Jib welded to the carriage

8.5 Lifting Motor, sprocket, and chain assembly

To lift and lower the carriage vertically on the columns, a motor, sprocket, and chain assembly were fabricated. A 12V DC motor was used to provide enough torque to adequately lift and lower the machine. The motor was mounted with the shaft perpendicular to the columns, using a bent plate motor mount with bolts through the front flange seen in Figure 90

Figure 90: Lifting motor mount

To allow for the machine to be lifted from directly in the center, a sprocket and shaft adapter was attached to the shaft of the motor as seen in Figure 91. This is the first sprocket connected in a series of five sprockets. From the shaft sprocket, the chain runs through the cutout in the column to a second sprocket that is positioned in line with the first sprocket shown in Figure 92.

Figure 91: Sprocket and shaft adapter

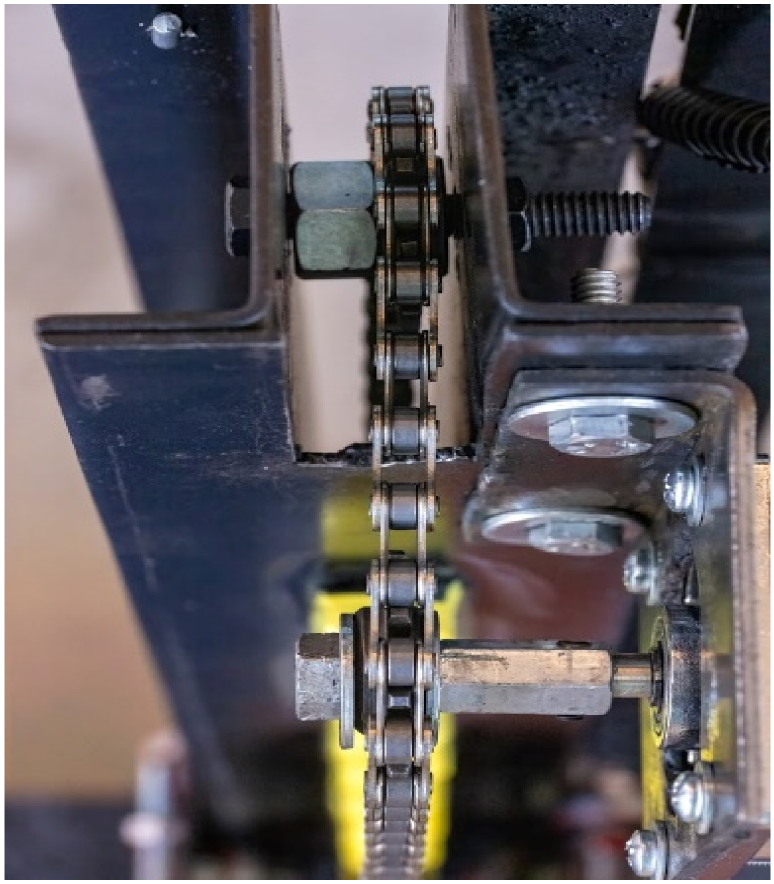


Figure 92: Second sprocket

From the second sprocket, the chain then runs down through the column to the second set of inline sprockets, one located inside the column and the other located directly in front of the column. This fourth sprocket, seen in Figure 93, is positioned so that the chain will line up with a mounting tab located on the bottom of the carriage using a small bolt. The fifth sprocket is an idler sprocket that is located inside the column. Its purpose is to keep tension on the chain so that it remains on the teeth of the sprockets. The sprockets located inside the columns are not shown for clarity. The next step in the fabrication of the prototype was to connect the chain to the top of the carriage using a safety latch mechanism.

