Automated Truss Stacking
Final Report

A design project produced by

Design Team 16

for

All-Fab Building Components Inc.

Project Advisor:	Tarek Elmekkawy	
Team Manager:	Stephen Foord	______________________
Team Member:	Kristin Johnston	______________________
Team Member:	Adrian Iancu	______________________
Team Secretary:	Neil Krawchuk	______________________

Letter of Transmittal

University of Manitoba
Winnipeg, MB R3T 2N2
November 24, 2010

Professor Paul Labossière
Department of Mechanical and Manufacturing Engineering
University of Manitoba
Winnipeg, MB R3T 2N2

Dear Professor Labossière:

The attached report, "*Design of a Horizontal Truss Stacking System*" is submitted in accordance with the requirements laid out on the following website: http://home.cc.umanitoba.ca/~labossip.

The report was created to explain the final design, and demonstrate how we met our customer's design criteria. The report includes detailed drawings and descriptions of each design element. These drawings and descriptions will aid in the overall comprehension of the operation and feasibility of the design. Team 16 has also included an overall cost analysis and stress analysis of all major components.

We would like to acknowledge Wolfe Smythe and all of the employees at All-Fab Building Components Incorporated for allowing us to visit the site and for providing us with valuable input.

We hope that this report provides you will all necessary details and exceeds the requirements of this project. If you have any questions, please do not hesitate to contact us using the information provided below.

Respectfully,

Stephen Foord
Team 16 Manager

umfoords@cc.umanitoba.ca

November 24, 2010

Mr. Wolfe Smythe
Lead Facilitator
All-Fab Building Components Incorporated
Winnipeg, MB R2J 0H3

Dear Mr. Wolfe Smythe:

The attached report, "*Design of a Horizontal Truss Stacking System*" is submitted in accordance with the requirements set by Professor Paul Labossière in the MECH 4680 Engineering Design course.

The purpose of our report is to explain the final design and demonstrate how we met your company's design criteria. Our report includes detailed drawings and explanations of each element, in order to aid in the understanding of the operation and functionality of the design. We have also conducted a cost analysis and a stress analysis on all major design components.

We would like to thank Professor Tarek Elmekkawy and Professor Subramanian Balakrishnan for imparting upon us some of their real world experience and technical advice.

We truly hope that this report will exceed your expectations. If you have any questions please do not hesitate to contact us by email or phone.

Respectfully,

Stephen Foord
Team Manager

umfoords@cc.umanitoba.ca

Abstract

Team 16 was tasked by All-Fabrications Incorporated (All-Fab) in Winnipeg to solve their truss stacking problem. Currently, All-Fab is using manual labor to vertically stack the trusses. The use of manual labor creates health and safety issues for the workers and reduces overall productivity. Current procedures require production to be shut down when extremely large trusses come off the assembly line, in order to reassign extra hands to help stack the truss. All-Fab asked our team to design an automated system that horizontally stacks trusses up to 60 feet long.

Although All-fab creates trusses up to 80 feet in length, these trusses cannot be stacked horizontally as they cannot withstand being lifted by a forklift in the horizontal position. Due to time constraints, our team has decided (with approval from the customer) to leave the existing vertical stacking system in place for when longer trusses come off the assembly line. Our client has given us a budget of $ 30 000, however, after speaking with Professor Balakrishnan (November 8, 2010), an expert in the field of robotics, we saw that complete automation could not be achieved with the existing budget. During the design phase we tried to minimize cost as much as possible. Despite our best efforts, staying within the given budget proved to be impossible. Our final design is a horizontal truss stacking system that was modified from an existing design created by Clark Industries [1]. Our design uses lead screws in order to raise and lower the trusses, as well as a removable I-beam that nests within a C-Channel beam allowing the trusses to be moved left and right. In order to move the trusses, we have decided to use a roller system. Our entire project is electrically powered and designed to work outdoors in Winnipeg's extreme climate. The stacking system on the ground is a simple system that uses railway ties and gusseted steel poles. Although our project was unable to be achieved within the original budget, we felt that this system is a viable option for solving the horizontal stacking problem at All-Fabrications Incorporated.

Table of Contents

List of Figures

List of Tables

1.0 Introduction

Team 16 was tasked by All-Fab Building Components in Winnipeg, Manitoba, to design an automated system to stack roof trusses. Currently, All-Fab is stacking trusses in both a vertical and horizontal configurations using manual labour. The current method involves using workers to slide the trusses off of an existing roller transfer table and onto a stacking platform. The trusses are then either shifted into an orderly horizontal stack or manually lifted and rotated 90 degrees into a vertical stacking configuration. This current method is inefficient and extremely unsafe. All-Fab believes safety and efficiency can be greatly improved through the use of an automated mechanical system.

All-Fab has provided our team with many specific needs, which the automated truss stacking system design must address. These needs are tabulated below in Table I:

TABLE I: CUSTOMER SPECIFIED NEEDS

Need Number	Need	Importance (1-5)
1	Operation in Extreme Climate	5
2	Adaptability to Different Truss Lengths	5
3	Adaptability to Different Truss Heights	5
4	Adaptability to Different Truss Weights	5
5	Adaptability to Different Truss Stacking Heights	5
6	Ability to Move Left or Right	5
7	Run Constantly Over Full Shift Time	4
8	System at Rest- No Higher than Existing Rollers	5
9	To Not Exceed Power Capacity	3
10	Cycle Time	5

As illustrated above in Table I, these needs were ranked based on their relative importance to the overall finished design. This ranking was done using a numerical scale of 1-5, with 5 being most important and 1 being least important. Our design meets all of these customer needs and accounts for the relative importance of each one.

In order to meet the aforementioned customer needs, the design team brainstormed and developed a list of technical specifications to be included in our design. These technical specifications each have a corresponding need or needs to which they provide a solution. The technical specifications of our design are as follows:

- The system will be able to operate in temperatures of -40 degrees Celsius to +40 degrees Celsius.
- The system will be able to remain operational during and after large snowfall events of up to and including 24 in. of snowfall.
- The design will be built to operate using existing electrical and pneumatic services. These include compressed air lines, 600 Volts (single and three phase) AC electrical, 240 Volts (single and three phase) AC electrical, and 110 Volts (60 Hertz, single phase) AC electrical services.
- The design must be able to stack trusses 12 to 80 ft in length and 4 to 13 ft in height.
- Standard trusses weighing up to 825 lbs, and structural gables weighing up to 1200 lbs must be handled and stacked by the final design.
- The system must be able to stack trusses right and left of the existing rollers in 4 different zones.
- The system must not have dimensions that are more than 38.5 in. high.
- The design must remain operational for 18 hours each working day.
- Cycle time must be less than or equal to 1 minute.

When taking into account the customer needs and the corresponding technical specifications, the main objectives of the final design can be summed up in the following statement:

The objective of this project is to design a faster, safer and less worker intensive stacking system than the one currently used at All-Fab. Our design must take into account the cycle time, constraints and needs set out by the customer and provide the company with a long lasting, agreeable, and effective solution prior to 6 December 2010.

2.0 Details of Design

As previously stated in the abstract, our final design is a modified version of the Clark Design [1]. The following section details the different components that are contained within our final design. Our team used the SolidWorks CAD program as well as standard stress equations in order to complete the stress analysis of all the components. The Finite Element Analysis photos are shown in Appendix B. Due to time constraints and a lack of familiarity with the subject matter, our team was unable to complete the automation aspect of the design. We will, however, discuss the various components and their placement throughout our system. Our system will require six sets the complete stacking assembly. We will require a minimum of two sets of the stacking platform and a flow stopper per zone. We require these numbers so that no matter the length of a truss, we can stack it in the different zones. We decided to include a safety factor of two in order to make sure that our system can withstand the maximum weight of 1200 lbs.

2.1 Major Components

The following sections discuss the major components of our design, including the translating assembly, the lift assembly, the stacking platform, the roller platform, and the stoppers. We will discuss our reasons for choosing the parts that we did and provide a SolidWorks Drawing for each part. The equations used in the analysis of each part will be provided in Appendix B of the report along with the Finite Element Analysis photos.

2.1.1 Translating Assembly

The translating assembly is the part of the system that supports the truss. This system is comprised of an I-beam, a C-channel beam, drive components, roller supports, and a control system. The translating assembly has the ability to move left or right, thus negating the requirement for two systems (one for each direction). This allows us to use only six horizontal stackers rather than twelve. Figure 1 (below) shows a final assembly of the translating section of the design.

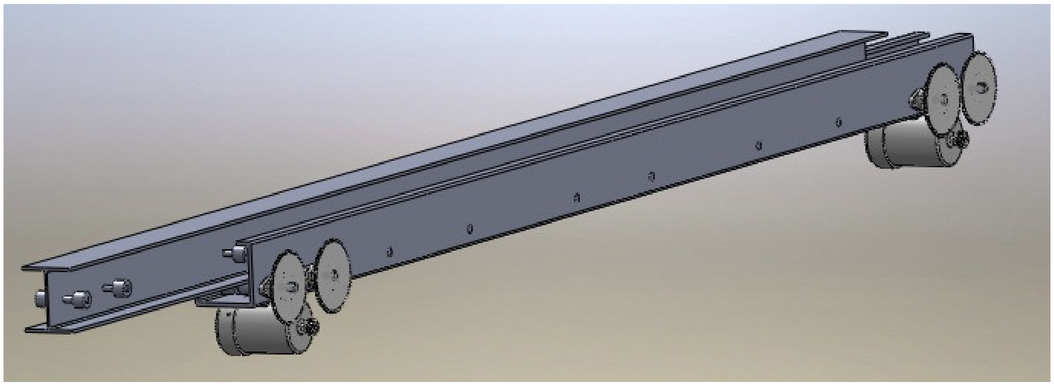


Figure 1: Translating Assembly

2.1.1.1 I-beam

We chose a W8x24, 17 ft long A36 hot steel beam for the I-beam. We went with a standard size I-beam so that the manufacturing would be cheaper and easier. A detailed schematic of the I-beam can be found in Appendix C. We chose A36 steel because it is a common type of steel that is comparatively speaking, cheaper to acquire. Figure 2 (below) shows a SolidWorks drawing of the I-beam portion of the translating assembly.

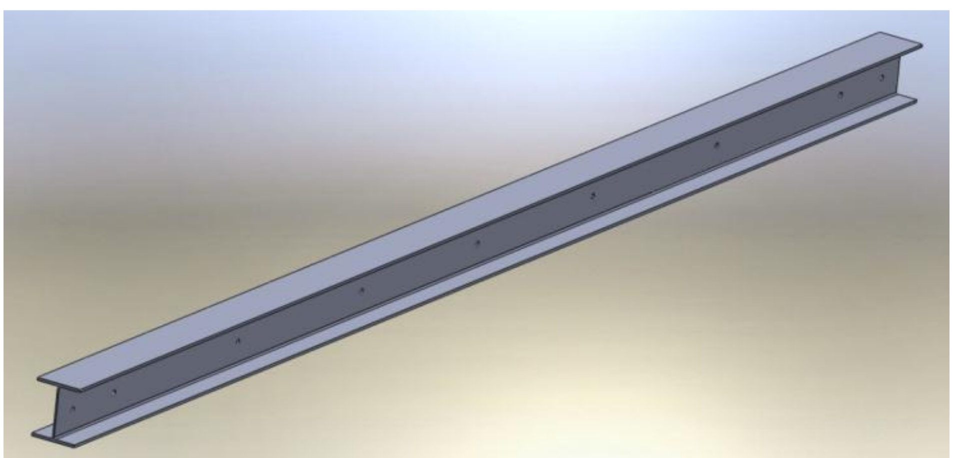


Figure 2: I-beam

This beam was selected because it satisfied a number of design criteria. It provides ample height, which allows the trusses to be lifted above the main roller platform when the system is at its maximum height. It also has adequate width for the surface of the friction drive wheels to move it laterally left and right with minimal slip. The 17 ft length allows the beam to move the trusses past the stacking platform stopping rods, yet still remain sufficiently supported by the rollers at the fixed end.

2.1.1.2 C-channel Beam

For this beam we decided to use an 8 in. X 8 in. X 0.5 in. box beam. This beam will be 15 ft long and once again made of A36 hot rolled steel. A 4 in. strip will be cut into the top of the beam making it a C-channel beam. We decided on the 8 in. X 8 in. dimensions in order to allow the remainder of the translating system to move freely inside of the C-channel. A thickness of 0.5 in. was chosen so that it could withstand the shearing forces on the side walls from the roller shafts. A 15 ft length was selected in order to span the lift supports, while carrying the weight of both the I-beam and truss. The A36 steel was once again chosen because it is a common type of steel and therefore more cost efficient to find and manufacture. By removing a 4 in. strip from the center of the top flange, space was created to insert and repair the wheels and shafts. This adjustment also creates enough of a flange for the upper idler rollers to push against when the I-beam is extended. Figure 3 (below) shows an isometric view of the C-channel beam within the translating assembly.

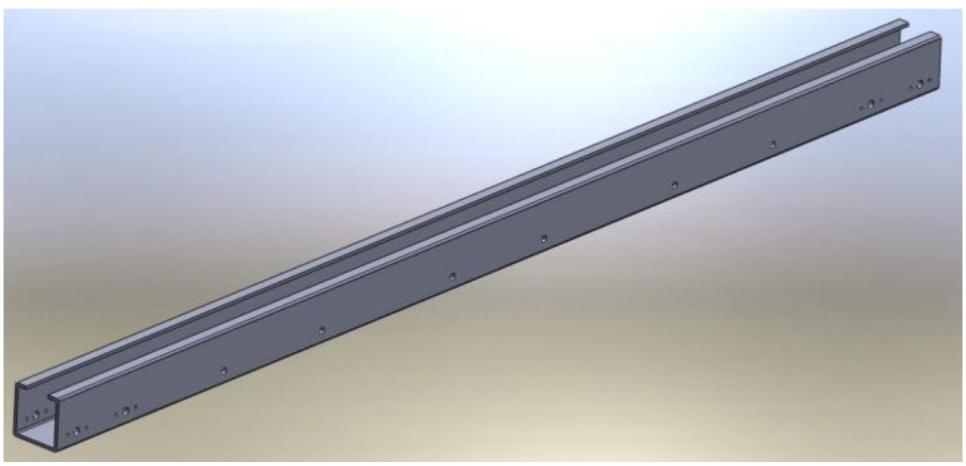


Figure 3: C-channel beam

2.1.1.3 Drive Components

The translating system will be powered using a chain drive system. The chain drive is comprised of two motors for the drive wheels, two roller drive sprockets, four roller driven sprockets, four keys for the sprockets, and four roller chains. The equations that were used to determine the selection of the different components are shown in Appendix B.

The motor for the lateral drive system was developed to satisfy a number of criteria including: transmitting power in both the counterclockwise and clockwise directions, being able to withstand Winnipeg's harsh, year round climate, and being compact enough to mount on the bottom of the C-channel beam. The selected motor is a base mount 3 phase AC motor NEMA 56 (Figure 4). It is rated at 1 hp, rotates at 1725 rpm and includes a 3/16 in. key. This motor is available from McMaster-Carr under the part # 6136K78 [2].

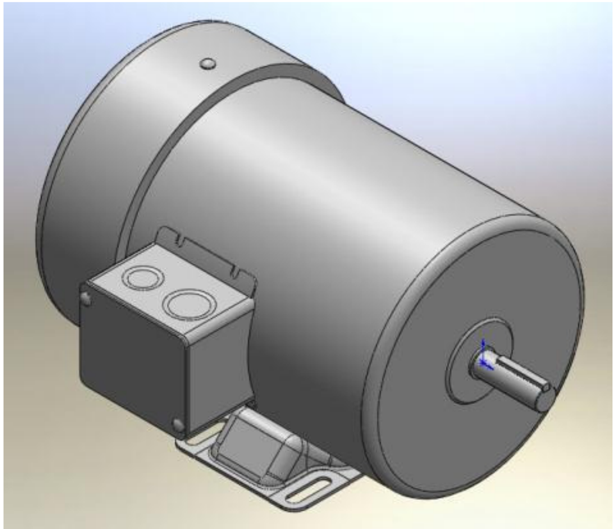


Figure 4: Horizontal drive motor [2]

The roller drive sprockets were also selected from McMaster-Carr under the part # 2784K16 [2]. This particular sprocket (Figure 5) has a machinable bore and a very desirable double-strand configuration. This sprocket has 11 teeth and drives two ANSI # 40, ½ in. pitch chains which satisfy the requirements of the lateral drive system. The bore in these sprockets will need to be machined to 5/8 in. with a 3/32 in. x 3/16 in. keyway cut through the hub in order to accommodate the drive shaft of the motor.

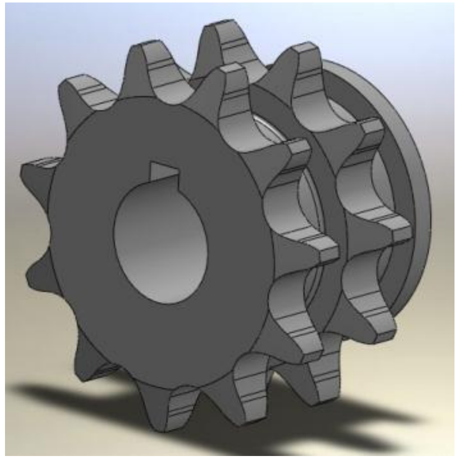


Figure 5: Roller drive sprocket [2]

The driven sprockets were again selected from McMaster-Carr under the part # 2737T533 [2]. This sprocket (Figure 6) has a ½ in. pitch, 54 teeth, 1 in. bore and a ¼ in. keyway machined in the hub, which satisfy the design requirements of the roller drive system (Appendix B). The sprocket is attached with set screws; thus, eliminating the need to further machine the shaft and reducing cost.

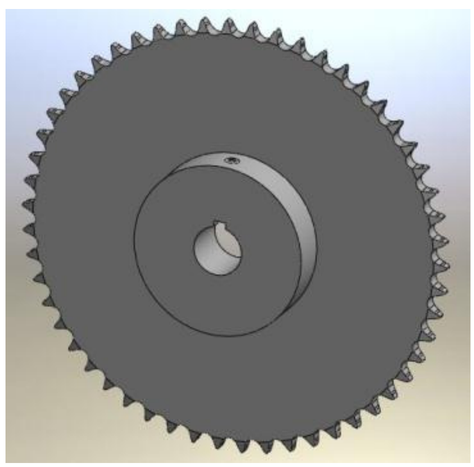


Figure 6: Roller driven sprocket [2]

The keys for the driven sprockets can be bought from McMaster-Carr (part # 98870A245) [2]. The keys (Figure 7) are each C1018 plain steel machine key with square ends. They are undersized with 0.25 in. square sides and are each 2 in. long. This key is the recommended size for a 1 in. diameter bore. These keys are required to mate the driven sprocket to the drive shaft.

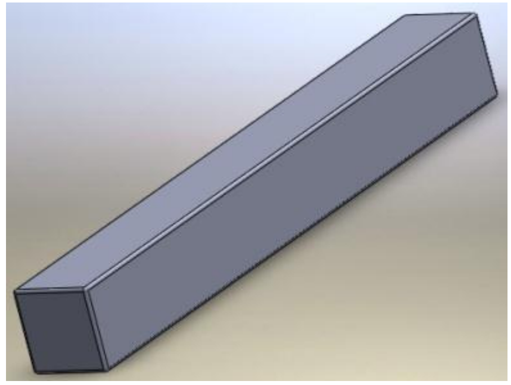


Figure 7: Roller drive shaft key

Each of the four roller chains are required to be an ANSI # 40, 1/2 in. pitch, and 33 in. in length. A chain with the same configuration, but in 36 in. length was also used (part # 6043K103 [2]).

2.1.1.4 Lateral Roller Supports

The lateral roller supports of a single translating assembly are comprised of:

- 8 drive wheels
- 12 lower roller idler wheels
- 18 upper roller idler wheels
- 4 lower roller drive shafts
- 6 lower roller idler shaft
- 9 upper roller idler shafts
- 8 drive roller bearings
- 36 upper wheel retaining rings
- 24 lower wheel retaining rings

The drive wheels that were selected are urethane drive rollers that have a 2 in. diameter, 1 in. bore and 1.94 in. width. These wheels can be purchased from McMaster-Carr under (part # 2475K11 [2]). We chose these rollers because the diameter must be large enough for the I-beam to clear the lift supports. Adequate width was also required to provide a sufficient contact patch to move the I-beam laterally by friction. The bore must accommodate a 1 in. drive shaft and the material must be able to withstand extreme temperatures as low as-40 degrees Celsius. These drive wheels satisfy all of the above listed requirements and are held in place with set screws, which eliminates the need to machine the drive shaft. Our system requires the drive wheels to be able to support a 600 lbs load, however, there is no load capacity specification available for these wheels (we are assuming given the material type that they are suitable).