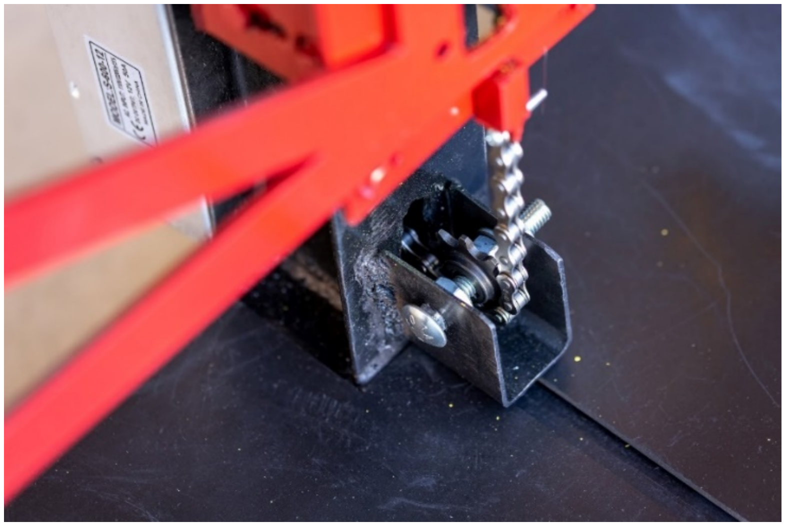


Figure 93: Bottom sprocket mount

8.6 Safety latch mechanism

The safety latch mechanism, seen in Figure 94, doubles as the lift point for the carriage. Where the chain was attached to the latch, metal was removed using a bandsaw and then holes were drilled through both sides to allow for an Allen wrench to be inserted in the place of a pin. Using an Allen wrench will allow for easy removal and a quick release when demonstrating the safety latch in action.

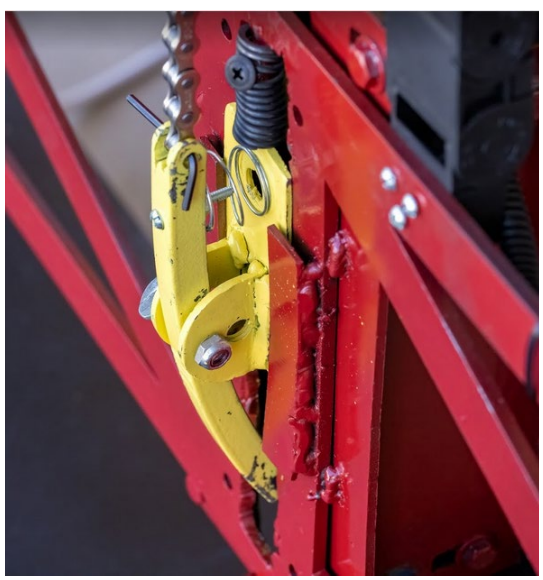


Figure 94: Safety latch mechanism

The latch, in yellow, was made from carbon steel and bent in a bench vise to give a slight curvature to engage the sawtooth track on the column. The latch is mounted onto the yellow sliding plate with two mounting tabs, allowing for the latch to pivot when engaging and disengaging the sawtooth track. For clearance when engaging and disengaging, a rectangular hole was cut out in the red carriage, extending almost the entire height of the carriage. This was done to allow for the entire yellow latch assembly to freely slide up and down the carriage and allow for the latch to pass through to contact the sawtooth track. The yellow latch assembly can slide because it has been sandwiched in between another welded piece of steel to the carriage, creating a guided channel that holds the latch securely in place. At the top of the yellow sliding latch assembly, a large spring is secured in order to support the weight of the carriage, and then when a chain break occurs, extend, before compressing again to absorb the energy of the fall. While lifting and lowering, the tension on the chain compresses a second spring, located in between the yellow latch and yellow mounting plate, keeping the latch disengaged. The instant that tension is released in the chain during a chain break, the spring extends, pivoting the yellow curved latch to the point that it contacts the sawtooth track. After as many as 50 tests, the latch assembly successfully stopped the carriage from falling, without machine damage, every time.

8.7 Jib Motor, Sprocket, and Chain Assembly

The second motor assembly installed on the prototype is the motor that swings the jibs horizontally across the machine. The same 12V DC motor was used for this application. A mounting bracket was made of bent plate and bolted to the back of the carriage, shown in Figure 95.

Figure 95: 12V motor mounted using the mounting bracket

Next, a stationary sprocket with a bolt was welded to the top of the pivoting rod on the jib assembly seen in Figure 96. The bolt was welded to the top of the pivoting rod to create a shaft for the sprocket to be mounted on. A nut with a rubber washer and a metal washer was then used to tighten the sprocket to the shaft.

Figure 96: Stationary sprocket for the jib motor

The second sprocket in this assembly was installed on the shaft of the motor using an adapter with a set screw. Next, the chain was installed around the two aligned sprockets and tested for proper tension. The final jib motor assembly is seen in Figure 97.

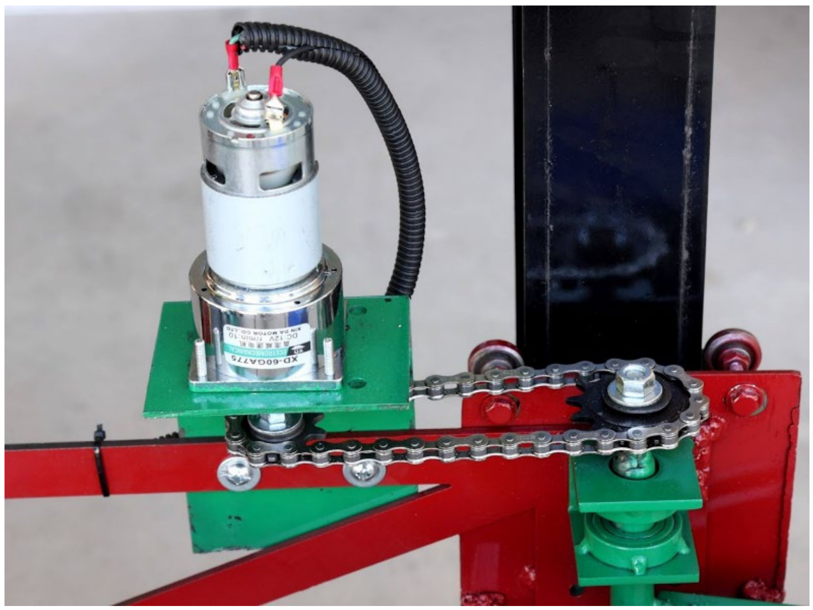


Figure 97: Final jib motor assembly

8.8 Limit Switch

The limit switch demonstration is mounted on the base of the plywood platform as shown in Figure 98. It consists of an 8mm threaded rod that is chain-driven to rotate with the rotation of the drive motor. This display would be modified with the correct gear ratio so that the sled (the green plastic piece in the figure) moves a set distance according to the number of rotations from the motor. Since motor revolutions are directly measurable against the lift height of the carriage or jib rotation in the full-scale welding manipulator, the limit switches on this assembly can be precisely positioned to limit the displacement of the machine. They do this by interrupting the input wires into the VFD for the forward and reverse control of the motor. For example, if the max height of the machine is required to be 15ft, then a limit switch within this enclosure can be positioned to stop the motor at that height.

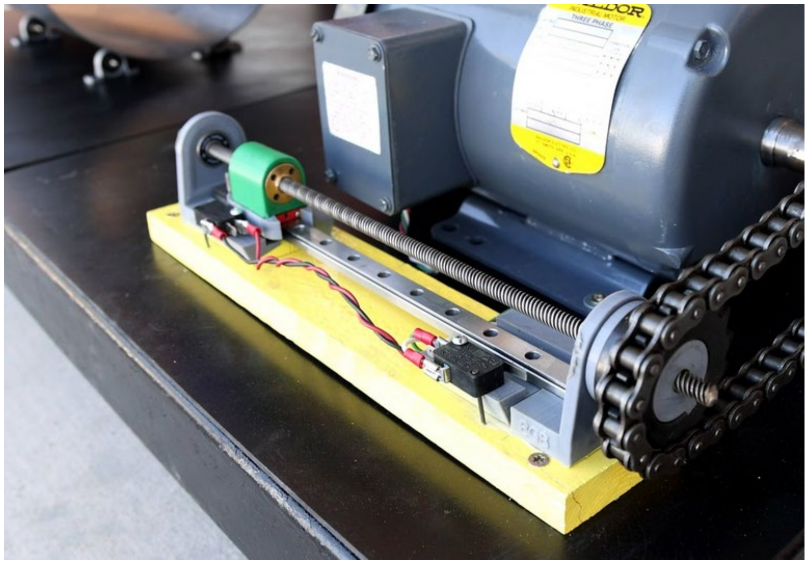


Figure 98: Limit switch assembly

As the green sled moves along the threaded rod, it brushes up against the limit switches, engaging them. It’s important to only brush against the limit switches in order to eliminate the chance that if the sled decelerates too gradually, it would crush the limit switch. In this demonstration, the motor chosen is a 3-phase motor that is used for industrial applications and is not connected to the operation of the prototype lift and jib assemblies.

The VFD, as seen in Figure 99, is similar to what would be used for the full-scale design and demonstrates accurately how the operation would work for Wendland Manufacturing. The additional benefit to having a limit switch enclosure of this type is that the limit switches themselves are protected from the elements inside of an enclosure and are customizable to accommodate different machinery limits.

Figure 99: VFD

8.9 Remote Controls

After the main mechanical assemblies were installed, the next step was to wire the motors to two separate remote controls. The first remote would control the lifting and the jib rotation. The second remote would control the VFD and the limit switch enclosure. For controlling the lifting and the rotation of the machine, a set of two pushbuttons were wired to each of the 12V DC motors. The red and white buttons control the lifting motor, signaling forward and reverse motion. The blue and green buttons control the jib motor, signaling forward and reverse motion. A housing was designed, and 3D printed to use for mounting the buttons, and can be seen in Figure 100 below. A hole on the side of the remote control allows for the wiring to exit the remote and be run to the motors.