Figure 8: Drive wheel [2]

The twelve lower roller idler wheels should have the same dimensions as the drive wheels, 2 in. diameter, 2 in. width, and a 1 in. bore. Again these rollers have to withstand temperatures as low as -40 degrees Celsius and be able to support roughly a 600 lbs load. The lower roller idler wheels will be held in place with retaining rings in order to eliminate any machining of the shaft. We were unable to find a commercially available idler wheel; however, companies such as Sunray Inc., can custom make idler rollers to the desired specifications.

The twenty four lower wheel retaining rings must fit on a 1 in. diameter shaft. The selected rings (Figure 9) are 1 in. external self-locking retaining rings made of stainless steel from McMaster-Carr with an associated part # 97622A137 [2]. These can be used because the load on the wheels is strictly radial and as such, the rings will be subjected to minimal thrust loads. These rings will be located on each side of every wheel.

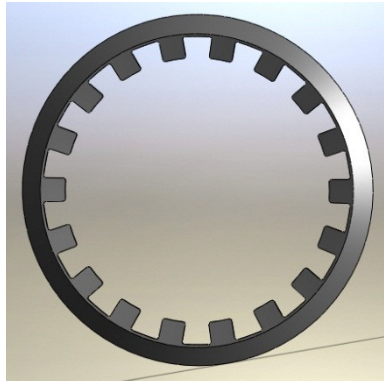


Figure 9: External self-locking retaining ring [2]

The dimensions of the upper roller idler wheels were selected as: 0.75 in. bore, 2 in. diameter and 1.25 in. width. These wheels are held in place with retaining rings in order to eliminate machining of the shaft and must withstand temperatures down to -40 degrees Celsius and support a 600 lb load.

The upper wheel retaining rings that were chosen are once again external self-locking retaining rings made of stainless steel. These rings are suitable for a 0.75 in. diameter shaft and can be found under the part number 97622A132 [2]. They look similar to the retaining ring shown in Figure 9.

The four lower roller drive shafts (Figure 10) were designed to have a 13 in. length, a 1 in. diameter and a 2 in. long, ¼ in. square keyway machined into one end of it. These drive shafts are made of 1020 CD steel. The equations used to evaluate them can be found in Appendix B. These shafts can be purchased, in bulk, from a steel manufacturer and machined to the correct dimensions. The spacing of these drive shafts was chosen to allow both sets of wheels to be in contact with the I-beam even when it is extended to its maximum horizontal position.

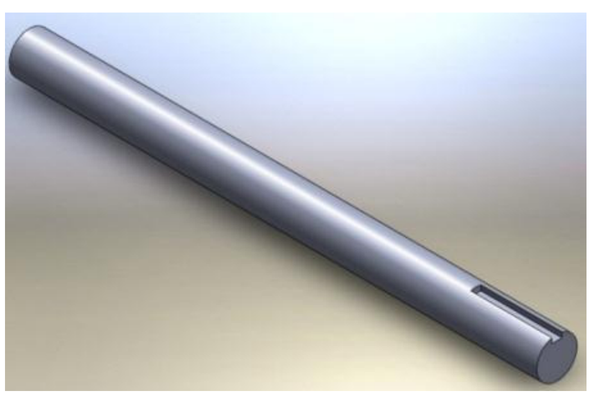


Figure 10: Lower roller drive shaft

The six lower roller idler shafts are each 8 in. long and 1 in. in diameter. The shafts are once again made of 1020 CD steel and the equations used to evaluate them can be found in Appendix B. The number and spacing of the lower idler shafts was designed to evenly distribute the weight of the I-beam and truss; as well as, to provide support for the I-beam throughout its entire lateral movement.

The nine upper roller idler shafts are each 6.5 in. long and ¾ in. in diameter. The shafts are made of 1020 CD steel and the equations used to evaluate them can be found in Appendix B. These shafts are mounted to the I-beam in order to allow the wheels to contact the upper flanges of the c-channel beam and keep the I-beam from pivoting on the lower rollers. The spacing was selected to provide support throughout the entire lateral movement of the I-beam, especially when the I-beam is extended to its maximum horizontal position.

Both sets of idler shafts can be purchased in bulk from a steel manufacturer and cut to the proper size.

The eight drive roller bearings (Figure 11) selected from McMaster-Carr are high temperature, cast iron, mounted, stainless steel ball bearings with a two bolt flange mount. These bearings work for a 1 in. shaft diameter and the calculations for sizing the bearings can be found in Appendix B. The associated part number is 6665K24 [2].

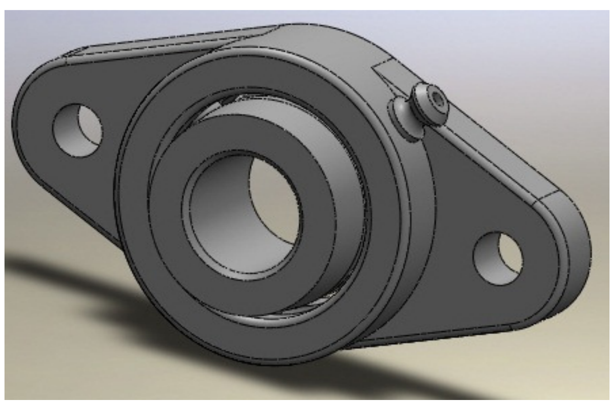


Figure 11: Drive roller bearings [2]

2.1.2 Lift Assembly

The following sections describe the design of the lift assembly. All equations and associated Finite Element Analysis (FEA) photos are listed in Appendix B.

2.1.2.1 Lead Screw and Lead Screw Nut

Following the advice of Professor Paul Labossière, our team decided to use lead screws to raise and lower the trusses. The main reason that we chose to use lead screws was that once stopped, they do not move.

A ½ in. – 10 pitch ACME thread lead screw was chosen to drive the lift assembly based on the shear and tensile stresses required by our design. The detailed lead screw calculations can be found in Appendix B. The tensile and shear stress data for a ½ in. – 10 pitch ACME thread lead screw are shown below in Table II.

TABLE II: ½ in. - 10 PITCH ACME LEAD SCREW [4]

	Tensile Stress Area (in^2)	**Shear Stress Area (in^2)**
Required	0.109	0.218
½ in. – 10 Pitch ACME THREAD	0.1225	0.7278

A lead screw with ½ in. – 10 pitch ACME threads measuring 36 in. in length was selected for our design off of the McMaster - Carr, part # 93410A615 [2]. Following the addition of sprockets and bearings onto the shaft, collars were added to the lead screw to accommodate the increased diameters at these regions. The upper portion of the shaft is machined down to 3/8 in. to accommodate an additional bearing. The design is shown below in Figure 12 and a detailed schematic is available in Appendix C.

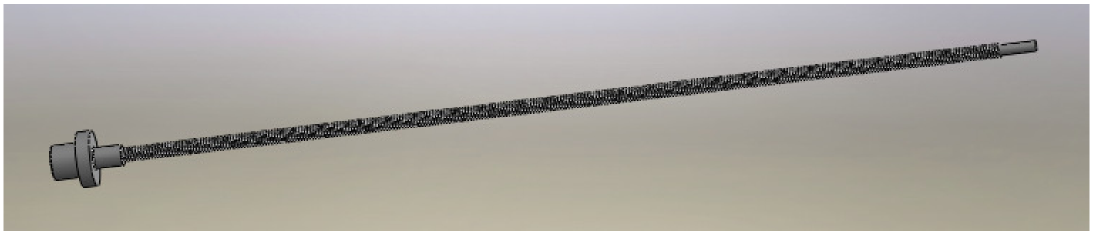


Figure 12: ½ in. – 10 Pitch lead screw

A bronze lead screw nut was selected to transfer the rotational motion of the lead screw into linear motion of the screw plate. The nut is accompanied by a threaded flange to assist in the mounting. The part numbers for the nut and the flange are 95072A109 and 95082A642 respectively [2]. They are shown below in Figure 13.

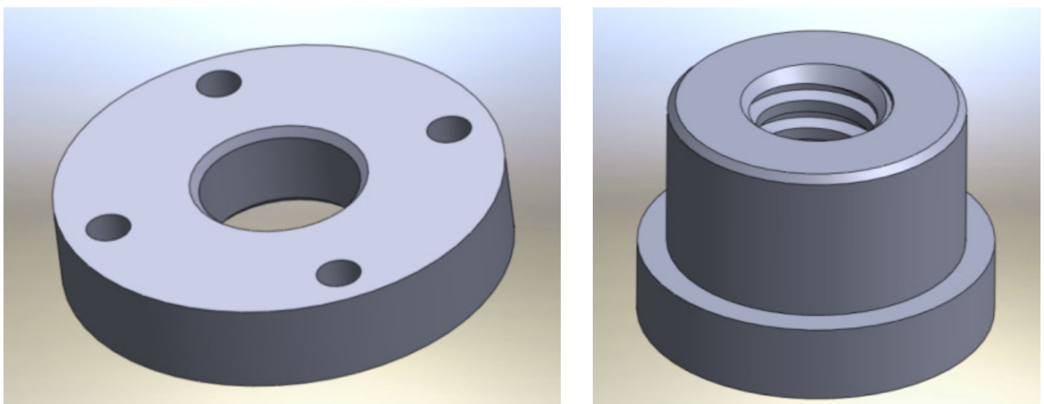


Figure 13: Lead screw nut (right) and flange (left) [2]

2.1.2.2 Lead Screw Plate

Due to translation, when the I-Beam is fully extended, most of the weight is cantilevered on one set of lead screws. This cantilevered state adds bending and shear stresses onto the lead screws. Since lead screws are designed to take primarily thrust and radial loading on their shafts, a system to remove these stresses must be implemented.

A lead screw plate design was created using a 20 in. length of C6 x 13 Steel C-Beam. The lead screw plate has holes drilled into it, allowing the lead screws to move freely perpendicular to the 9 in. depth of the plate. It also has holes drilled into the center allowing it to be bolted to the translating assembly using four 1 ½ in. long, ½ in. diameter bolts. In an attempt to counteract any bending or shear force on the lead screws themselves, a track and runner system was designed. The plate comes equipped with two runners on each of the four corners. These prevent side-to-side movement as well as rotational motion. The design is shown below in Figure 14 and a detailed schematic is available in Appendix C.

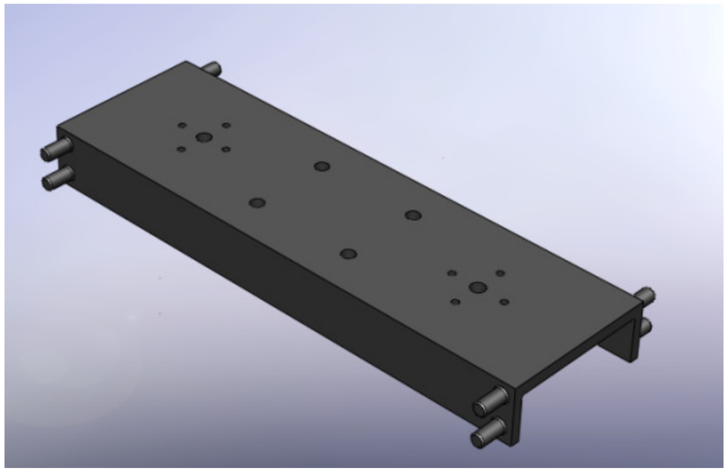


Figure 14: Lead screw - plate

2.1.2.3 Base Configuration

The base design uses two vertical C9 x 20 A36 steel channels, 38 in. in length. These are used to support the vertical motion of the lift assembly while relieving the bending forces applied by the cantilever action when the translating beam is fully extended. The steel channels are supported on three sides by gussets that add lateral stability. Runners were welded to the inside lips of the steel channels. A steel motor mounting plate was also welded to the outside of one steel channel. The motor mount is located in this position due to clearances required by both the chain drive and sprockets and to ensure that the translating assembly is cleared when lowered. Due to the motor mount location, a small section was cut out of the bottom of one of the steel channels to accommodate the chain and sprocket size. The Base design is shown below in Figure 15 and a dimensioned schematic can be found in Appendix C.

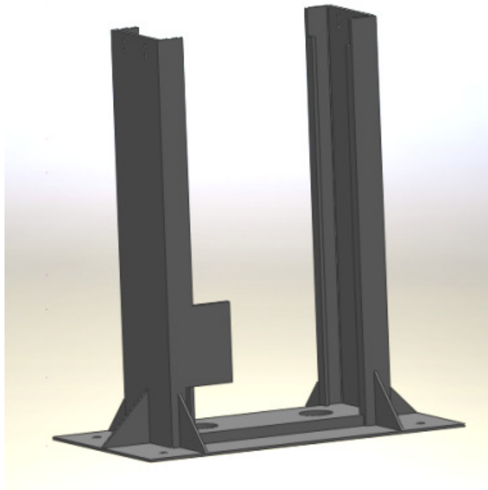


Figure 15: Lead screw – base

Owing to the complexity of the design and the unknown effects of the loading on the base, it was difficult to create an FEA model that will accurately illustrate the shear stresses experienced by this part. As a result it was designed with a high factor of safety to meet the varying load conditions.

2.1.2.3 Upper Supports

At the top of the base, upper supports were custom made to secure the top side of the lead screw. The design shown in Appendix C is made out of 3/16 in. A36 steel. The upper support and base are bolted together with four 3/8 in. diameter bolts. There are mounting holes to accommodate the upper bearings of the lead screw on the lower face shown below in Figure 16. A dimensioned schematic is located in Appendix C.

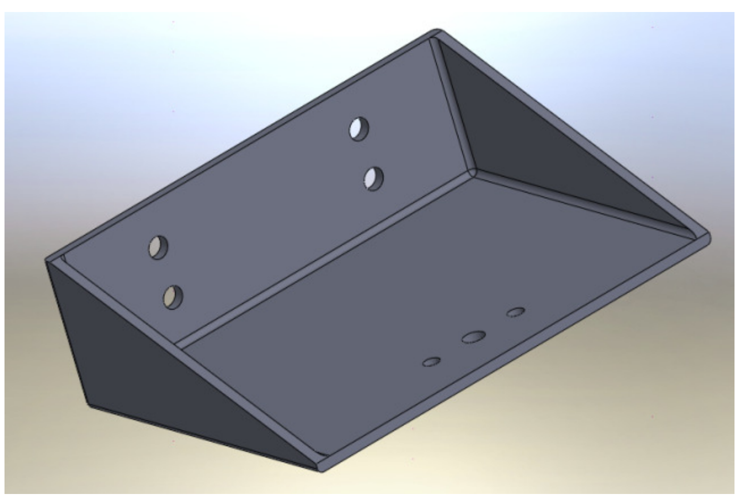


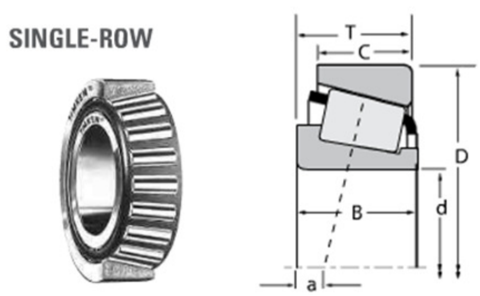
Figure 16: Lead screw - top support

2.1.2.4 Bearings

The following sections deal with the bearings that were used in our design.

2.1.2.4.1 Bottom Bearings

The bearings on the bottom of lead screw were the selected from the Timken Bearing Catalog [5]. They are single row tapered roller bearings with a maximum dynamic radial load rating of 23900 lb. This load rating was determined in Lead Screw Bearings Design Calculations in Appendix B. The bearing part numbers are 3379 and 3339 for the inner and outer sections respectively [5]. The bearing and its dimensions are shown below in Figure 17.



Dimensions

Inner Diameter (d): 1.3750”

Outer Diameter (D): 3.1510”

Bearing Thickness (T): 1.1563”

Figure 17: Lead screw bottom bearing [5]

2.1.2.4.2 Upper Bearings

A self-lubricating aluminum mounted bearing was selected for the upper support from McMaster - Carr, part # 2820T35 [2]. It was selected for its speed rating, and resistance to temperatures as low as -40 degrees Celsius. The housing is also corrosion resistant, which is well suited for being out in the elements and is maintenance free. The bearing is attached to the lead screw upper support via ¼ in. bolts. The upper bearing is shown below in Figure 18 and a dimensioned schematic is found in Appendix C.

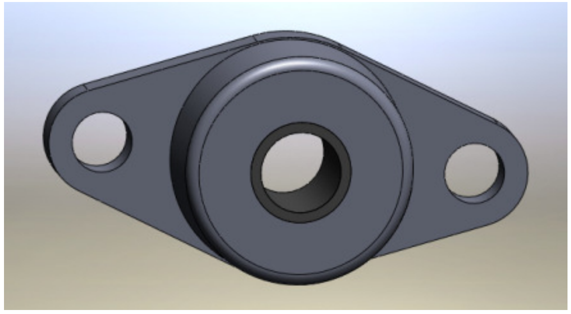


Figure 18: Lead screw upper bearing [2]

2.1.2.5 Drive System

Power Requirements (as stated above in Lead screw design): 0.52 hp
Motor Speed Requirements: 1140 – 1750 rpm.

2.1.2.5.1 Motor Selection

A single phase, 3/4 hp, 1140 rpm motor was selected from McMaster-Carr [2]. This motor is fully enclosed and fan-cooled, which is beneficial in harsh environments where moisture and temperature are issues. It also has a relatively low drive speed. As a result, the speed reduction is less severe when driving the lead screws. This 3/4 hp motor adequately meets the design power constraints, and includes start up capacitors, which give the motor increased torque at start-up. This is beneficial to the initial driving of the lead screws.

2.1.2.5.2 Chain Selection

In consideration of power and speed requirements, ANSI #40 roller chain was selected to best transmit the power from the motor to the leads screws. Due to time constraints and lack of knowledge the drive chain length was not calculated. However, the base's motor mount can be adjusted to the correct chain length and tension.

2.1.2.5.3 Drive Sprocket Selection

According to Mott's textbook, a minimum of 17 teeth in a sprocket should be used when operating at speeds above 100 rpm. Based on size constraints, an ANSI # 40 - 17 tooth sprocket was selected for the motor [4].

2.1.2.5.3 Lead Screw Sprocket Selection

As the output speed, input speed and number of teeth are already known we can determine the desired number of teeth for the sprocket.

$$\frac{n_p}{n_g} = \frac{N_g}{N_p}$$

$$17 * \frac{1140}{600} = N_g = 32.2 \approx 32\ teeth$$

2.1.3 Stacking Platform

Our stacking platform is a combination of railway ties and gusseted steel rods. Our team chose this design as it is cost effective and All-Fab already has the required railway ties. The steel poles had to be gusseted in order to increase their stability and allow them to withstand the opposing forces of a 1200 lbs truss. The railway ties require a 1.5 in. diameter hole to be drilled 4 in. deep to enable the vertical rod to be inserted. The rod is 36 in. in length and made of 1020 CD steel, 1.5 in. in diameter. The equations used to size the rod are listed in Appendix B. Figure 19 (below) shows the stacking support assembly.