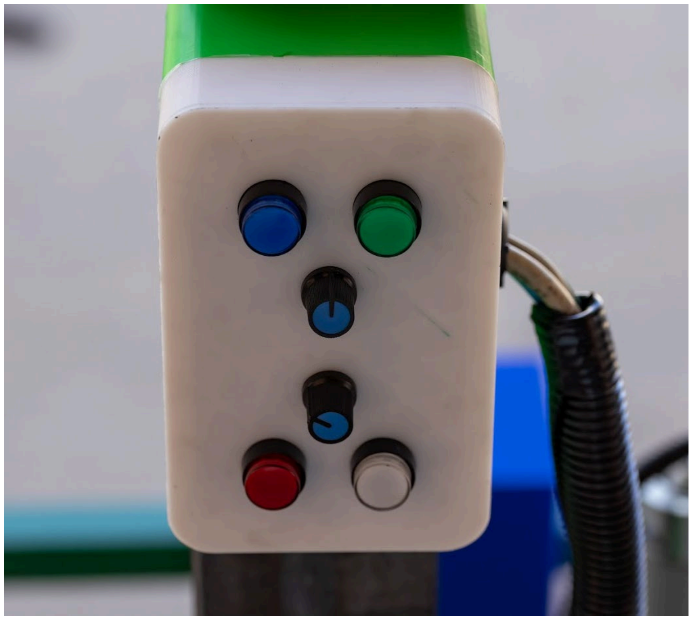


Figure 100: Lift and Jib Remote Control

There are two non-functional potentiometer dials on the face of the remote. These mock dials are installed to replicate what the full-scale model remotes would have. Since this remote was not wired directly to the VFD due to motor sizing issues, a rocker switch was not included. In the full-scale model, this rocker switch would tell the VFD which motor to control.

The second remote control that controls the VFD and the limit switch enclosure demonstration can be seen in Figure 101. On this remote, the white and yellow push buttons control the forward and reverse directions of the motor. A potentiometer with a dial is located next to the direction buttons, wired to the motor that is controlled by the VFD.The potentiometer controls the speed of the motor, allowing the speed to be increased or decreased. To demonstrate the function of the emergency stop from the full-scale model concept, an e-stop was wired to the VFD and installed on the top of the remote control. When the e-stop is engaged, the VFD cuts power to the motor that is driving the limit switch assembly.

Figure 101: VFD and limit switch remote control

8.10 Final Prototype Design

The final prototype design is shown in Figure 102 below. Extra features were added to the prototype to simulate the actual full-scale model. Figure 103 gives another view as shown from an angle. A small shell of a tank, located beneath the weld head track, is placed on a set of pillow block bearings that is meant to resemble the tank rollers in the Wendland shop.