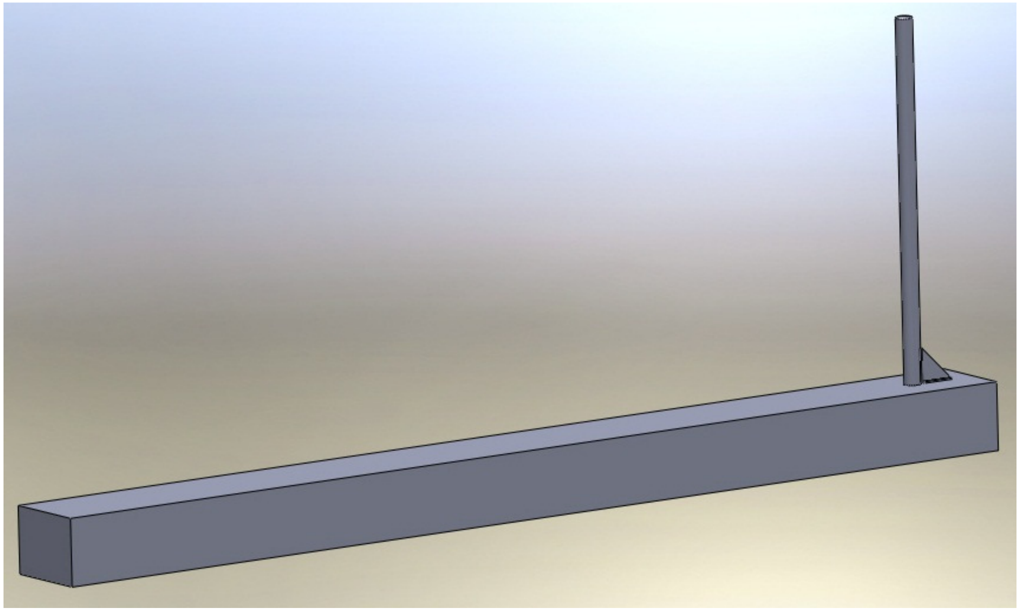


Figure 19: Stacking platform assembly

2.1.4 Main Roller Platform

A large rolling platform currently exists at the All-Fab manufacturing facility to move the trusses manually from the facility exit to the proper stacking location. These rollers consist of 4.25 in. pipes mounted on 36 in. vertical stands via pillow block bearings. Every third roller has a shaft extending from the pillow block to which a chain sprocket is attached. These stands are well secured into the pavement that forms the manufacturing facility's apron. As requested by our client we have made only minor changes, to these rollers. Figure 20 shows one of the powered rollers that make up the main roller support platform.

Figure 20: Main roller support drive assembly

2.1.4.1 Drive System

The rollers are powered in order to improve safety and efficiency, while moving the trusses from the manufacturing facility to the designed stacking system. We believe that a similar system was previously used at this facility as there was already a drive sprocket on every third roller. Powered rollers allow the trusses to be moved out of the manufacturing facility without assistance from workers and therefore, help reach our goal of minimizing physical labor requirements.

Based on the desired safe horizontal speed of a truss along the rollers, each powered roller should turn at approximately 400 rpm. A 1 hp motor will be used to power every third roller. This will account for the forces of the trusses acting upon the rollers and allow for the variation in trusses that the system is intended to handle. This means that even the smallest trusses will remain in contact with a minimum of two rollers at any given time; therefore, ensuring that enough friction and rolling force is applied to the truss to keep it moving forward without slipping. Providing sufficient friction and rolling force is particularly important in cold weather when ice and snow buildup, affect the hardness of materials and cause a reduction of the coefficient of friction between the rollers and the truss. This principle also applies to the larger trusses, where increased mass necessitates an increase in pushing force to keep the truss moving. Having every third roller powered ensures that the largest of trusses are in contact with several powered rollers at any time.

The key components of the roller drive mechanism are an electric motor, chain drive components and a mounting system. A 1 hp electric motor was chosen to drive the powered rollers. This motor, available from McMaster-Carr (Part # 5990K99) is capable of continuous duty operation and has a fan cooled enclosure to protect it from rain and snow [2]. This motor also features automatic overload protection and a steel housing that coated for corrosion protection. At full load, the motor will draw a maximum of 12.8 amps at either 115 or 208-230 A/C volts. This multi- voltage feature allows the customer to select their power supply and maximize their efficiency and power usage ratings for the lowest possible utilities costs. Developing a cost and energy efficient solution was particularly important as the rollers will be continuously running for approximately 18 hours/day. Our solution’s efficiency and usage will be discussed in more depth in the automation and control section. The motor has a built in start capacitor to ensure that is turns on easily at the beginning of every operation session. The capacitors provide the engine with a 400% torque boost over normal operating levels and will allow the motor to start quickly regardless of the weather conditions. The motor also features a run capacitor that makes continuous duty operation more efficient. Since there will be five powered rollers in the design, five of these motors will be purchased through McMaster-Carr [2].

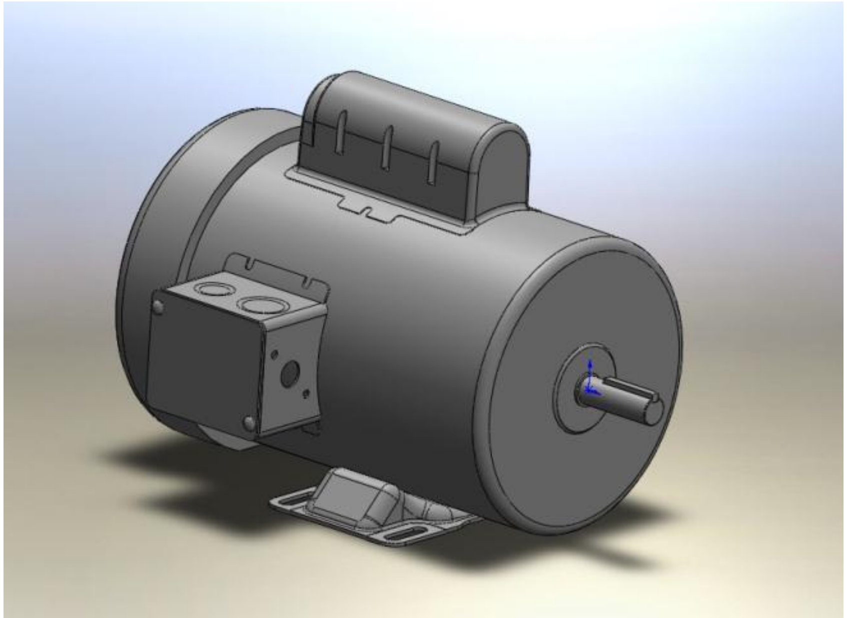


Figure 21: Main roller drive motor [2]

Roller chain was selected to transfer power from the motor to the roller; an ANSI #60 standard roller chain was selected. This chain was selected using Mott's Text and the included horsepower ratings charts for ANSI roller chain. The expected rotational speed was compared to the number of teeth in the smaller sprocket. The charts showed that the chain drive will be adequate for our power requirements. This was acceptable due to the high factor of safety required in order to limit downtime, allow for the damages and strain of outdoor service, and to reduce the chance of a dangerous chain breakage [4].

The motor selected for the drive mechanism rotates at a speed of 1750 rpm. This is much faster than the rollers final output speed of 400 rpm. Therefore, it was decided that a speed reduction system be incorporated in the chain drive through the use of sprockets with dissimilar diameters and numbers of teeth. The two sprockets selected are tabulated below:

TABLE III: ROLLER DRIVE-CHAIN DRIVE MECHANISM SPROCKET SIZES [2]

Sprocket	Number of Teeth	Outside Diameter
Driving	10	2.76 inch
Driven	45	11.18 inch

The driving sprocket is manufactured out of alloy steel and is designed to be durable and corrosion resistant in an outdoor environment. The 5/8 in. bore, was selected to match the output shaft diameter of the electric motor. It also incorporates an ANSI 3/16 in. x 3/32 in. keyway and two set screws in the hub to prevent rotation of the sprocket on the motor shaft [2].

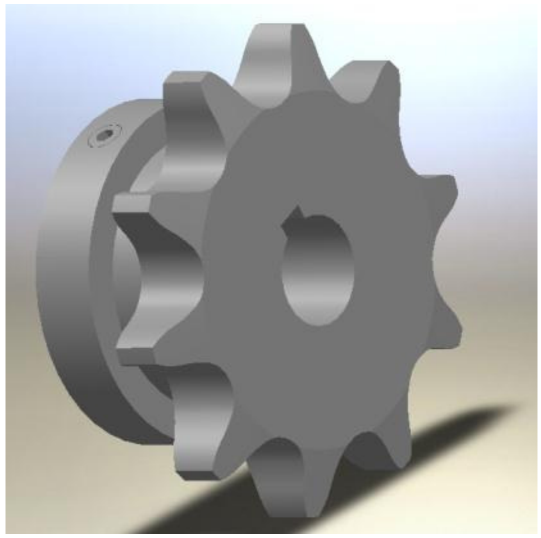


Figure 22: Driving sprocket [2]

The driven sprocket is also an alloy steel component equally capable of providing long service life in an outdoor environment. This sprocket incorporates a 1 in. bore to enable matching of the roller shaft to the sprocket. Similar to the driving sprocket, it incorporates a 1/4 in. by 1/8 in. ANSI keyway as well as two set screws. These will ensure that the sprocket does not rotate on the roller shaft and will transfer its power without slippage. It should be noted that the outside diameter of this sprocket is larger than that of the rollers. This means that in the final design, the sprocket will project above the roller surface. Such a setup is necessary to ensure that the rollers will rotate at the required safe speed and will keep the total cost of the system down by avoiding the use of costly worm gear speed reduction transmissions. The setup is also acceptable due to the fact that at no time will a truss be placed on the rollers in such a way that it will project outside of the roller area. Also, the stacking system has been designed to lift the trusses off of the rollers at a sufficient high to clear the roller drive sprockets during the horizontal translational motion [2].

Figure 23- Driven sprocket [2]

All chain drive components can be purchased from McMaster-Carr for the costs tabulated below. It should be noted that since there will be five power driven rollers in the final design; therefore, five of each component must be purchased. Table IV shows the individual costs of each component as found through McMaster-Carr:

TABLE IV: CHAIN DRIVE COMPONENET COSTS [2]

Component	Price
Drive Chain	$22.12
Driving Sprocket	$13.79
Driven Sprocket	$74.91

The roller drive mechanism is to be mounted well above ground level on the vertical roller support posts. This means that a bracket is required to mount the electric drive motor in this position. A 3/16 in. thick steel reinforced L bracket was designed to perform this task. The bracket has holes that correctly match the bolt pattern of the motor and allow for accurate positioning. The mounting holes for the vertical post mounts are slotted to allow for vertical adjustment and permit initial chain tensioning and adjustment. The ability to adjust the chains will become increasingly necessary as the system ages and the sprockets are worn, causing the chain to slacken.

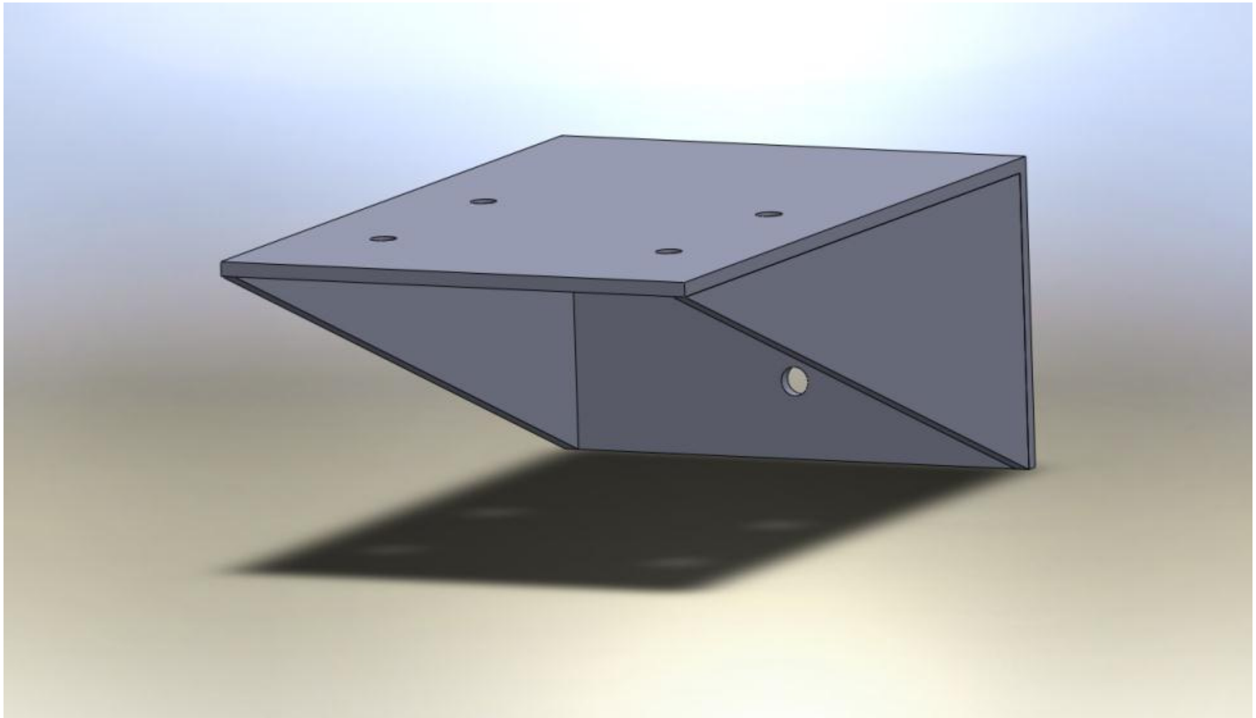


Figure 24: Motor mount

Since the roller drive mechanism is a rotating system, it poses a hazard to nearby workers if left unguarded. In order to protect hands, arms and loose clothing from becoming caught in the rotating chain, guards were designed to surround the chain. These guards, made of 12 grade sheet steel, surround the chain and sprockets and allow for tension adjustment by giving ample spacing. These guards also defend the chain from the elements and the effects of direct rain, snow, and ice. They also help keep the chain lubricated and corrosion-free. It is assumed that this sheet metal can be purchased from local suppliers. It is, therefore, not included in the overall cost requirements.

2.1.5 Flow Stoppers

The flow stoppers that we will use are micro switches. When the assembly line pushes the appropriate zone button, the appropriate micro switch will be activated. Once the truss moves over the micro switch, the associated lift assemblies will begin to move upwards. The lift assembly will elevate the truss off the rollers once it is in position. Due to time constraints the actual automation of the timing was placed outside the scope of this project and will need to be further explored at a later date.

2.1.6 Control Systems

A semi-automatic system was created in order to meet the goal of using limited worker resources to operate the system. This style of automation still requires some user input to operate efficiently, whereas a fully automatic system does not. For example, the semi-automatic truss stacking system will require a worker to input the size of the outgoing truss, as well as to specify in which stacking zone he or she desires the truss to end up. A fully automatic system would be able to calculate outgoing truss dimensions and, based on the previously determined parameters, decide in which stacking zone the truss will be deposited. Although a fully automatic system is more desirable from the standpoint of reducing worker input, a semi-automatic system was specified due to financial constraints.

Due to time constraints, however, an automation system was not fully designed, specified or priced. The following is a list of devices and their location that the design team recommends for proper automation of the truss stacking system.

To properly automate the system there are several machine parameters that must be controlled. These parameters include:

- The powered rollers in the roller platform
- The stopping mechanism
- The translational motion of the W-beam
- The vertical motion of the truss stacking platform

In order to properly stack each truss, they must be stopped at the correct location along the roller platform. This will allow the trusses to be engaged in safe locations by the lifting platform. The trusses can be stopped in one of two ways: either the motion of the powered rollers can be stopped by removing power to the electric drive motors, or a secondary stop can be deployed to block the horizontal motion of the trusses along the rollers by forcing slippage between the trusses and the rollers. For ease of automation and a fast cycle time, it was decided that a secondary stopper be employed instead of stopping the roller drive mechanism altogether. To automate the truss stopper process, an electric micro switch attached to the contact, located slightly higher than the roller, at the outside boundary of the stacking zone in which the truss is to be stacked. When a truss rolls over this switch, it will send a signal to the processor that will then send power to the electric drive in the truss stopping mechanism. The stopping mechanism will be deployed and the truss will butt against it, stopping its horizontal motion.

Once the horizontal motion of the truss is halted, the vertical motion of the stacker platform is able to commence sending power to the lead screw drive motors. This will then start the truss stacking platform moving upwards. Since multiple truss stackers will need to be operated in order to properly lift and stack a single truss, power will have to be sent to each lead screw drive in each truss stacker. As multiple stackers will be used for each truss, the zone assignments for each stacker will have to be pre-assigned in the automation control. For example, stackers 1 and 2 will be pre-assigned to load trusses in zones 1 and 4. When dealing with very large trusses many stackers will have to be used, this system allows for the flexibility required in this occurrence.

A sensor will be positioned at the upper bound of our lift mechanism to prevent over extension and impingement on the upper supports. Once the truss is lifted to its maximum height above the rollers the sensor will send a signal to the automation control, which will then cut power to the lead screw drives. This will halt the vertical motion of the truss stacker platform and cause the lead screws to lock in this position. Once again, since multiple stackers will be used for each truss, this control signal will have to be sent from multiple stackers and the power cut to each lead screw drive simultaneously. This allows for the inclusion of safeguards. For example, should any of the stacker platforms appear to not be reaching the same height as the rest of the stackers within a reasonable amount of time, the system can be halted, the lead screws locked and an alarm signal sent to the operator.

Finally, the truss will have to be extended horizontally out on the translating I-beam for stacking in the appropriate area. Once the stacker platform is lifted to the set height, the sensor that sent a signal to the controller to stop the lead screws will simultaneously send a signal to start the translation drive motors on the stackers. This will cause the I-beams of the engaged stackers to move horizontally, taking the truss with it. When the I-beam reaches the fully extended position, a sensor mounted on the inside of the C-channel, into which the I-beam slides, will send a signal to trigger the translational drive motors to stop. At the same time, this sensor sends a signal to the lead screw drive motors to resume motion except in the reverse direction. This means that the controller must have the capability to reverse polarity of the voltage going to the lead screw drive. This will then cause the truss to lower and become engaged in the stacking stands.

At the stacking level, a row of micro switches would be required to indicate to the control system the height of the stacked trusses. Once a truss is sensed to be coming within a preset distance of this area, the controller will indicate to the lead screw drive motors to stop and lock it in place. Simultaneously, the controller will start the translational drive motor turning in the reverse direction to retract the I-beam back into the C-channel. Once the I-beam is fully retracted, the micro switch in the C-channel indicates that the translational drive motors should be halted.

The process then returns to the original starting position by turning on the lead screw drive and lifting the platform to starting level.