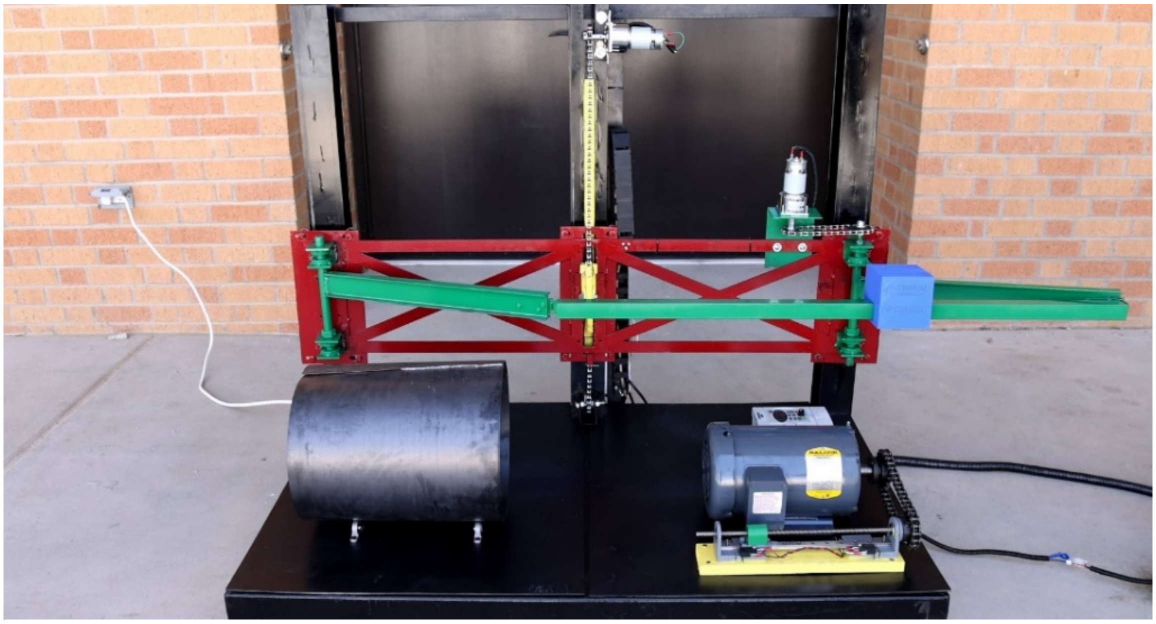


Figure 102: Final prototype design

Figure 103: Final prototype design

There is also the addition of a small 3D printed cube, as seen in Figure 104, that is meant to simulate the sub-arc weld head that traverses the weld t rack of the full-scale model. On the front of the cube is the Wendland logo as well as the BGB team name. One other extra feature that was incorporated into the prototype was the addition of LED lights, as seen in Figure 105. These lights are signaled to turn on when either the lifting or the jib movement is initiated. When the lifting movement is initiated, the LED lights turn on and are red. This corresponds with the red color associated with the carriage and the lifting mechanism. When the jib movement is initiated, the mechanism.

Figure 104: Weld headbox

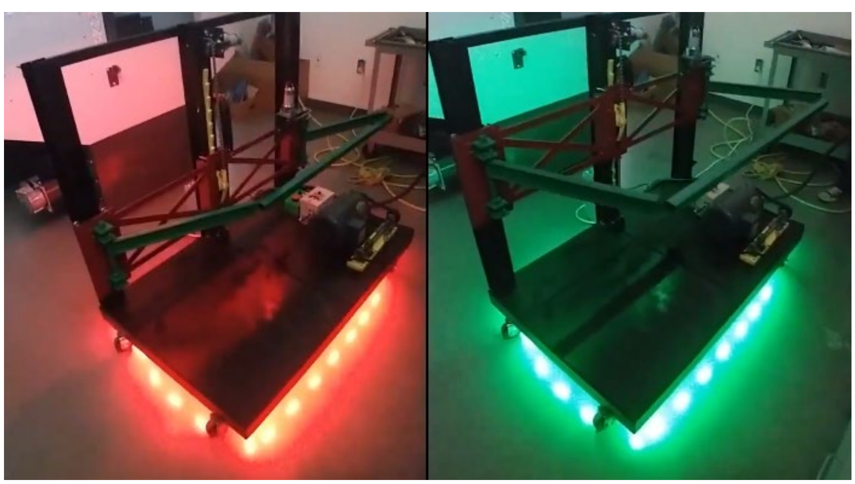


Figure 105: LED Lighting

9 User Manual

9.1 Safety First

The image above is known as the “Safety Alert Symbol”, which will be used throughout the entirety of this manual to indicate warnings that could lead to personal injury or death. It is extremely important to read thoroughly and understand this manual before performing any operations with the machinery. Using this machine without first going through the details listed in this manual is not recommended and could cause damages to the machine and or harm to operators.

9.2 Electrical and Circuit diagram

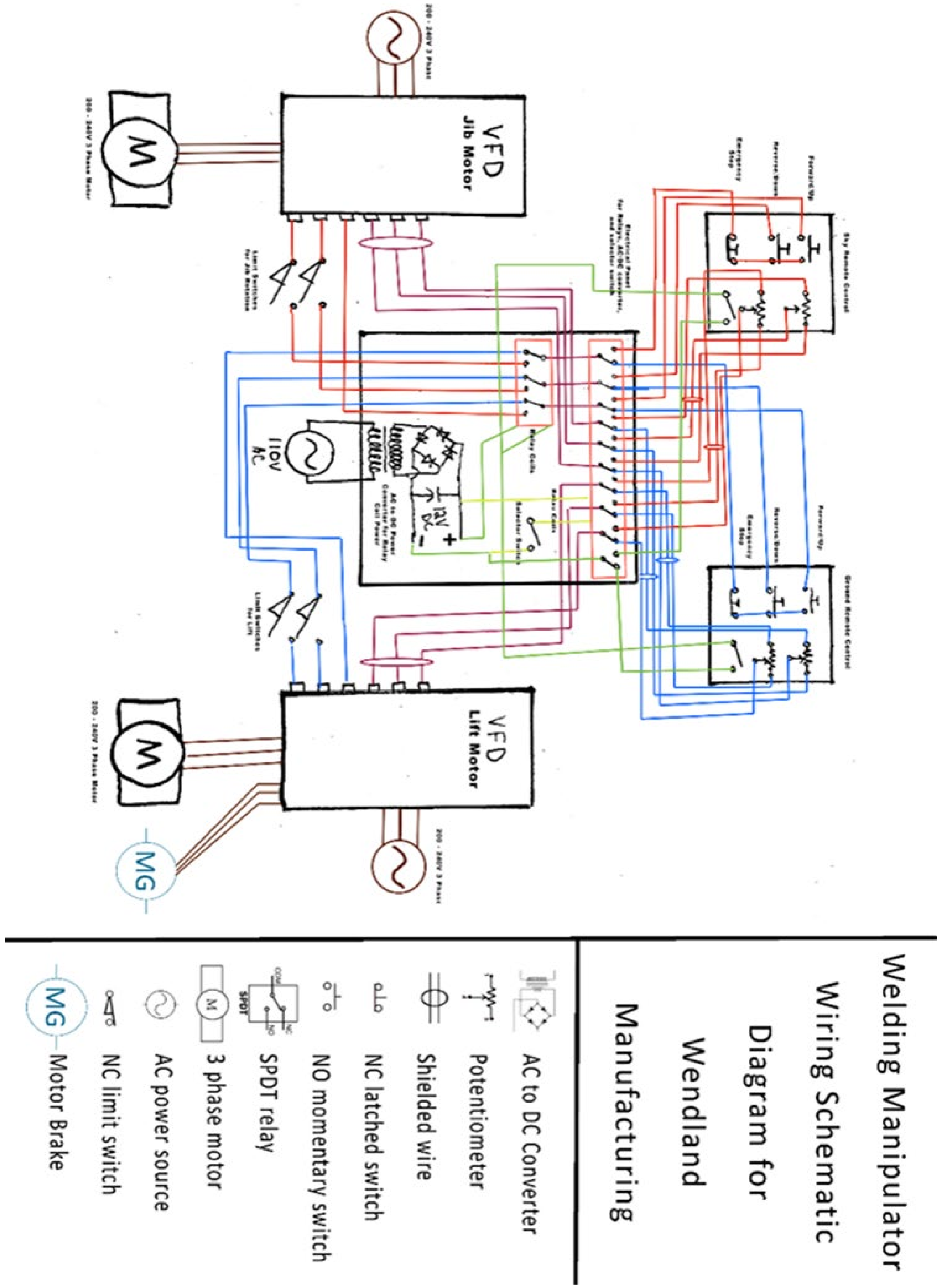


Figure 106: Electrical Wiring Diagram

9.3 Safe Operation

- Before using the equipment read and understand the manual before operations and know the limitations of the equipment before attempting to operate.
- Beware and follow all posted warning signs.
- Always be aware of your surroundings.
- Not recommended operating this equipment if under the influence of alcohol, medication, or any other controlled substance.
- Be aware of any cords, wires, or any other trip hazards.
- Do not allow any person within the working motion of the jib rotation or lift area.
- Do not remove or modify the system or control unit unless given proper training.

9.4 Daily Checks

- Keep the area clear of any debris. Walk around the area make sure the area is free of tripping hazards, trash, or obstructions that may cause harm to the operator or damage to the machine.
- Inspect the equipment before operating to check for loose, broken, or wear on the chains.
- Check for any indications of leaks or exposed electrical wires.
- Make a quick check no person is within the vicinity of the work area before moving assembly from storage position and when lifting the machine is in progress.
- Check to make sure chains and sprockets in line and are connected correctly.
- Check for any inconstant sounds when it moves.

9.5 Operating the Remote Control

Warning: It is not recommended to remove, disassemble, or modify any part on the machine, or it may not work properly.

1. Flip the switch to the position for the correct functionality. Up for lift function. Down for the jib function.
2. With the correct function selected on the switch, press the left/up or right/down button to perform the corresponding function.
3. Adjust speed using the potentiometer adjustment.

9.6 Operating the VFD

Refer to the user manual that came with this product.

9.7 Switching from Sky to Ground Remotes

1. Go to the location that controls the transfer power to the remotes and switch the toggle switch to position for the remote needing to be in control.
2. Turn the switch to the correct position for the function for the ground remote controls or the function from the sky remote control.
3. The position control will activate one remote and disable the other remote so only one remote can be used at any time.

9.8 Operation of the Carriage Lift

Warning: When this machine is in motion be aware of the surroundings and no person is in the area during operation.

1. To begin the operation with the remotes first know which remote will be used. Go to the switch that controls the function from the ground or sky and switch to the position the gives control to the remote which will be used.
2. Once the correct remote has been selected, proceed to the chosen remote and press the toggle switch to the lift position. This will change the motion buttons to the functions set for the lift and connect the correct potentiometer.
3. Press and hold either button marked left or right buttons to begin the rotate function for the jibs to the desired position.
4. The connected potentiometer connected for the jib will allow the operator to adjust the speed of the rotation.

9.9 Operating of the Jibs

Warning: When this machine is in motion be aware of the surroundings and no person is in the area during operation. Keep hands clear from moving parts.