2.2 Final Design

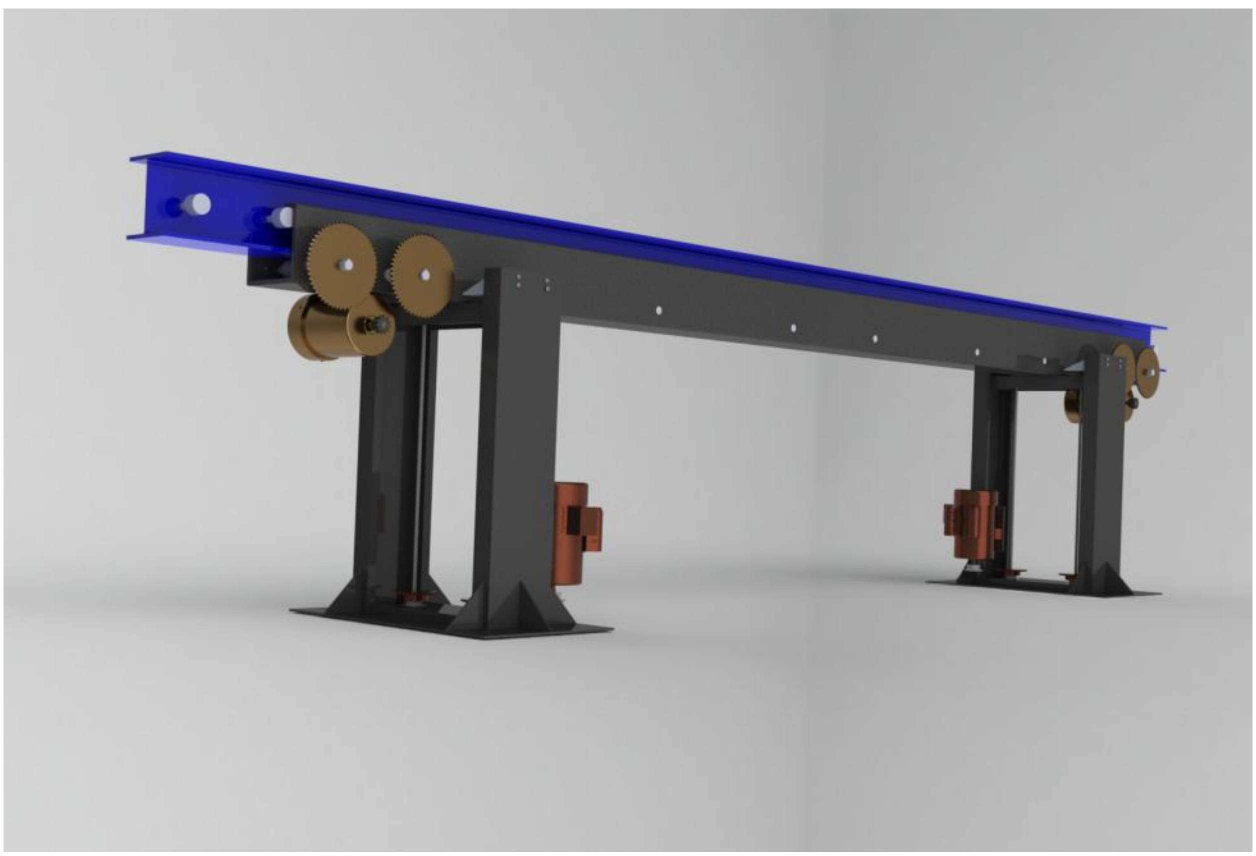


Figure 25: Final stacker design

Figure 26- Driven roller assembly

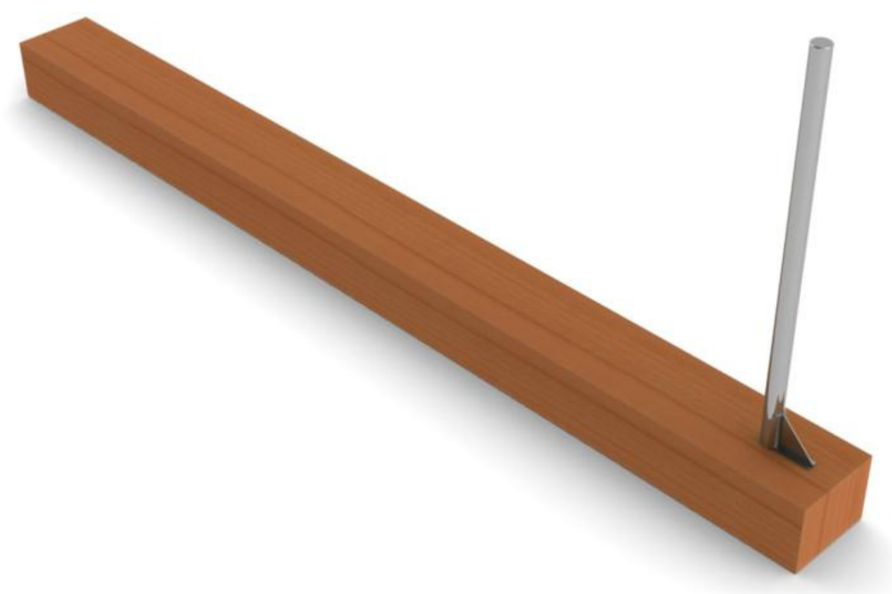


Figure 27- Stacking support assembly

2.3 Operation of Design

The following is a step-by-step guide of how the four designs (main roller platform drive, flow stoppers, horizontal stackers and stacking platforms) work in unison to move a truss from the warehouse to a horizontally stacked position.

Step 1: As the truss exits the warehouse onto the main roller platform the main roller platform motors engage the chain drives and rotate the rollers at 400 rpm (Figure 28). This moves the truss down the main roller platform at about 7 ft/s towards the horizontal stackers.

Figure 28: Main roller platform drive system

Step 2: Once the truss reaches the correct position adjacent to the stacking zone, the horizontal stackers begin the operation of lifting and stacking the truss. The position adjacent to the stacking zone is reached when the truss contacts the correct micro switch.

Step 3: The micro switch initiates the lead screw motors, which then engage the chain drive and rotate the lead screws (Figure 29). This rotation causes the lead screw plate to lift the translating assembly and truss at a rate of 1 in/s.

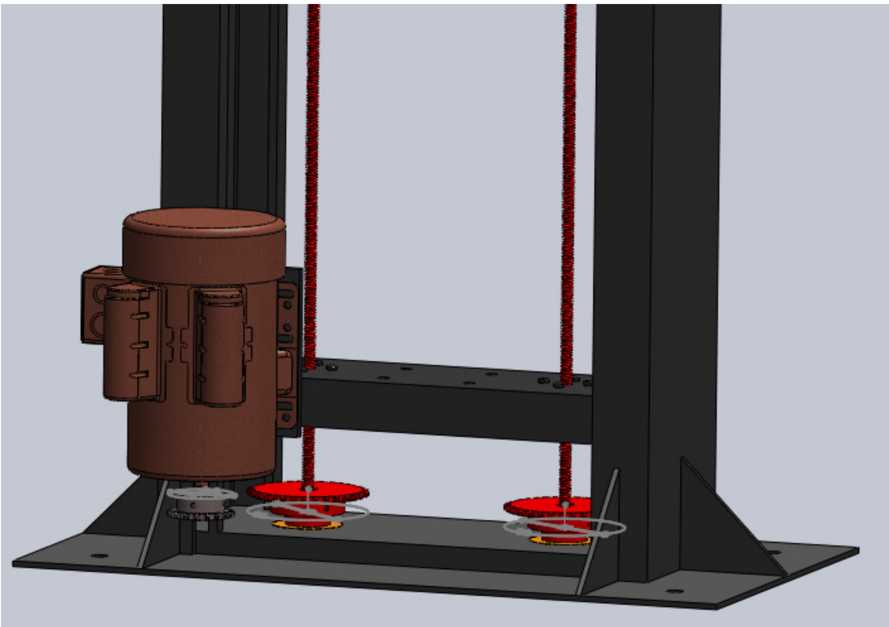


Figure 29: Close up view of vertical drive system

Step 4: The translating assembly begins to move from its home position (I-beam is 2 in. below the plane of the main roller platform) shown in Figure 30. Once the lead screw plate has reached its maximum height (Figure 31), the truss now clears the rollers by 6 in. distance. The distance that the lead screw plate has moved is 8 in. and 8 seconds have passed. Once the lead screw plate passes a micro switch, the lead screw motors are disengaged.

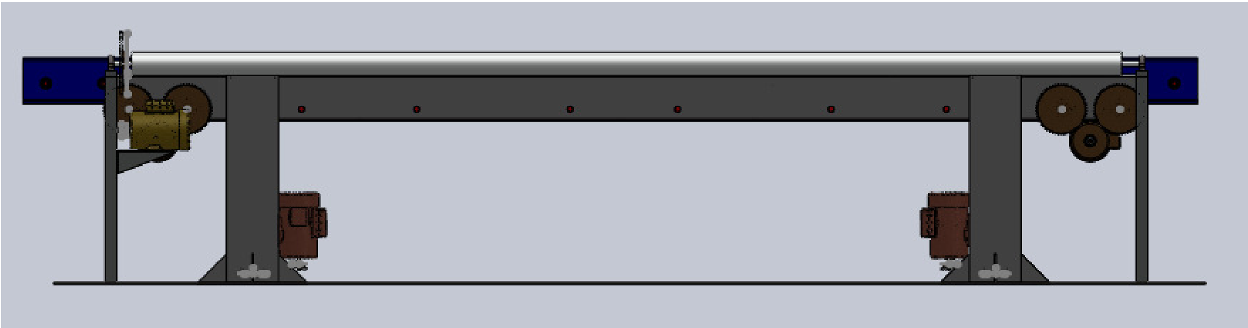


Figure 30: Horizontal stacker in home position

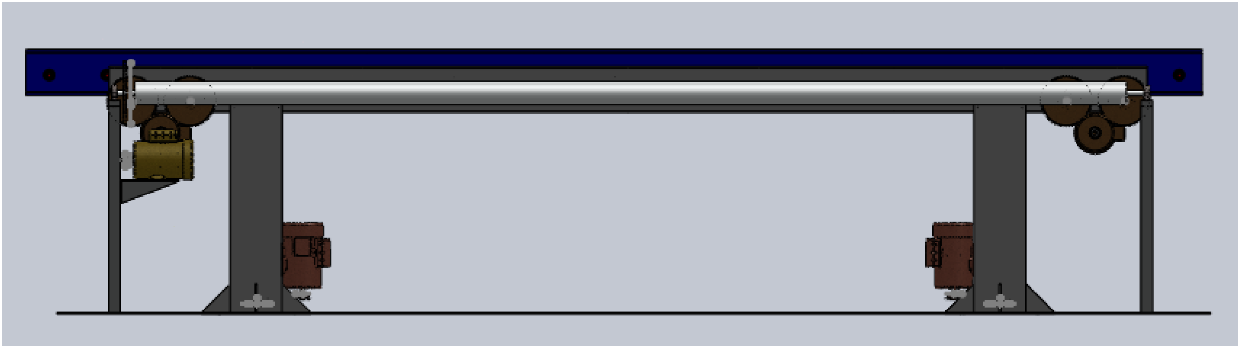


Figure 31: Horizontal stacker at maximum height

Step 5: At the maximum height the horizontal drive motors engage the chain drive and rotate the shafts and wheels at 351.4 rpm (Figure 32). The friction between the drive wheels and the I-beam cause it to move laterally with at speed of about 3 ft/s.

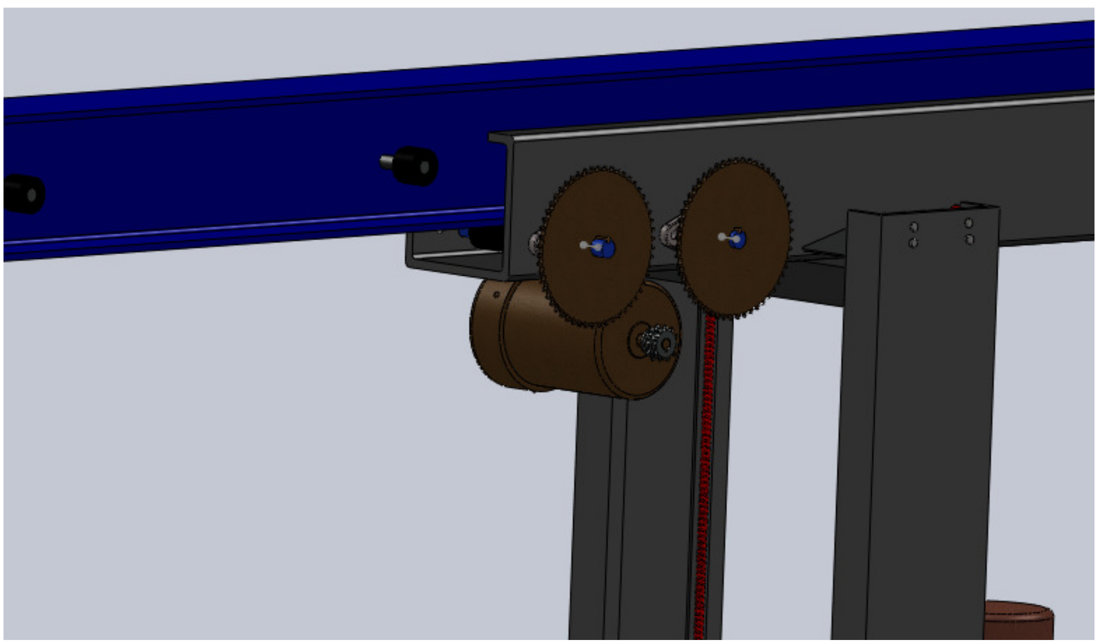


Figure 32: Close up view of horizontal drive system

Step 6: The I-beam begins to move from its home position (side of the beam is 1 ft outside the c-channel beam) to the maximum extended position (side of the beam is 20 in. inside the c-channel beam) as shown in Figure 33. The beam has travelled 172 in. to get to its maximum position over the stacking zone and roughly 5 seconds have passed. Once the side of the I-beam passes a micro switch, the horizontal drive motors are disengaged.

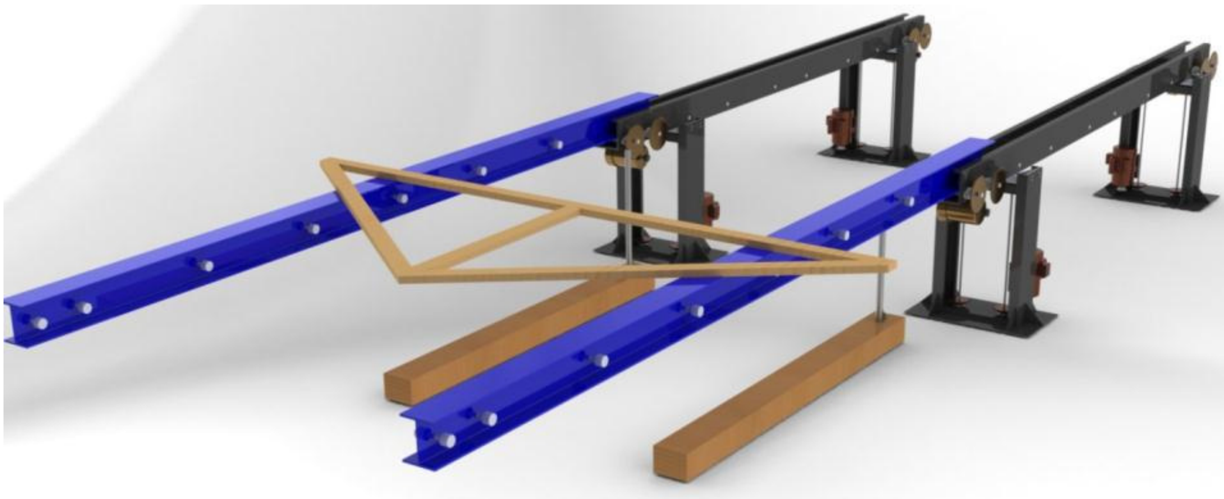


Figure 33: Fully extended I-beam at maximum height

Step 7: At the maximum extended position the lead screw motors engage in the reverse direction to lower the lead screw plate, translating assembly and truss to the stacking position. The different stacking positions start from the truss being 20 in. off the ground (so that the horizontal drive motor does not hit the ground) and index up 1.5 in. (the nominal thickness of 2x4 lumber) for each subsequent truss. Figure 34 shows the horizontal system at its lowest height.

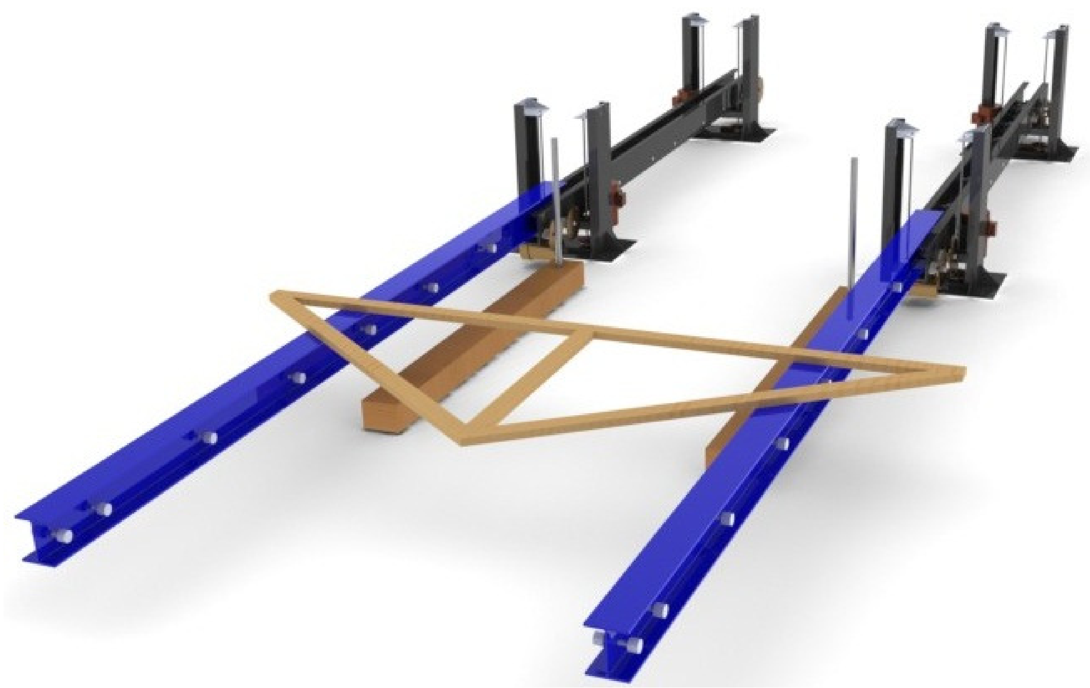


Figure 34: Horizontal stacker, fully extended at lowest height

Step 8: Once the lead screw plate gets to the proper stacking position (computer controlled) the lead screw motors disengage again. The lead screw plate has travelled a maximum vertical distance of 26 in. and a maximum of 26 seconds has passed.

Step 9: At the stacking height, the horizontal drive motors engage (in the opposite direction) and move the I-beam back to its home position. At this point the truss contacts the stopping rods and is displaced onto the railway ties (of the stacking platforms). Figure 35 shows the truss beginning to slide off the I-beam and onto the stacking platform.

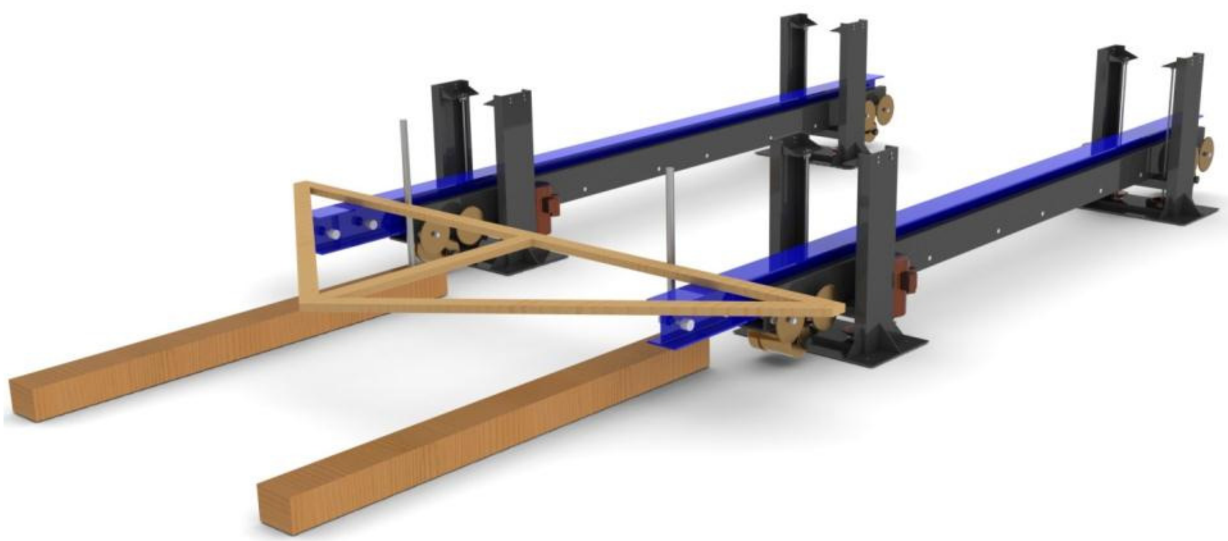


Figure 35: Partially retracted I-beam with truss sliding off

Step 10: Once the I-beam has reached the home position it will pass a micro switch that will disengage the horizontal drive motor. The I-beam has moved back 172 in. and roughly 5 seconds have passed.

Step 11: The lead screw motors engage once more and move the lead screw plate and translating assembly back to the home position. This distance is a maximum of 26 in. and a maximum of 26 seconds have passed.

The total stacking time from the truss sitting on the rollers above the horizontal stackers, to the truss stacked in its final position is a maximum of 44 seconds.

2.4 Cost Analysis

After completing the design and estimating the material and manufacturing costs, the following cost analysis table (Table V) was formed:

TABLE V: COST ANALYSIS SUMMARY

Design Section	Material Cost per Unit	Manufacturing Cost per Unit
Horizontal Stacker	$6497.45	$11209.00
Main Roller Platform	$429.88	$230.00
Stacking Support	$46.00	$65.00

Considering that six Horizontal Stacker units, five Main roller platforms, and eight stacking supports are required, the grand total for the complete design is:

Grand Total = $132 844.10

The material costs of each unit were estimated using several online parts and material stores such as McMaster-Carr. It should note that these costs are only estimated values as parts could be available locally at a significant discount. Also, certain discounts could be incurred based on the size of the order.

The manufacturing costs for each component were estimated assuming a $60 per hour machine/fabrication shop rate. These costs, however, do not take into account the possibility that All-Fab may have an employee on site that is capable of performing the fabrication and machining required to correctly build this design. This estimate also does not account for the possibility that All-Fab could have tools and equipment available for workers to perform some of the simpler assembly duties on-site.

It should also be noted that some materials were not considered to have a cost. This is due to the fact that All-Fab either currently had them on site or is able to obtain them at no additional cost.

The final calculated cost of $132 844.10, is well over our design budget of $30000. These cost overruns are due to several factors. First, it was found that, to correctly stack the trusses safely and efficiently, multiple stacking units were required. This overrun was further multiplied by the variation in sizes of trusses manufactured by All-Fab. This necessitated a wide range of flexibility in our design, which resulted in a more complicated and costlier design.

High material costs in addition to high fabrication costs could also be blamed for the overrun. As mentioned earlier, All-Fab may have personnel available with the necessary skills to build this system. This would reduce manufacturing costs drastically as only material costs would have to be taken into account.

Our analysis also neglects the cost of designing and incorporating a control system to run the design. In a conversation with Dr. Balakrishnan at the University of Manitoba, we were informed that adding control systems to our design would result in an additional cost of approximately $10000-$20000.

For a full breakdown of the materials and manufacturing costs for each design component, please see Appendix E at the end of this report.

2.5 Bill of Materials

For one translating assembly: x6

I-beam: x1

W8x24, 17 ft long A36 hot rolled steel

Channel beam: x1

8x8x1/2 box beam, 15 ft long A36 hot rolled steel

Drive wheels: x8

Urethane Drive Roller 60A Durometer, 2 in. Diameter, 1.94 in. Width, 1 in. ID Bore

Part #: 2475K11 [2]

Lower roller idler wheels: x12

2 in. Diameter, 2 in. width, 1 in. bore

Custom Made

Lower wheel retaining rings: x24

External Self-Locking Retaining Ring Stainless Steel, for 1 in. Shaft Diameter

Part#: 97622A137 [2]

Upper roller idler wheels: x18

2 in. diameter, 1-1/4 in. width, ¾ in. bore

Custom Made

Upper wheel retaining rings: x36

External Self-Locking Retaining Ring Stainless Steel, for ¾ in. Shaft Diameter

Part#: 97622A132 [2]

Lower roller drive shaft: x4

1020 CD steel 1 in. diameter, 13 in. length, ¼ in. square keyway 2 in. long machined on one end

Upper roller idler shaft: x9

1020 CD steel, ¾ in. diameter, 6.5 in. length

Roller drive shaft bearings: x8

High-Temp Cast Iron Mounted Steel Ball Bearing 2-Bolt Flange Mount, for 1 in. Shaft Diameter

Part#: 6665K24 [2]

Horizontal drive motor x2

Base-Mount Three-Phase AC Motor NEMA 56, Tefc, 1 hp, 1725 rpm (includes 3/16 in. key)

Part#: 6136K78 [2]

Roller drive sprocket: x2

Steel Machinable-Bore Double-Strand Sprocket for #40-2 Chain, ½ in. Pitch, 11 Teeth

Bored out to 5/8 in. with a 3/16 in. square keyway

Part#: 2784K16 [2]

Roller driven sprocket: x4

Steel Finished-Bore Roller Chain Sprocket for #40-2 Chain, ½ in. Pitch, 54 Teeth, 1 in. Bore

Part#: 2737T533 [2]

Roller drive shaft key: x4

C1018 Plain Steel Machine Key Square Ends, Undersized, ¼ in. Square, 2 in. Length

Part#: 98870A245 [2]

Horizontal drive chain: x4

We require a 66 pitch or 33 in. long chain ½ in. pitch, ANSI # 40

Low-Temperature ANSI Roller Chain #40, Steel, Single Strand, ½ in. Pitch, 3 ft length

Part#: 6043K103 [2]

For one lift assembly: x12

Lead Screw x 2

½ in. diameter, 10 pitch ACME Thread lead screw – 36 in. in length

Part # 93410A615 [2]

Lead Screw Nut and Flange x 2

Fits ½ in. diameter, 10 pitch ACME Thread lead screw

Part # 95072A109 and 95082A642 [2]

Lead Screw Plate x 1

See technical drawing in Appendix B.1.2

Base x 1

See technical drawing in Appendix B.1.2

Upper Support x 1

See technical drawing in Appendix B.1.2

Lead Screw Bottom Bearing x 2

Timken Tapered Ball Bearing; ID (d): 1.3750 in., OD (D): 3.1510 in., Thickness (T): 1.1563 in.

Part # 3379 and 3339 [5]

Lead Screw Upper Bearing x 2

3/8 in. ID, Self-lubricated aluminum mounted ball bearing

Part # 2820T35 [2]

Electric Motor x 1

NEMA 56H Base-Mount, ¾ hp AC Motor, See Appendix B.1.2 for technical drawing

Part # 5990K74 [2]

Motor Drive Sprocket x 1

17 teeth, Steel Hardened – Teeth, Finished 5/8 in. diameter bore sprocket for ANSI #40 Roller Chain

Part # 2500T491 [2]

Lead Screw Sprocket x 2

34 teeth, Steel, Finished ¾ in. diameter bore sprocket for ANSI #40 Roller Chain

Part # 6236K166 [2]

Horizontal drive chain: x 1

Low-Temperature ANSI Roller Chain #40, Steel, Single Strand, ½ in. Pitch

For one stacking platform: x8

Railway tie:

Drill a 1.5 in. diameter hole, 4 in. deep for stopping rod

Stopping rod:

1020 CD steel, 1.5 in. diameter, 36 in. length

Gusset: x8

¼ in. A36 steel

For one roller drive mechanism: x3

Motor Mount:

A36 steel, dimensions shown on technical drawing

Drive Sprocket:

Alloy Steel, ANSI 60 Chain (¾ in. pitch), 10 teeth, 5/8 in. bore

Driven Sprocket:

Alloy Steel, ANSI 60 Chain (¾ in. pitch), 45 teeth, 1 in. bore

Drive Chain:

Low-Temperature ANSI Roller Chain #60, Steel, Single Strand, ¾ in. Pitch

Drive Motor:

115/208-230V single phase AC electric motor, 1725 rpm, 1 hp, base mount, totally enclosed fan cooled enclosure

Part#: 5990K99 [2]

3.0 Conclusion

In conclusion, our team has decided to modify an existing design fabricated by Clark Industries [1], due to the fact that it meets all of the design requirements except for the budget. After speaking with Professor Balakrishnan, we realized that complete automation within the given budget was impossible. Therefore our team has decided to recommend the modified Clark Design [1]. This design has been recommended for implementation due to the fact that it provides a safe, automated process of horizontally stacking trusses.

4.0 References

[1] Clark Industries Inc. ‘Horizontal Stacker.” Internet: http://www.clark-ind.com/trussStacking.aspx [December 3, 2010].

[2] McMaster-Carr. “All Categories.” Internet: http://www.mcmaster.com/ [December 3, 2010].

[3] Sunray Inc. “Polyurethane Products.” Internet: http://www.sunray-inc.com/ [December 3, 2010].

[4] R.L. Mott. *Machine Elements in Mechanical Design*, 4th ed. Upper Saddle River, NJ: Pearson Education Inc., 2004.

[5] The Timkin Company. “Catalogs.” Internet: http://www.timken.com/EN-US/products/Pages/Catalogs.aspx [December 3, 2010].

[6] Mahoney, Donald S. “Article transfer apparatus.” U. S. Patent 6361265, December 22, 1998.

[7] MiTek Industries Inc. ‘MiTek Horizontal Truss Stacker.” Internet: http://www.mii.com/artefact/download.asp?aid=16525 [December 3, 2010]

Appendix A: Concepts Considered and Selection Criteria

The following appendix will explain the concepts that were considered as well as why they were either rejected or accepted. We have provided a comparison table in the concept selection section which evaluates each of the designs.

A.1 Concept Selection and Analysis

The following sections contain a description of our concept selection process as well as a simplified cost and technical analysis of our chosen design.

A.1.1 Concept Selection

As previously stated in the Research Results section, our project definition was changed a number of times. As a result of these changes our decision scoring matrix includes both horizontal and vertical stacking methods that were considered. The customer needs, and decision scoring matrix are shown in Table VI and Table VII on the following pages. On the decision scoring matrix; 1 is excellent, 0.5 is neutral, and 0 is inadequate.

TABLE VI: DESIGN TEAM - CUSTOMER NEEDS TABLE

Customer Need Number	Customer Need	Importance Rating (1-5)
1	Weather	5
2	Adaptability to different sizes	5
3	Integration of Vertical Stacking	5
4	Left and Right Movement	5
5	Cycle Time	5
6	Reliability	5
7	Simplicity	3
8	Cost	5
9	Ease of Automation	3
10	Truss Damage	2
11	Downtime Intrusiveness	4
12	Stackability	5
13	Safety	5

TABLE VII: CONCEPT DECISION MATRIX

		Need / Importance Rating															
		1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**Total**	**Percentage Of Needs Meet**	**Ranking**
Number	**Idea**	5	5	5	5	5	5	3	5	3	2	4	5	5	57		
1	Scissor Lift - Hydraulic	1	1	0.8	1	1	0.8	0.5	0.5	0.8	1	0.5	1	1	48.4	84.9%	**1**
2	Scissor Lift - Electric	1	1	0.8	1	1	0.9	0.3	0.5	0.8	1	0.5	1	1	48.3	84.7%	**2**
3	Article Transfer Apparatus	1	1	0.8	0	1	0.6	0.1	0.5	0.8	1	0.5	1	1	41.2	72.3%	**3**
4	Angling Conveyor Platform	0.5	1	0.7	1	1	0.5	1	0.8	0.5	0.5	0.2	0	0.5	36.3	63.7%	**4**
5	Crane - Electromagnetic	0.2	0.2	0.5	1	0.7	0.5	0	0	0.5	1	1	1	0	28	49.1%	**9**
6	Crane – Grippers	0.4	0.5	1	1	0.7	0.5	0.2	0	0.5	0.2	1	1	0.1	32.5	57.0%	**7**
7	Gravity-Fed Stacker	0.75	1	0	1	1	0.8	0.5	0.7	0.5	0.4	0	0	0.5	32.55	57.1%	**5**
8	180 Degree Bar Clamps	0.7	0.5	1	0	1	0.5	0.5	0.5	0.2	0.2	0.5	1	0.2	31.5	55.3%	**8**
9	Four Bar linkage	0.5	1	0	0	1	0.5	0.3	0.7	0.5	1	0.3	0	0.7	27.6	48.4%	**6**
10	Bar Separator	0.2	0	0	0	0.2	0.2	0.2	1	0.4	0	1	0	0	13.8	24.2%	**10**

As we can see in Table VII, above we highlighted the article transfer apparatus even though it came in third. The reasons for this will be discussed in length in the sections below.

A.1.2 Concept Evaluations

The following is an evaluation of each all ten concepts from the tenth ranked concept to the first. All of the strictly vertical systems (Truss Stacking Apparatus, Slide and Flip, and C-Shaped Conveyor) have been eliminated as possible solutions based on All-Fab's decision to focus on a horizontal stacking system.

Bar Separator

The Bar Separator was not chosen because it was an extremely complex system requiring movement in the x, y and z directions in order to stack the trusses. This system also creates an increased the risk of damage to the trusses by releasing them above the rest of the stack. Finally, the system is immobile, and therefore, limited to stacking on only one side of the main roller platform.

Electromagnetic Crane

The Electromagnetic Crane design was not chosen primarily due its potential size and concerns over whether the magnetic force between the crane and the steel nailing plates would be enough to support the weight of the truss. The performance of the crane's electromagnet would have also decreased in the cold weather. Finally, the cost to implement and maintain the system would have exceeded our budget.

180º Bar Clamp

This system was not chosen as the grippers increased the risk of truss damage. This system is also extremely complex to design, build and use due to requirement for changeable gripper locations that account for the size and weight of each truss. The cost of the system would have been too great as we would have needed two duplicate systems (for the left and right) and four vertical stepper platforms in order to stack the trusses.

Crane Grippers

This design has many of the same disadvantages as the Electromagnetic Crane, but ranks higher because the clamping method, used to pick up the truss, is more feasible. Size, cost and truss damage presented considerable concern with this design and ultimately led to it not being selected.

Four Bar Linkage

This method was not chosen because it is not a viable stacking system. Due to interference caused by the swing arms an additional system would have been required to stack the trusses. The system also required a complex automation sequence with multiple sensors and a very powerful, reversible electric/hydraulic motor. Although the cost of the system is viable, the system was rejected because it does not fulfill the primary objective of horizontally stacking trusses.

Gravity Fed Stacker

This system was rejected because it did not allow for the integration of a vertical stacking method. It is also not possible to integrate this solution with the equipment that is already present at All-Fab. Workers would also be unable to gain access between the rollers to remove the trusses from the main system, creating a safety concern. This system also relies on sliding the trusses on top of one another using only the aid of gravity, which creates stacking problems. Although this system was within our budget, the system's logistics present significant problems resulting in its rejection as a viable design.

Angling Conveyor Platform

This system was rejected because it once again relies on sliding one truss on top of another. Although this design used power to direct the movement of the truss, there were concerns over the friction and that the nailing plates could catch during the sliding motion. This system is financially viable; however, it would also require a complex automation sequence.

Article Transfer Apparatus

This system is a viable option based on its simple and effective design. The simplicity of the design allows it to handle both the left hand and right hand side stacking. Unfortunately, the cost of automating this design pushes it outside of our budget. The system is a bit harder to maintain due to the positioning of the I-beam inside of the C-Channel beam. Despite this drawback, the design has less parts to maintain.

Scissor Lift

This system is a viable option due to the inherent bi-directionality of the design. The bi-directionality will lower costs as only one system is required to meet all of the requirements for four zone stacking. This design has non-intrusive attributes and removable stacking platforms. These attributes allow All-Fab to revert to the previously used manual stacking technique when the system is under repair. Compared to the other designs the automation and cycle time requirements are relatively easy to accomplish. The overall cost of the scissor lift fabrication and required automation may be an issue. The complexity of this solution also creates more wear points on the system, which may require greater amounts of maintenance and downtime.

A.2.2 Concept Selection

Taking into account all of the advantages and disadvantages of the different design alternatives, it was determined that the article transfer apparatus design would be the most advantageous. We also spoke with our client and discovered that they too preferred the article transfer apparatus. This is mainly due to the fact that the design is simple and meets the cycle time requirement.

Appendix B: Technical Analysis

The following sections detail the equations used, as well as the Finite Element Analysis completed on all of the major design components of our design. All of the calculations were performed with the aid of Mott's textbook [4].

B.1 Major Components

This section will detail the equations that were used and the Finite Element Analysis that was done on the major components. The major components consist of the translating assembly, the lift assembly and the stacking platform.

B.1.1 Translating Assembly

The following equations were used when verifying the material selection of the translating assembly. The translating assembly consists of the I-bean, the roller supports, and the drive system.

B.1.1.1 I-beam

The following equations and material properties were used when verifying the material selection and sizing of the I-beam.

The maximum stress on the I-beam occurs when the system is extended to the maximum lateral position. At this position, 20 in. of the I-beam is being supported by the two drive rollers and the last two idler rollers, while the remaining 184 in. are supporting the distributed load of the truss.

The material properties of the A36 steel are:

Yield strength $\sigma_y = 36$ ksi

Shear strength $\tau = 21$ ksi

Endurance strength $s_n \cong 23$ ksi

Using a factor of safety of 2 and assuming the distributed load behaves as a point load halfway between the free end of the I-beam and the supported end we obtained:

Stress:

$$\sigma_{all} = \frac{\sigma_y}{2} = 18\ ksi$$

Max Load F = 600 lb at x = 92 in. in from the support

M_{max} = 55200 in-lb

$$Smin = \frac{M_{max}}{\sigma_{all}} = 3.067\ in^3$$

The minimum section modulus is far below the section modulus for the I-beam of S = 20.9 in^3.

Shear Stress:

V = F = 600 lb

A_{web} = 1.943 in^2

$$\tau_{max} = \frac{V}{A_{web}} = 308.8\ psi$$

The maximum shear stress the I-beam would experience is far below the shear stress for A36 steel.

Fatigue Stress:

$$\sigma_{max} = \frac{M_{max}}{S} = 2.64\ ksi$$

The maximum fatigue stress is also far below the endurance strength of the material.

To perform a finite element analysis of the I-beam in the maximum stress position we applied the distributed stress of 1200 lb onto the top flange of the I-beam. We then fixed both the area where the drive wheels contact the I-beam when it is fully extended and the two end holes where the upper idler shaft is located. These fixtures were chosen to give a more accurate picture of how the beam will behave in service. Figure 45 shows the stress distribution on the fixed end of the I-beam and Figure 46 shows the displacement of the free end.

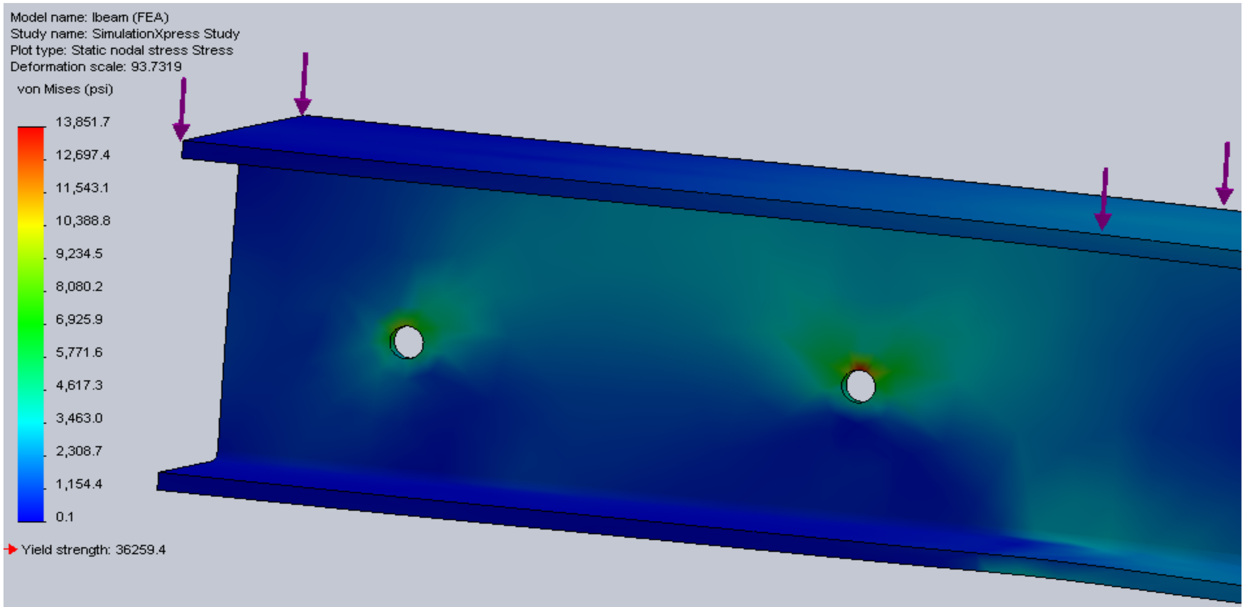


Figure 36: FEA von Mises (psi) I- Beam - Side View (Deformation scale 93.7319)

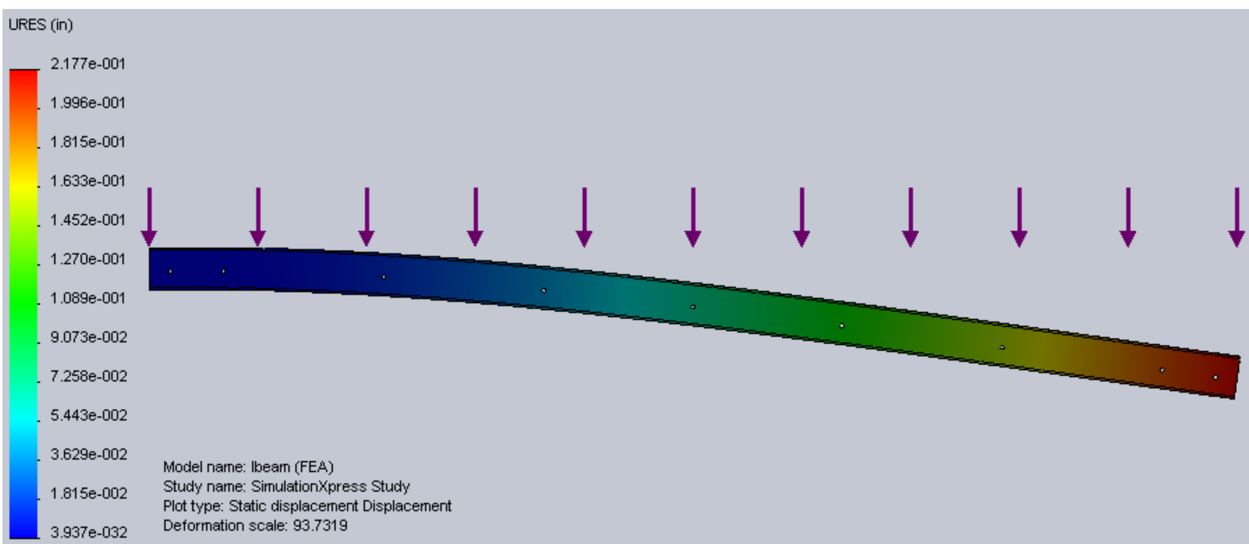


Figure 37: FEA Relative Deformation (in) I - Beam - Side View (Deformation scale 93.7319)

The maximum stress in the I-beam occurs at the edge of the idler shaft hole for the end idler shaft. This stress is only about one third of the yield stress of the A36 steel. The deflection of the beam is, as expected, maximized at the free end of the beam. Therefore, the amount of deflection is negligible (less than ¼ in.). The results of the finite element analysis further justify the selected dimensions and material of the I-beam.