1. To begin the operation with the remotes first know which remote will be used. Go to the switch that controls the function from the ground or sky and switch to the position to give control to the remote which will be used.
2. Once the correct remote has been selected, proceed to the chosen remote and press the toggle switch to the jib. This will change the motion buttons to the functions set for the jibs rotation and connect the correct potentiometer.
3. Press and hold either button marked left or right buttons to begin the rotate function for the jibs to the desired position.
4. 3. The connected potentiometer connected for the jib will allow the operator to adjust the speed of the rotation.

9.10 Maintenance

Warning: Maintenance checks for the following components should be done on a weekly basis. Failure to inspect and maintain these parts could result is serious injury or death. Current maintenance schedules should be continued on pre-existing components.

9.10.1 ***Roller Chain***

- If the chain needs repair, it is strongly recommended to replace immediately to not try and replace specific links which could damage the chain.
- Refer to the manufacture manual or website for elongation requirements for replacing chains.
- Check for damages any part of the link has cracks, or distortion, or irregular movements.
- With a measuring tool, such as a caliper or tape measure check for chain elongation.
- The roller chain should be oiled monthly to ensure proper functioning of the rollers.

9.10.2 ***Motor***

- Check all oil levels.
- Make sure belts are secured and gear guards are in place and in working order.
- Remove any debris that may be on poles.
- Remove dirt from the motor which could cause issues.

9.10.3 ***Safety Latch***

- Check to make sure the spring behind the safety latch is in good working order.
- Make sure that the shock absorber is in good working order, free of debris and rust.
- Remove any debris or dust that may have settled inside the sliding track of the safety latch.
- Check the pin connection to the roller chain on the top of the safety latch for corrosion or wear and tear.

9.10.4 ***Motor Brake***

- Check to make sure the motor brake is free from dust and debris accumulation.
- Verify that the brake disc is still within manufacturers standard thickness tolerance using calipers. If the disc is below the tolerance it needs to be replaced.
- Verify that the air gap is set to the proper distance. Refer to the user manual for the motor brake to obtain these distances.

9.11 Troubleshooting

9.11.1 ***Troubleshooting the Remote***

Problems	Causes	Solutions
Nothing happens with pushing the buttons.	• The remote may not be set as the primary remote. • E-Stop button may be engaged.	• Go to the switch that controls power to the sky or ground and make sure the correct remote is selected. • Twist to disengage.
When pushing the left/right button to move the jibs but the carriage lifts	• The wrong function is selected.	• Switch the toggle switch on the remote to jib selection.
When pushing the up/down button to lift the carriage but the jibs rotate	• The wrong function is selected.	• Switch the toggle switch on the remote to the lift selection.
When operating the lift or jib, the movement is too fast.	• Speed is set too high.	• Rotate the potentiometer to get the desired speed.
When operating the lift or jib, the movement is too slow.	• Speed is set to low.	• Rotate the potentiometer to the desired speed.

9.11.2 *Troubleshooting the Motor*

Warning: Turn off and lock out power before troubleshooting motor.

Problems	Causes	Solutions
Motor is not starting.	• Motor is wired incorrectly. • Power supply or line trouble	• Refer to the wiring diagram to verify. • Check the source of power and fuses.

9.11.3 *Troubleshooting the VFD*

Refer to the user manual that came with this product.

10 Risk Analysis

10.1 Safety and Risk assessment

A risk assessment is performed to determine potential hazards and the level of severity the impacts would have if an incident were to occur. With this information, critical areas of concern for the improvements could be properly accounted for and recommendations can be given to those with lower severity. Figure 107, is the risk assessment that was used to address and identify critical areas of concern and was used when considering possible innovations during the design process. Figure 108 is the risk matrix that displays a visual representation of the risk assessment. As seen in Figure 108, the potential hazards are listed and assigned a severity based on the type of injury which could result if the specified hazard were to occur. The likelihood of the hazard was evaluated and assigned values from highly unlikely, unlikely, likely, or highly likely. Using these conditions, the severity of the risk impacted was determined and highly considered when making the decisions for the design. Risk impact of medium and high was given priority when addressing and proposing the engineered solution.

Hazard	Severity	Likelihood	Risk Impact
Loss of limb from chain and sprocket assemblies	Major Injuries	Unlikely	Medium
Head injuries from jib rotation	Minor Injuries	Unlikely	Medium
Death from carriage falling due to chain break	Fatality	Unlikely	High
Death from carriage falling due to motor failure	Fatality	Unlikely	High
Death from carriage falling due to shaft/sprocket failue	Fatality	Unlikely	High
Electrocution from wiring	Major Injuries	Highly Unlikely	Medium
Fire from damaged wiring due to movement	Minor Injuries	Highly Unlikely	Low
Pinch points from jib rotation	Minor Injuries	Likely	Medium
Pinch points from sled movement	Minor Injuries	Highly Unlikely	Low
Falling off the bird's nest	Negligible Injuries	Unlikely	Low
Tripping hazzard from remote wiring	Negligible Injuries	Likely	Medium