B.1.1.2 C-Channel Beam

To perform a finite element analysis of the C-channel beam in the maximum stress position we applied the stress of 1200 lb to the top flanges of the C-channel beam, on the area where the last two sets of the upper roller idler wheels contact the flanges when the I-beam is fully extended. We then fixed the area where the C-channel beam attaches to the lead screw plate. These fixtures were chosen to give a more accurate picture of how the C-channel beam will behave in service. Figure 47 shows the stress distribution on the end of the C-channel beam and Figure 48 shows the displacement.

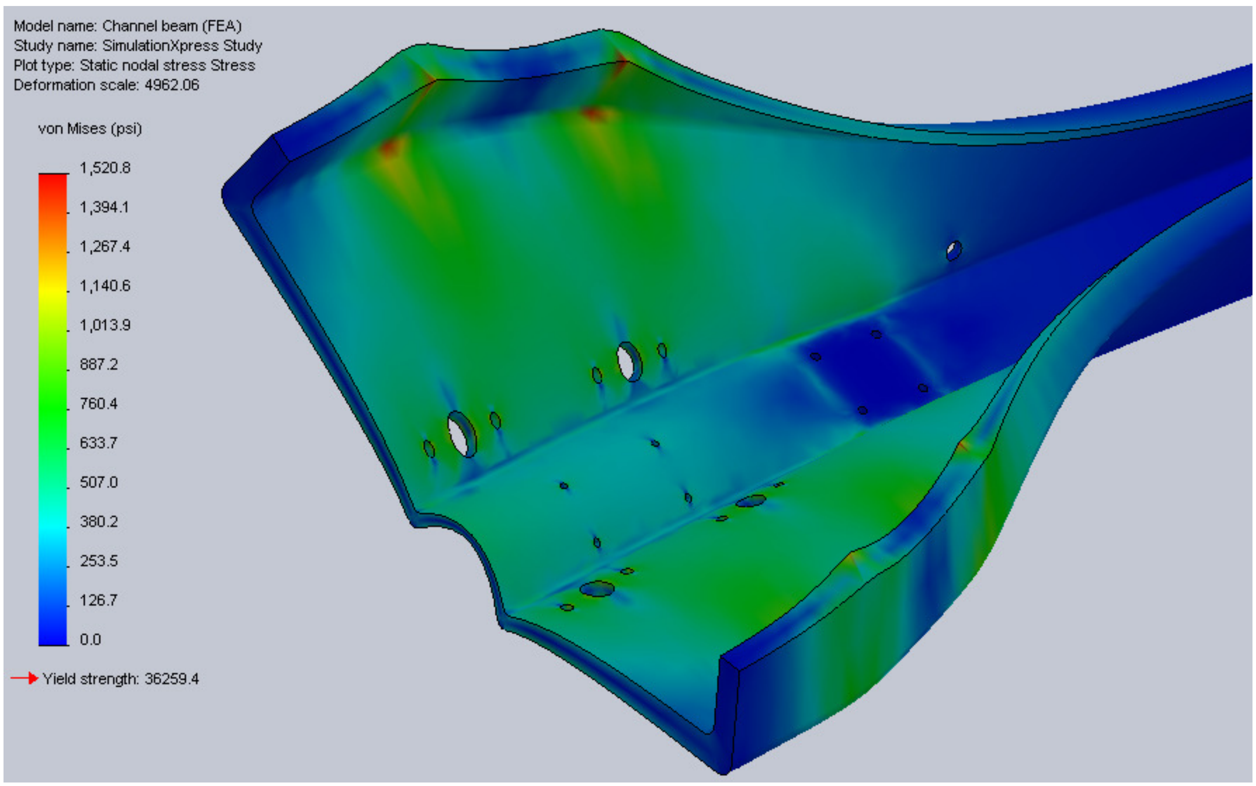


Figure 38: FEA von Mises (psi) Box Beam - Fully Extended Translation Model (Deformation scale 4962.06)

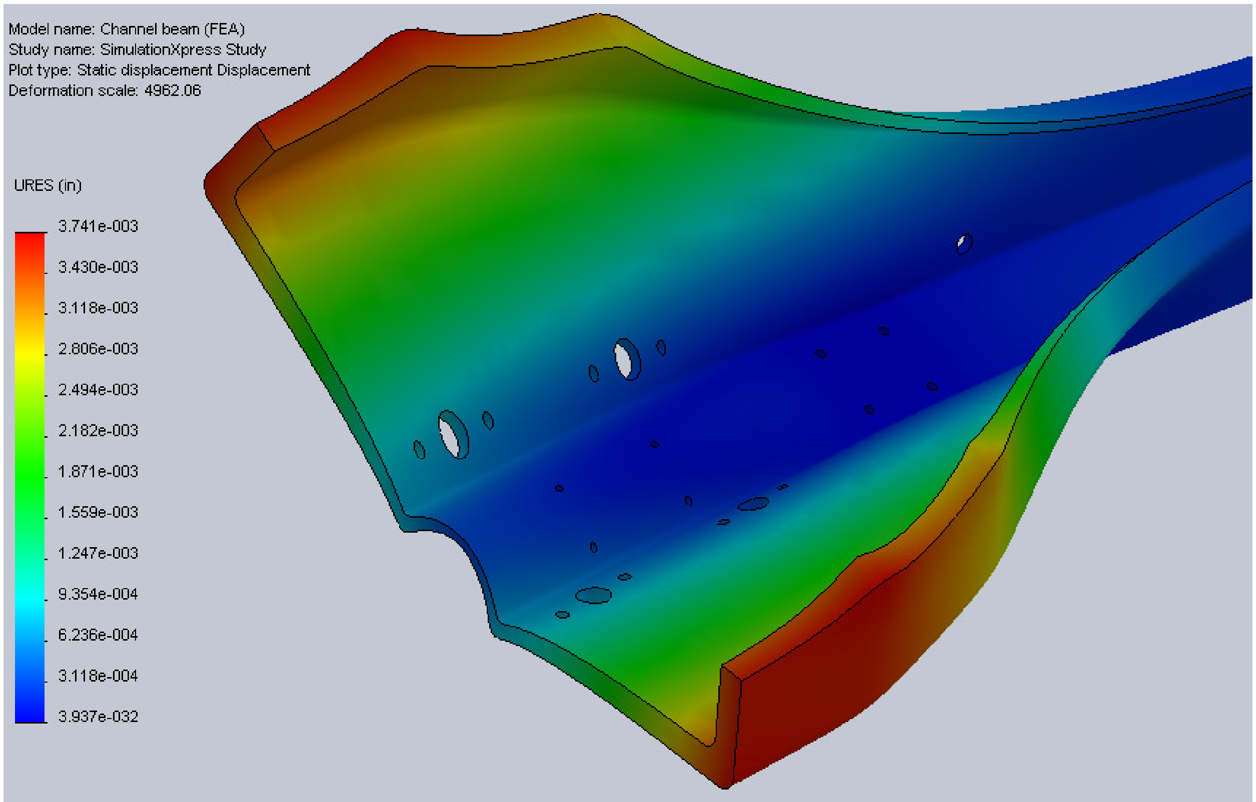


Figure 39: FEA Relative Deformation (in) Box Beam – Fully Extended Translation Model (Deformation scale 4962.06)

The maximum stress in the C-channel beam occurs in the area of the upper flanges where the upper roller idler wheels are contacting it. This stress however, is substantially lower than the yield stress of the A36 steel. The maximum deflection occurs in the same area as the maximum stress, but the amount of deflection is basically negligible (less than 4 thousands of an inch).

The results of the finite element analysis justify the selected size of the C-channel beam. The loading applied is the maximum the C-channel beam will experience and yet the factor of safety is still very high. We were unable to perform a simple evaluation of the beam by hand so we decided to select an over-sized beam to be certain that it would not fail prematurely.

B.1.1.3 Roller Support

The following equations and material properties were used when analyzing the roller support section of the translating assembly. We will be analyzing the lower roller drive shafts, the lower roller idler shafts, the upper roller idler shafts, and the roller drive shaft bearings.

B.1.1.3.1 Lower Roller Drive Shafts

The following equations and material properties were used in order to analyze the lower roller drive shafts. Since all of the components are held in place with set screws and self-locking retaining rings the shaft is assumed to be a constant diameter.

The material properties of the 1020 CD steel are:

Yield strength σ_Y = 54 ksi

Tensile strength s_u = 64ksi

Endurance strength s_n = 26 ksi

$$Actual\ endurance\ strength\ s_n{}' = s_n(C_m)(C_{st})(C_R)(C_S) = 18.45\ ksi$$

Using a factor of safety of 2 and assuming that the shaft behaves like a simply supported beam with multiple point loads we obtained:

$$\sigma_{all} = \frac{\sigma_y}{2} = 27\ ksi$$

Sprocket n = 352.4 rpm

Power P = 1 hp

The max force on the drive wheels is 1200 lb so we can therefore assume that:

F_B = 600 lb and F_C = 600 lb

The force from the sprocket is calculated as:

$$Torque\ on\ sprocket\ T = \frac{63000P}{n} = 179.5\ in - lb$$

$$Force\ on\ sprocket\ FE = \frac{T}{0.5D_{sprocket}} = 41.75\ lb$$

Applying force and moment equations yields:

R_A = 591 lb and R_D = 651 lb

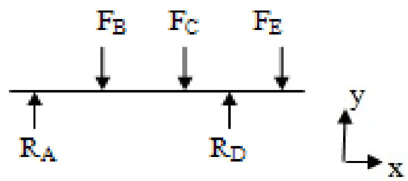


Figure 40: Force diagram

The shear force diagram:

A to B = 591 lb; B to C = -9 lb; C to D = -609 lb; D to E = 42 lb

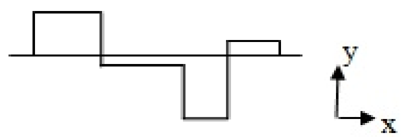


Figure 41: Shear diagram

The bending moment diagram:

M_B = 2040.7 in-lb; M_C = 2018.2 in-lb; M_D = -84.8 in-lb

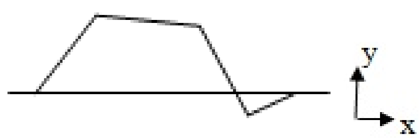


Figure 42: Moment diagram

The section modulus of the shaft is calculated to be:

$S_{shaft} = 0.25\pi r^3 = 0.0982$ in^3

The maximum stress in the shaft is calculated to be:

$$\sigma_{shaft} = \frac{M_{max}}{S_{shaft}} = 20.781\ ksi < \sigma_{all}$$

The maximum stress in the shaft is lower than the allowable stress, meaning that the selected shaft size can withstand the worst case scenario forces. Although σ_{shaft} is greater than the actual endurance strength, the shaft will not be experiencing this maximum stress on a regular basis and as such the size selected is adequate.

To perform the finite element analysis we applied a distributed load of 600 lb onto the area of the shaft where each drive wheel is located. We also applied a 41.75 lb load onto the area where the driven sprocket hub is located. We then fixed the two areas where the bearings are located. Figure 52 shows the resulting stress distribution of the loads applied and Figure 53 shows the displacement of the drive shaft.

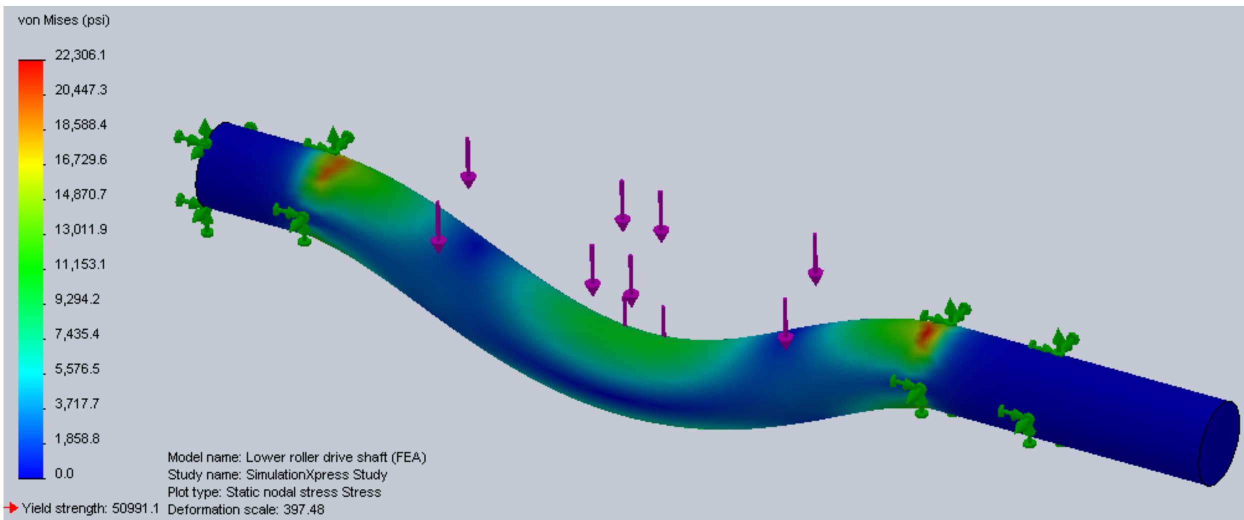


Figure 43: FEA von Mises (psi) Drive Shaft - (Deformation scale 397.48)

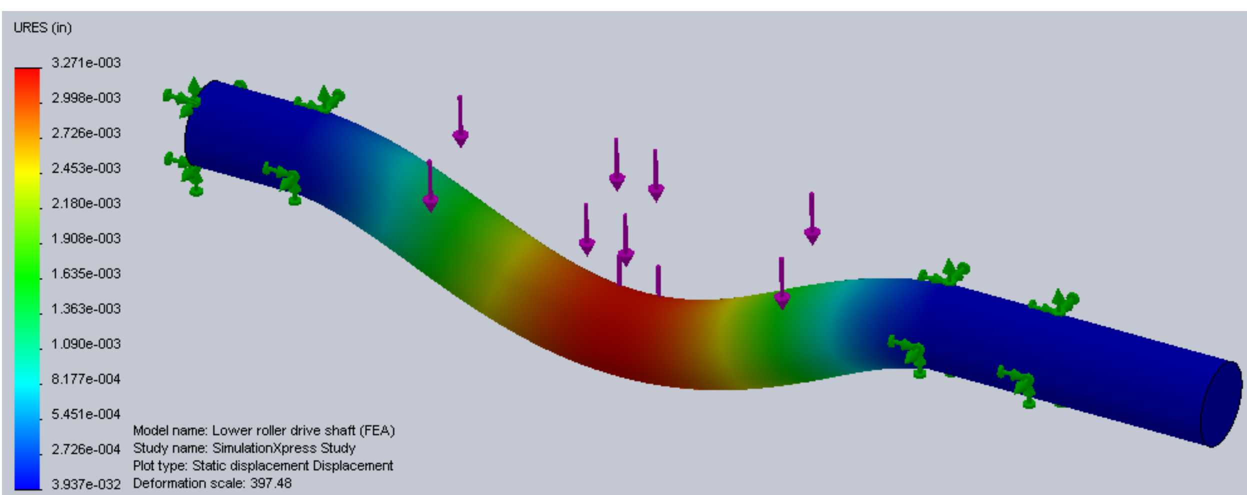


Figure 44: FEA Relative Deformation (in) Drive Shaft (Deformation scale 397.48)

The maximum stress on the shaft occurs at the edge of both bearings. This stress is only about one fifth of the yield strength of the 1020 cold drawn steel. The displacement is maximized at the point between the drive rollers, but the value is negligible. These results further justify our size and material selection of the drive shaft.

B.1.1.3.1 Lower Roller Idler Shafts

The following equations and material properties were used in the analysis and sizing of the lower roller idler shafts in the translating part of our design.

The material properties of the 1020 CD steel are:

Yield strength σ_Y = 54 ksi

Tensile strength s_u = 64ksi

Endurance strength s_n = 26 ksi

Using a factor of safety of 2 and assuming the shaft behaves like a simply supported beam, with two point loads we obtain:

$$\sigma_{all} = \frac{\sigma_y}{2} = 27\ ksi$$

The force on the idler wheels is assumed to be 1200 lbs again to allow for greater safety and strength.

F_B = 600 lb and F_C = 600 lb (max on each wheel)

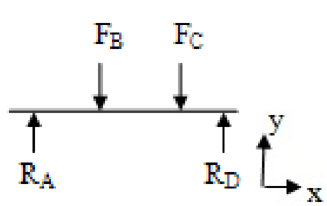


Figure 45: Force diagram

The shear force diagram:

A to B = 600 lb; B to C = 0 lb; C to D = -600 lb

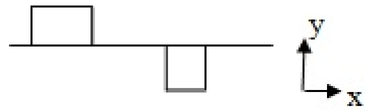


Figure 46: Shear diagram

The bending moment diagram:

M_B = 1500 in-lb; M_C = 1500 in-lb

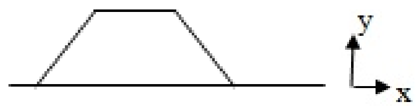


Figure 47: Moment diagram

The section modulus of the shaft is calculated to be:

$$S_{shaft} = 0.25\pi r^3 = 0.0982 \text{ in}^3$$

The maximum stress in the shaft is calculated to be:

$$\sigma_{shaft} = \frac{M_{max}}{S_{shaft}} = 15.275\ ksi < \sigma_{all}\ and\ s_n$$

The maximum stress in the shaft is lower than both the allowable stress and the endurance strength. This means that the selected shaft size can withstand these maximum forces and will not prematurely fail.

To perform the finite element analysis we applied a distributed load of 600 lbs onto the area of the shaft where each idler wheel is located. We then fixed the two areas where the channel beam side walls are located. Figure 57 shows the resulting stress distribution of the loads applied and Figure 58 shows the displacement of the idler shaft.