Figure 107: Identified Hazards, Severity, Likelihood and Risk Impact

SCALE OF SEVERITY	SCALE OF LIKELIHOOD: Very Likely	Likely	Unlikely	Highly Unlikely
Fatality			Death from chain break, Death from motor failure, Detah from shaft/sprocket failure	
Major Injuries			Loss of limb from chain and sprocket assemblies	Electrocution from wiring
Minor Injuries		Pinch points from rotation	Head injuries from jib rotation	Fire from damaged wire, Pinch points from sled movement
Negligible Injuries		Tripping hazzard from remote wiring	Injury from falling off of bird's nest	

Low
Medium
High

Figure 108: Risk Matrix for Identified Hazards.

After identifying the hazards, evaluating, and assigning risks, mitigations can be developed that will allow the risks to be controlled and managed. The most important risks to manage are the highest risks. In this evaluation, the highest risks are colored in red. They are death from the carriage falling due to chain breakage, death from the carriage falling due to motor failure, and death due to shaft/sprocket failure. Mitigations for these critical risks must be engineered solutions because developing protocols or implementing personal protective equipment would be unsatisfactory in this case. Mitigations were developed for each of the hazards that were identified and are shown in Figure 109 below.

Hazard	Recommended Action
Loss of limb from chain and sprocket assemblies	Incorporate self lubricating chain guards to reduce maintenance and interaction
Head injuries from jib rotation	Install proximity alarm system to alert when someone is near assembly.
Death from carriage falling due to chain break	Design a safety system to engage when chain breakage is detected.
Death from carriage falling due to motor failure	Design a safety system to secure chain from releasing.
Death from carriage falling due to shaft/sprocket failue	Design a safety system to engage when main assembly desends
Electrocution from wiring	Ground all electrical equipment and double insulate wires.
Fire from damaged wiring due to movement	Incorporate sodium bicarbonate extinquishing system near wires and outlets.
Pinch points from jib rotation	Incorporate hinge covers to reduce access to rotation pins.
Pinch points from sled movement	Install proximity alarm system to alert when someone is near assembly.
Falling off the bird's nest	Design a harness system to reduce the risk of falling.
Tripping hazzard from remote wiring	Incorporate wireless remote system.

Figure 109: Risk Mitigations

The areas of the highest risk were identified to be the main areas of focus during our design phases. To mitigate death due to chain breakage or sprocket/shaft failure, a safety latch system was designed that would prevent the machine from falling to the floor and possibly injuring a person standing beneath the machine. To mitigate death due to motor failure, a motor brake was recommended to be installed on the motor. Other mitigations for less severe hazards include installing guards around moving parts and proximity sensors that would not allow the machine to move if it detected a body within the vicinity of operation.

11 List of Standards

1. Guards for Moving Parts:

   - Section 2-1.10: Guards for moving parts: (a) Exposed moving parts, such as gears, set screws, projecting keys, and drive chain and sprockets, which constitute a hazard under normal operating conditions, shall be guarded. (b) Each guard shall be capable of supporting, without permanent deformation, the weight of a 200 lb (90 kg) person, unless the guard is located where 11 ASME 830.2-2005 it is not probable that a person will step on it (see ASME BI5.1)

2. Roller Chains, Attachments and Sprockets:

   - B29.1 Precision Power Transmission Roller Chains, Attachments, and Sprockets: This Standard is a revision of the B29.1 portion of ASME B29.100-2002 and its establishment as a separate Standard. This standard covers transmission roller chains, attachments, and sprockets. A roller chain is a series of alternately assembled roller links and pin links in which the pins articulate inside the bushings and the rollers are free to turn on the bushings. Pins and bushings are press-fit in their respective link plates. Roller chains may be single strand having one row of roller links, or multiple strand, having more than one row of roller links. Tables are also provided for (a) General chain dimensions; (b) Maximum width over regular pin; (c) Dimensional limits for interchangeable chain links; (d) Straight- and bent-link plate extension dimensions; (e) Sprocket-tooth section profile dimensions of commercial and precision sprockets; (f) Extended pin dimensions; (g) Maximum eccentricity and face runout tolerances for commercial and precision sprockets; (h) Sprocket flange thickness; (i) Seating curve dimensions and tolerances; (j) Minus tolerances on caliper diameters of commercial and precision sprockets; (k) Pitch diameter, outside diameter and measuring dimension factor for a chain of unity pitch; and (l) Whole depth of topping hob cut WD for each pitch and range. The Nonmandatory Appendices have been revised.

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