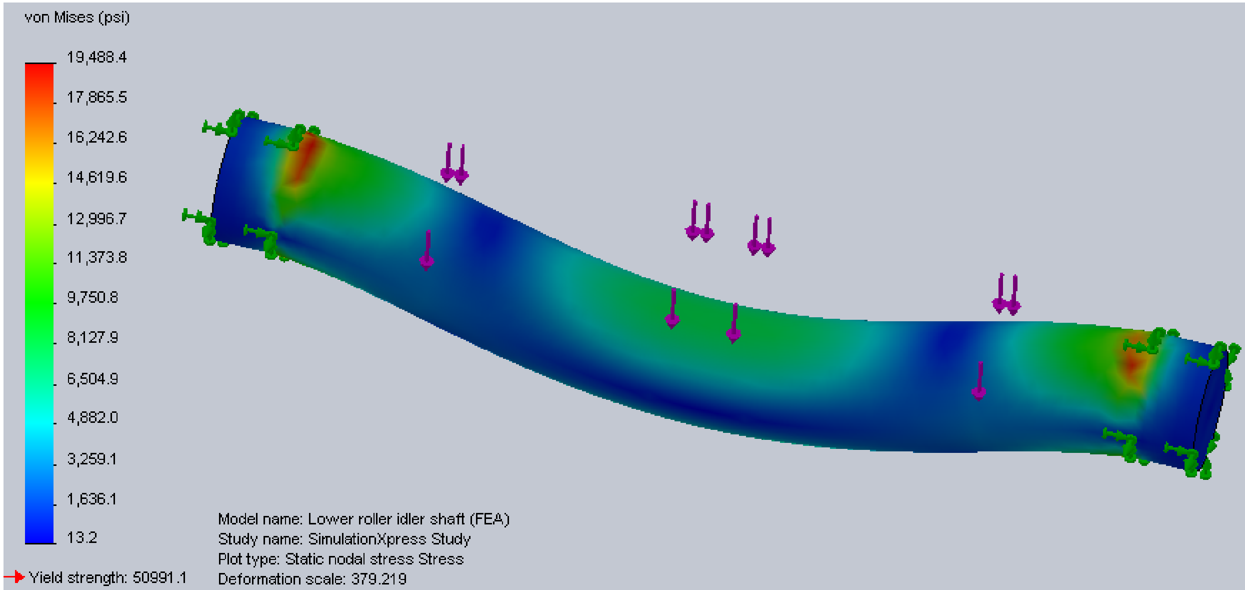


Figure 48: FEA von Mises (psi) Lower Roller Idler Shaft - (Deformation scale 379.219)

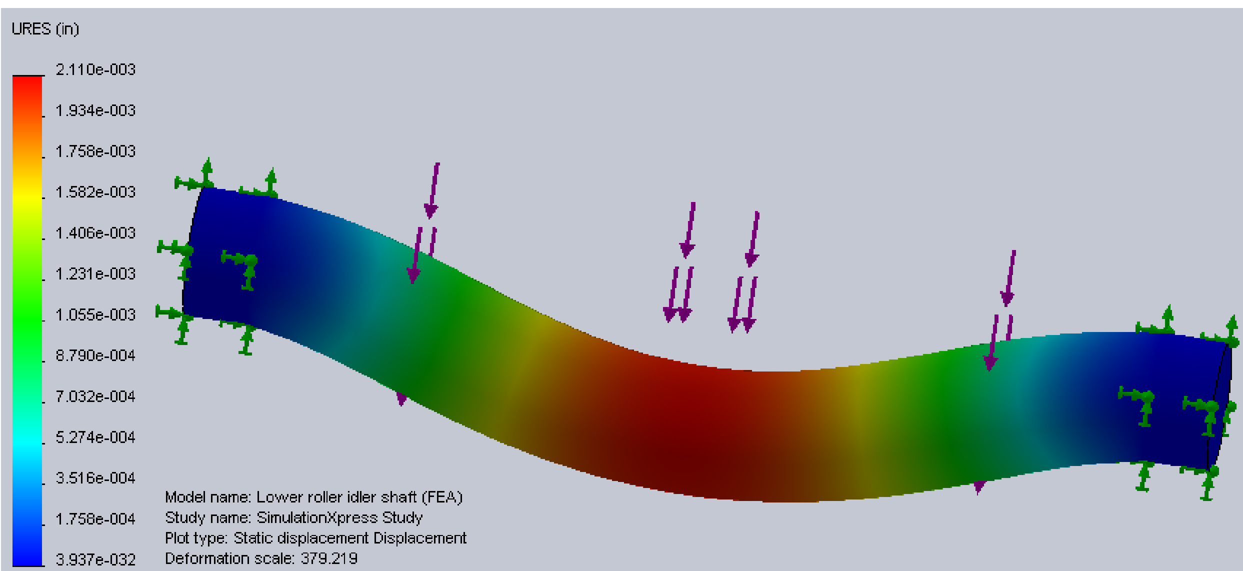


Figure 49: FEA Relative Deformation (in) Lower Roller Idler Shaft - (Deformation scale 379.219)

The maximum stress in the shaft occurs at edge of both channel beam side walls. This stress is only about one half of the yield strength of the 1020 cold drawn steel. The displacement is maximized at the point between the idler rollers, but the value is negligible. These results further justify our size and material selection of the lower roller idler shaft.

B.1.1.3.3 Upper Roller Idler Shafts

The following equations were used in the analysis of the upper roller idler shafts for the translating component of our design.

The material properties of the 1020 CD steel are:

Yield strength σ_Y = 54 ksi

Tensile strength s_u = 64ksi

Endurance strength s_n = 26 ksi

Using a factor of safety of 2 and assuming the shaft behaves like a simply supported beam with one point load in the center we obtain:

$$\sigma_{all} = \frac{\sigma_y}{2} = 27\ ksi$$

Again it is assumed that the absolute maximum load these idler wheels will have to withstand is 1200 lbs.

F_A = 600 lb and F_C = 600 lb (max on each wheel)

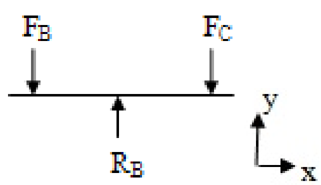


Figure 50: Force diagram

The maximum bending moment and section modulus of the shaft are calculated to be:

M_{max} = 600(2.625) = 1575 in-lb

$S_{shaft} = 0.25\pi r^3 = 0.0414\ in^3$

The maximum stress in the shaft is calculated to be:

$$\sigma_{shaft} = \frac{M_{max}}{S_{shaft}} = 38.043\ ksi$$

This stress is more than the allowable stress but below the yield stress. This is due to the large load that was assumed to be acting on the shaft, however, this load will rarely if ever be applied to the shaft and was selected as the absolute worst case scenario. Under regular service the size and material specified for the shaft should provide ample support so that the shaft will not fail prematurely.

To perform the finite element analysis we applied a distributed load of 600 lbs onto the area of the shaft of where each idler wheel is located. We then fixed the area in the middle of the shaft where the I-beam center web is located. Figure 60 shows the resulting stress distribution of the loads applied and Figure 61 shows the displacement of the idler shaft.

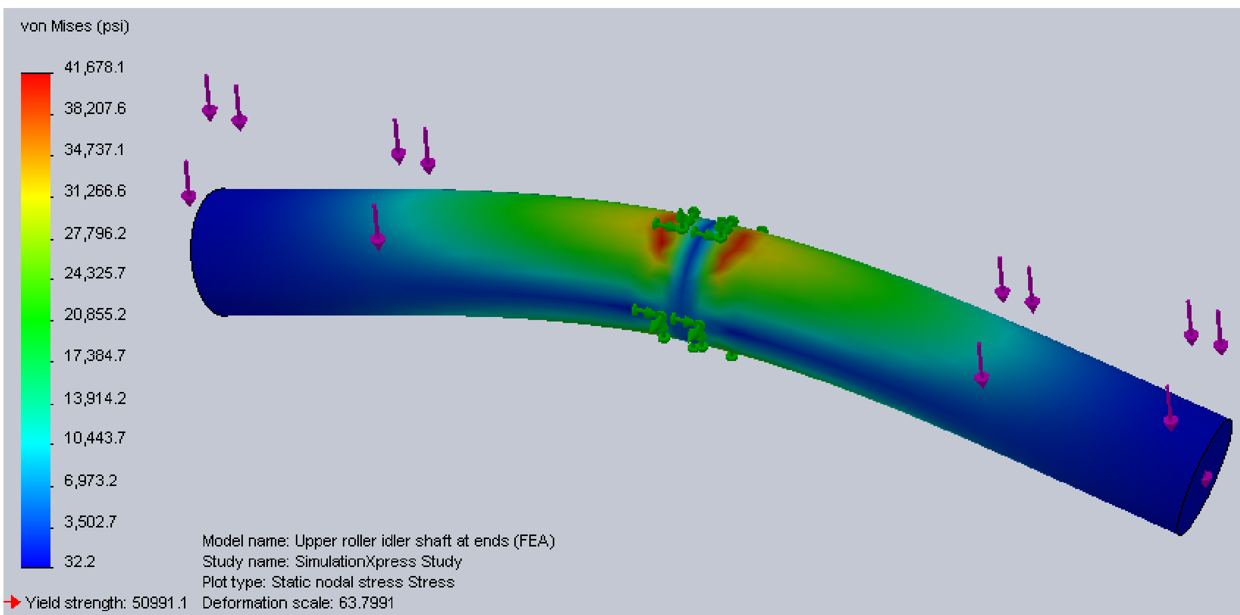


Figure 51: FEA von Mises (psi) Upper Roller Idler Shaft - (Deformation scale 63.7991)

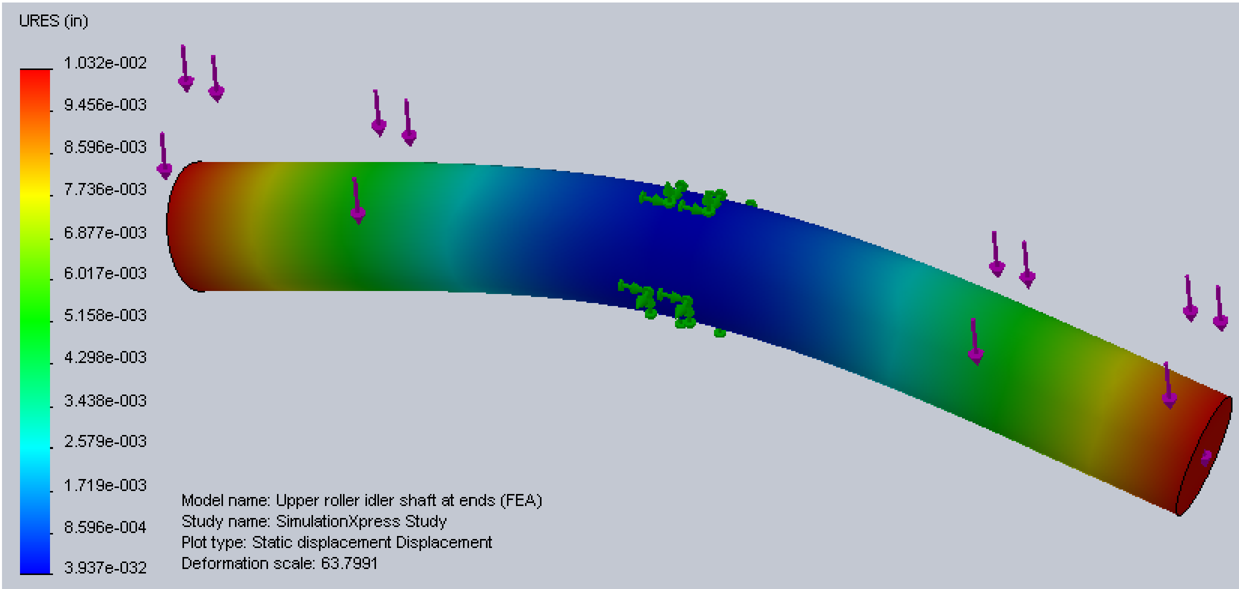


Figure 52: FEA Relative Deformation (in) Upper Roller Idler Shaft - (Deformation scale 63.7991)

The maximum stress in the shaft occurs at edge where the shaft meets the I-beam center web. Although this stress is very near the yield stress of the material, the loading has already incorporated a factor of safety; therefore this is not an area of concern. The displacement is maximized at the edges of the shaft, but the value is negligible. These results further justify our size and material selection of the lower roller idler shaft.

B.1.1.3.4 Roller Drive Shaft Bearings

The following equations were used to select a proper bearing size for the horizontal drive system.

Radial Load R = 600 lb per bearing

n = 351.4 rpm

Life = 20000 hrs

f_N = 0.46 and f_L = 3.4

$$Dynamic\ radial\ load\ C = R\left(\frac{f_L}{f_N}\right) = 4435\ lb$$

The bearing selected from McMaster-Carr [2] has a dynamic radial load of 3140 lbs. Since the stacker will not be subjected to the maximum radial load of 600 lbs on a regular basis this bearing was deemed suitable.

A finite element analysis was not required because the hand calculations provided enough information to select a proper sized bearing.

B.1.1.4 Drive Components

The following equations were used in the analysis of the different components of the drive system. The components that were analyzed were: the motor for the drive wheels, the roller sprockets and the keys for the sprockets.

B.1.1.4.1 Motor for Drive Wheels

The following equations were used to specify a motor for the lateral drive system. The torque required to overcome the friction in the bearings can be calculated using:

$$M_f = F\mu_f\left(\frac{D_{bearing}}{2}\right)$$

The force on the bearing is chosen to be F = 600 lbs and using a conservative coefficient of friction of a single groove ball bearing of $\mu_f = 0.01$ we get $M_f = 30$ in-lb of torque required to overcome the friction in the bearings.

Converting that torque into the power required to turn the drive shaft at 350 rpm we get:

$$P = \frac{M_f(n)}{63000} = 0.17\ hp$$

Therefore the motor selected provides ample torque for the lateral drive system.

B.1.1.4.2 Roller Sprockets

The following equations were used in order to determine the sizing and spacing of the sprockets. The desired lateral speed of the translating assembly was selected as 36 in/s, based on the equations:

$2\pi r_{drive\ wheel}$ = 6.28 in. of travel per turn of drive wheel

$$n_1 = \frac{36}{6.28} \, x \; 60 \; s = 344 \, rpm \; is \; desired \; rotational \; speed \; of \; drive \; wheels$$

Using the rotational speed of the motor we selected a chain:

n_2 = 1725 rpm for the motor

Choosing a no. 40, ½ in. pitch, N_1 = 11 tooth sprocket that can handle up to about 4.7 hp

The sprocket sizes were calculated using:

$$N_2 = N_1 \left(\frac{n_2}{n_1}\right) = 54 \, teeth$$

$$n_1 = n_2 \left(\frac{N_1}{N_2}\right) = 351.4 \, rpm \; which \; is \; nearly \; the \; desired \; speed$$

$$D1 = \frac{p}{\sin\left(\frac{180}{N_1}\right)} = 1.775 \; in.$$

$$D2 = \frac{p}{\sin\left(\frac{180}{N_2}\right)} = 8.599 \; in.$$

To get a usable center distance we chose a chain length L = 66 pitches and we used this to calculate the center distance as:

$$C = 0.25 \left\{ L - \frac{(N_2 + N_1)}{2} + \left[\left(L - \frac{(N_2 + N_1)}{2} \right)^2 - \frac{8\left(N_2 - N_1\right)^2}{4\pi^2} \right] \right\} 0.5 = 15.21 \, pitches = 7.605 \; in.$$

$$\theta 1 = 180 - sin^{-1} \left[\frac{(D_2 - D_1)}{2C} \right] = 126.7^o > 120^o \text{ therefore it is possible}$$

In order to achieve that center distance, it is required that the motor be lowered ¼ in. This can be accomplished with a spacer.

A finite element analysis was not performed because the hand calculations were deemed sufficient to select the sprockets.

B.1.1.4.3 Keys for the sprockets

The following analysis was completed in order to size and analyze the keys that are required in our translating assembly. For a 1 in. diameter a ¼ in. key is recommended. Knowing that the key material has the least strength we calculated the length as:

$$T = \frac{63000P}{n} = 179.8\, in - lb$$

Factor of safety N = 3

W = 1/8 in.

Yield stress of shaft σ_Y = 54 psi

$$Length\ L = \frac{4TN}{DW\sigma Y} = 0.32\ in.$$

Therefore, the 2 in. key specified is more than enough to handle the torque from the motor.

A finite element analysis was not required because the difference between the required length and actual length offers enough of an allowance for safety that the hand calculations sufficed.

B.1.2 Lift Assembly

The following equations were used to determine the lift assembly components.

TABLE VIII: OVERALL WEIGHT DISTRIBUTION

Part	lb/ ft.	Length (ft)	Weight (lb)
W8 x 24 Square Steel Lubing	48.9	15	733.5
8 x 8 x ½" Steel W- Beam	24	17	408
Max Truss Weight / Lift	-	-	2400lbs/ 6 Lifts = 400
Motors, Gears etc. (approx.)	-	-	450
Total			1991.5

Approximately half of the total weight is stationary and remains evenly distributed above the main lift assembly. The I-beam, including its rollers and the truss, are the only translating masses. Therefore, the load experienced on the load screws varies as the mass translates side to side. Identifying the amount of this load is necessary to accurately measure the maximum loads applied to the lead screws.

$$Total\ Weight\ (lb)/Lift\ \cong 2000\ lbs$$

$$Total\ Weight\ Translating(lb)/Lift\ \cong 900\ lb$$

When calculating the maximum weight per lead screw it was assumed that all of the translating weight would be applied above one set of lead screws, while the stationary weight would remain evenly distribute amongst all four lead screws.

$$Stationary\ Weight\ (lb)/Lead\ Screw\ \ \cong {}^{1100}/_{4} = 275\ lb$$

$$Max\ Translating\ Weight\ (lb)/Lead\ Screw\ \cong {}^{900}/_{2}\ = 450\ lb$$

$$Total\ Weight\ (lb)/Lead\ Screw * Safety\ Factor \cong (275 + 450) * 1.5 = 1090\ lb$$

B.1.2.1 Lead Screw Design Calculations

In the selection process of a screw, the tensile and shear stresses must be examined.

$$Tensile\ Stress\ Area\ (A_t) = \frac{F}{\sigma_d} = \frac{1090\ lb}{10\ 000\ lb/in^2} = 0.109\ in^2$$

$$Shear\ Stress\ Area\ (A_s) = \frac{F}{\tau_d} = \frac{1090\ lb}{5\ 000\ lb/in^2} = 0.218\ in^2$$

From this data a lead screw diameter of ½ in. was selected.

B.1.2.2 Drive System

Minimum Length of Yoke

$$h_{min} = 0.218\ in^2 \left[\frac{1.0\ in.}{0.7278\ in^2}\right] = 0.30\ in.$$

Lead Angle

$$\lambda = tan^{-1}\frac{p}{\pi D_p} = \ tan^{-1}\frac{0.10}{\pi(0.4306)} = 4.23°$$

Lift Torque

$$T_u = \frac{FD_p}{2}\left[\frac{\cos\emptyset\tan\lambda\ +\ f}{\cos\emptyset - ftan\lambda}\right] = \frac{(1090\ lb)(0.4306\ in)}{2}\left[\frac{\cos(14.5)\tan(4.23) +\ 0.15}{\cos(14.5) - 0.15\tan(4.23)}\right] = 55\ lb \cdot in$$

Lower Torque

$$T_d = \frac{FD_p}{2}\left[\frac{f - \cos\emptyset\tan\lambda}{\cos\emptyset + ftan\lambda}\right] = \frac{(1090\ lb)(0.4306\ in)}{2}\left[\frac{0.15 - \cos(14.5)\tan(4.23)}{\cos(14.5) + 0.15\tan(4.23)}\right] = 19\ lb \cdot in$$

Efficiency

$$e = \frac{FL}{2\pi T_u} = \frac{(1090\ lb)(0.10\ in)}{2\pi(55\ lb \cdot in)} = 31.5\ \%$$

Required Velocity

$$V = \frac{30\ in}{30\ s} = 1.0\ in/s$$

Rotational Speed

$$n = \frac{1.0\ in}{s}\frac{1\ rev}{0.1\ in}\frac{60\ s}{min} = 600\ rpm$$

Required Power

$$P = \frac{Tn}{63000} = \frac{(55)(600\ rpm)}{63000} = 0.52\ hp$$

B.1.2.3 Lead Screw Bearings Design Calculations

$$F_{Sprocket} = \frac{T}{D_s/2} = \frac{55\ lb \cdot in}{5.7\ in/2} = 19.30\ lbs$$

Radial Force= 19.30 lbs

Thrust Force = 1090 lbs

$$Fr_{Bottom} = 19.30\left(\frac{2}{34.75}\right) = 1.11\ lbs$$

$$Fr_{Top} = 19.30\left(\frac{32.75}{34.75}\right) = 18.19\ lbs$$

$$T_A = 1090\ lbs$$

$$P_{Bottom} = 0.4Fr_A + 0.5\frac{Y_A}{Y_B}Fr_B + Y_AT_A;\ where\ Y_A\ and\ Y_B = 1.75$$

$$P_{Bottom} = 0.4(1.11) + 0.5(18.19) + 1.75(1090) = 1917.04\ lb$$

$$P_{Top} = 18.19\ lbs$$

Design Life = 100 000 hrs

$$L_d = 100\ 000(600\ rpm)\left(\frac{60\ min}{hr}\right) = 3.6\ x\ 10^9\ rev$$

Equivalent Dynamic Radial Load

$$C_i = P_i\left(\frac{L_d}{10^6}\right)^{1/k},where\ k = 3.33$$

$$C_{Bottom} = 1917.04\left(\frac{3.6\ x\ 10^9}{10^6}\right)^{1/3.33} = 22417\ lbs$$

$$C_{Top} = 18.19\left(\frac{3.6\ x\ 10^9}{10^6}\right)^{1/3.33} = 213\ lbs$$

B.1.2.4 Lead Screw Mounting Plate

A FEA analysis of the lead screw mounting plate was done to evaluate the stresses and relative deformation when fully loaded. The plate was fixed on the bottom face at the area of contact between the plate and the lead screw nut flange. To determine the force applied by the translating assembly onto the plate, half of the stationary weight and all of the translating weight were used times a safety factor of 1.5.

$$Total\ Weight\ on\ Plate\ (lb) * Safety\ Factor = \left(\frac{1100}{2} + 900\right) * 1.5 = 2175\ lb$$

The FEA analysis of such loading is shown below in Figures 62-65. The deformation scale for these figures is 843.977 and is exaggerated for illustration purposes.

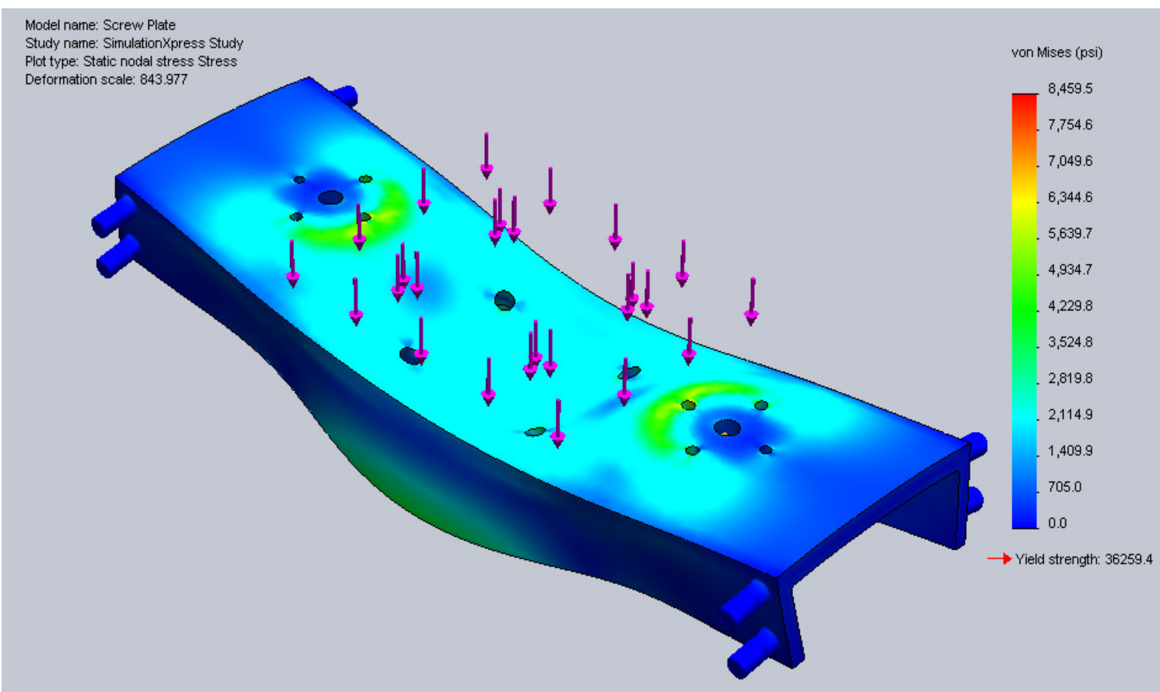


Figure 53: FEA von Mises (psi) Lead Screw Mounting Plate - Top View (Deformation scale 843.977)

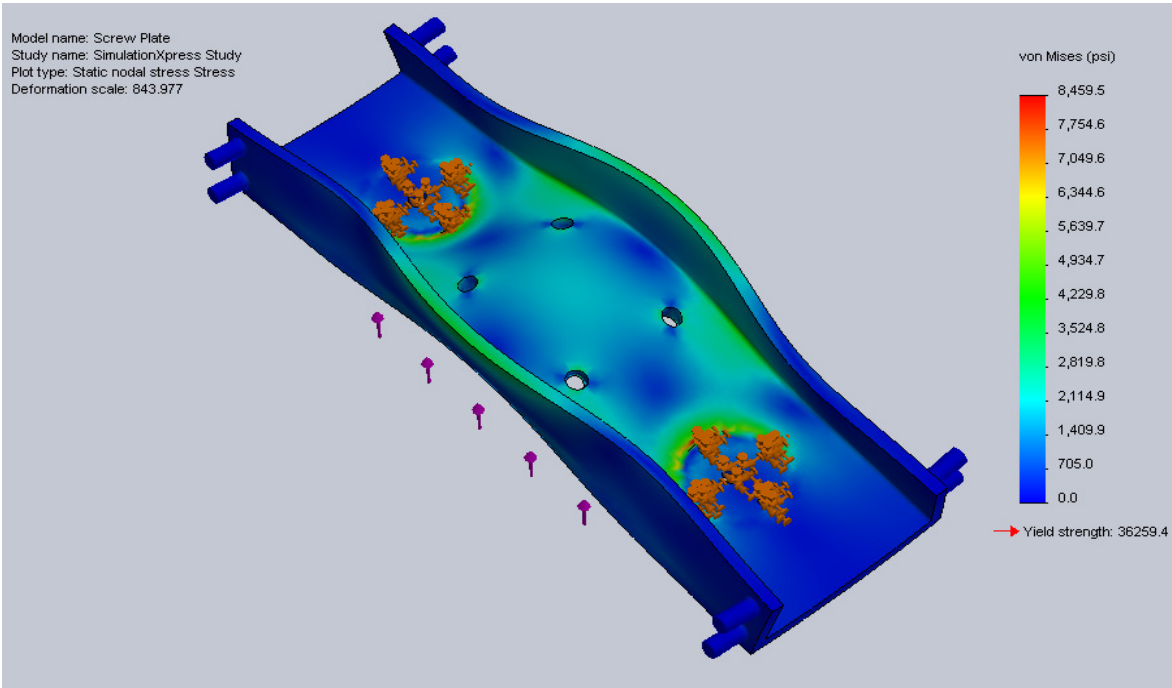


Figure 54: FEA von Mises (psi) Lead Screw Mounting Plate - Bottom View (Deformation scale 843.977)

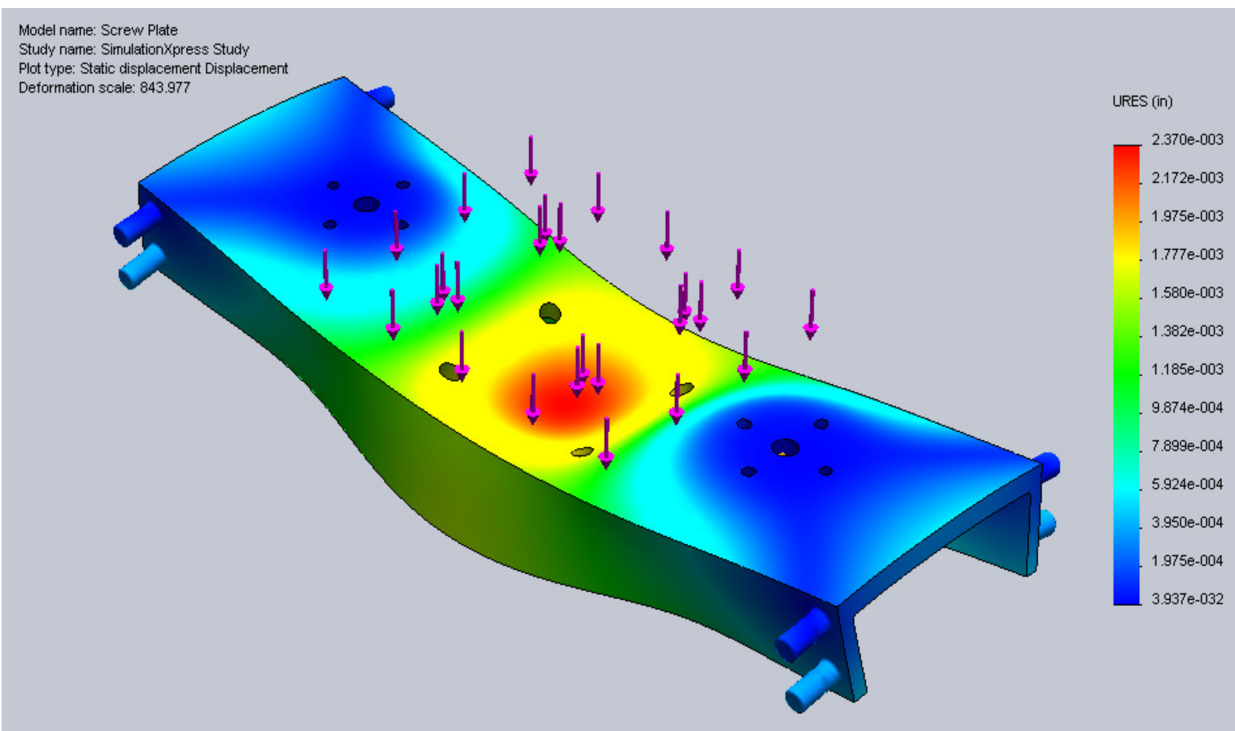


Figure 55: FEA Relative Displacement (in) Lead Screw Mounting Plate - Top View (Deformation scale 843.977)

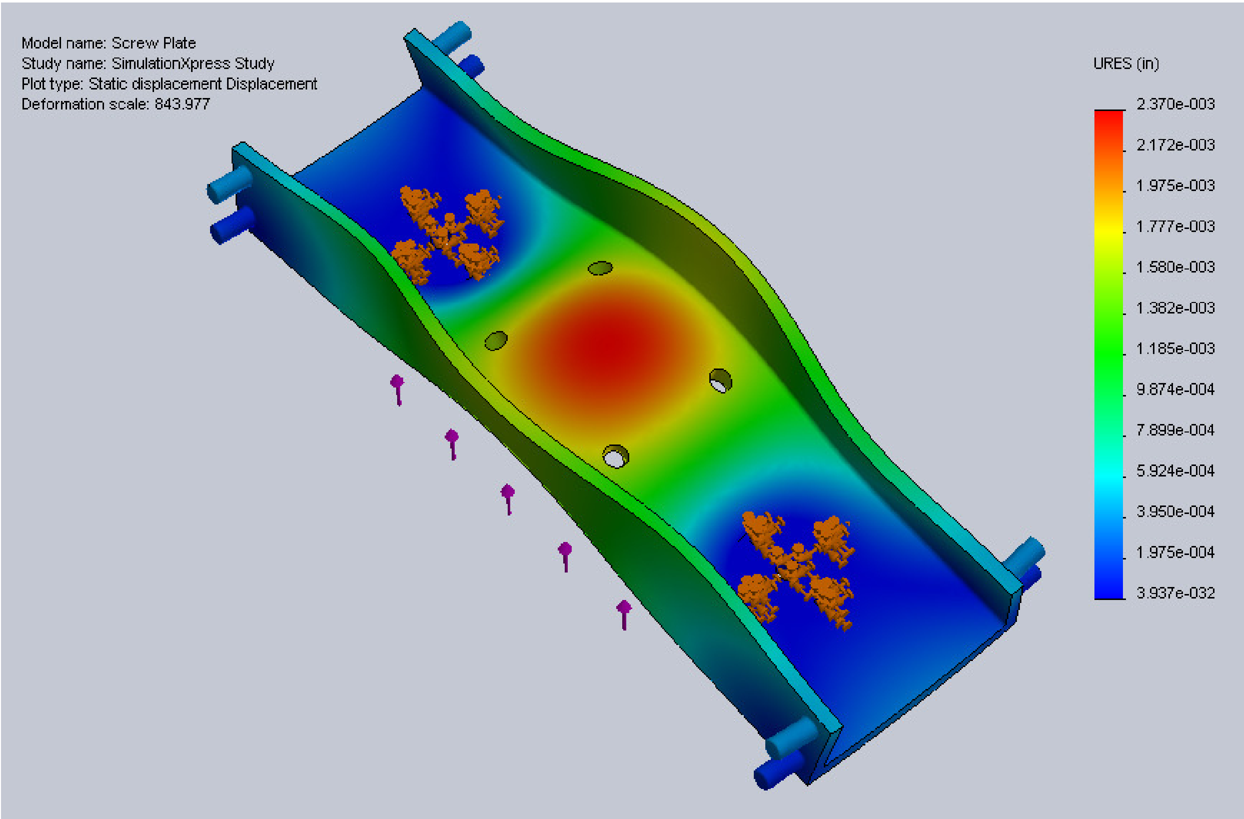


Figure 56: FEA Relative Displacement (in) Lead Screw Mounting Plate - Bottom View (Deformation scale 843.977)

From the FEA test, we can see that the maximum Von Mises stress in the screw plate occurs at the outer radius of the lead screw nut flange. The stresses were primarily focused at the interface between the plate and the flange (as shown in Figure 63), but also at in the lower middle region of the plate (as shown in Figure 62). Despite the level of stresses shown in the Figures 62 and 63 the stress was found to be well below the acceptable yield strength of the A36 material chosen for the part and yielded a factor of safety of almost 4.3. This high safety factor was picked for this particular part due to the repeated fluctuating loading that will be applied to the plate. Additionally the greater the factor of safety the less relative deformation that takes place, which allows more of the fluctuating load to be applied into the walls of the Base support.

The analysis of the displacement of the part yielded similarly acceptable results with a maximum displacement of approximately 0.0024 in. (0.05986 mm as shown in Figure 64 and 65). This deformation is next to negligible, and is therefore not an area of concern in our design. This maximum displacement occurs at the very center of the plate where the translating assembly is mounted to the plate. This increased deformation makes sense due to the proximity to the large mounting bolt holes.

B.1.3 Stacking Platform

The following equations were used to analyze the stopping rod for the stacking platform.

B.1.3.1 Stopping Rod

The following equations were completed to size and analyze the steel rod used as part of the stopping platform.

The material properties of 1020 CD steel are:

Yield strength σ_Y = 54 ksi

Tensile strength s_u = 64 ksi

Endurance strength s_n = 26 ksi

Using a factor of safety of 2 and assuming the stopping rod behaves as a simple cantilever beam with a point load (F_f) at the free end we calculated:

$$Allowable\ stress\ \sigma_{all} = \frac{\sigma_y}{2} = 27\ ksi$$

Coefficient of static friction between wood and steel is about μ_S = 0.4:

Friction force $F_f = 600\mu_S$ = 240 lb

Maximum moment M_{max} = 32F = 7680 in-lb

The section modulus of the shaft is:

$S_{shaft} = 0.25\pi r^3 = 0.331\ in^3$

The stress in the shaft is found to be:

$$\sigma_{shaft} = \frac{M_{max}}{S_{shaft}} = 23.202\ ksi$$

The stress in the shaft is less than the allowable stress and the endurance strength. Therefore, the size selected for the stopping rod is sufficient and will not result in premature failure.

To perform the finite element analysis we applied a distributed 240 lbs load onto the area of the rod that the last truss will push against. We then fixed the area at the bottom of the rod that is inserted into the railway tie. Figure 56 shows the resulting stress distribution of the load applied and Figure 57 shows the displacement of the idler shaft.

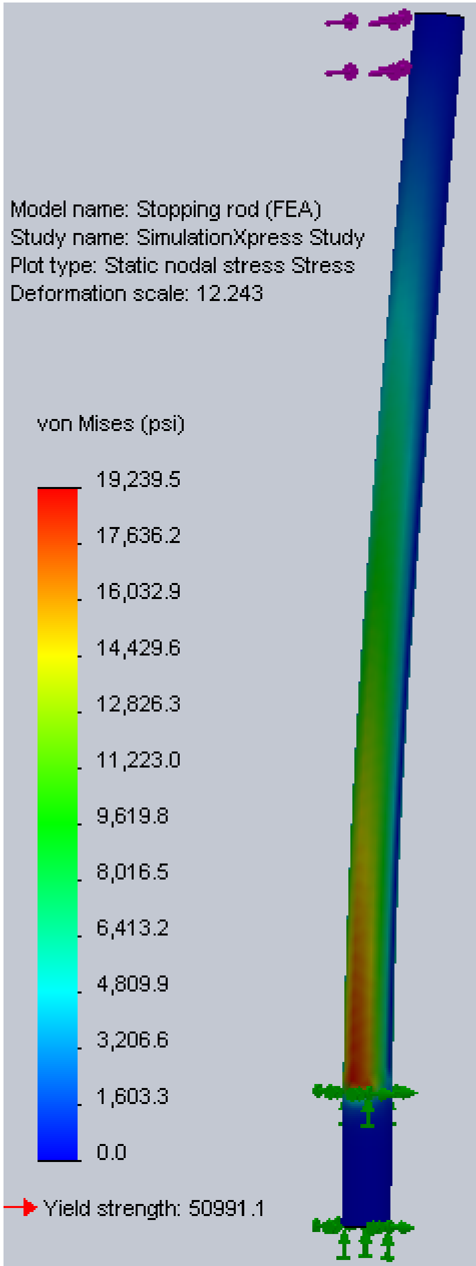


Figure 57: FEA von Mises (psi) Stopping Rod - (Deformation scale 12.243)

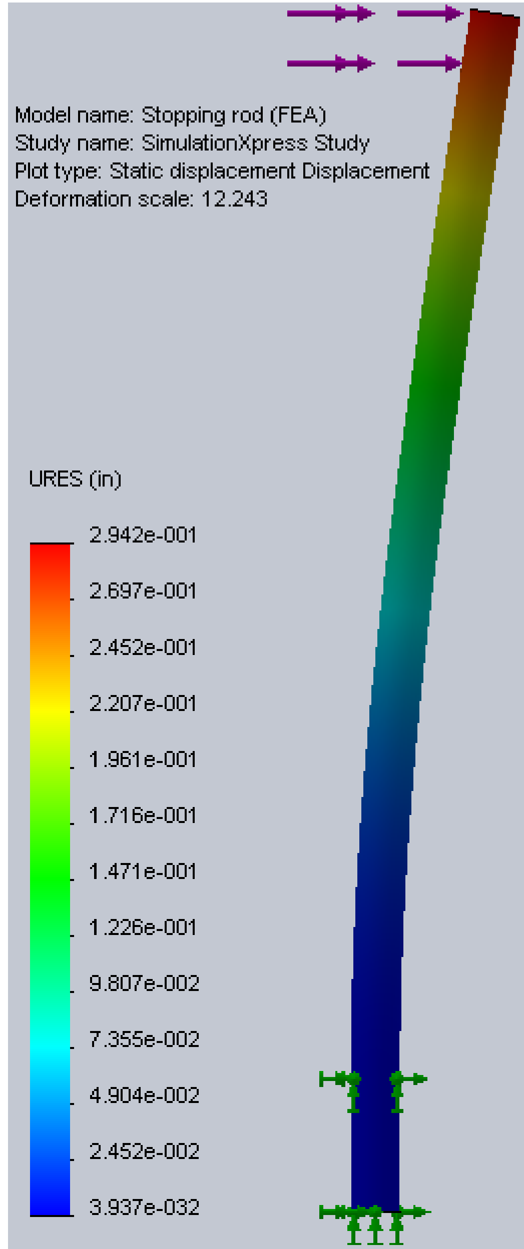


Figure 58: FEA Relative Displacement (in) Stopping Rod (Deformation scale 12.243)

m

The maximum stress in the rod occurs at edge where the shaft meets the railway tie. This stress, however, only about half of the yield stress. The displacement is maximized at the free end of the rod, but the value is negligible. For stability concerns we added a gusset to the bottom of the rod, which will decrease the maximum stress and displacement in the rod. These results further justify our size and material selection of the stopping rod.

B.1.4 Roller Platform Drive Mechanism

The following sections describe the powering of the rollers of the main platform. The following section will detail the equations used to size the motor as well as the chain drives.

B.1.4.1 Drive Mechanism Selection

It was decided that an acceptable final roller drive speed would be approximately 400 rpm. The horizontal speed of a truss was then calculated as:

$$\frac{400rpm}{60} * \frac{1}{3\pi} = 7\ ft/sec$$

The truss will travel slower than the running speed of a human and thought to be acceptable.

Due to high torque requirements, a 1 hp motor was selected. This motor is rated for this power transmission at 1725 rpm. Therefore, the torque output of the motor was calculated as:

$$Hp = 1 = \frac{Tn}{63000} = \frac{T * 1725}{63000}$$
$$T = 36.52\ lb\ in$$

Therefore, the small sprocket turns at 1725 rpm and transfers 1 hp. Assuming that a 10 tooth small sprocket, the tables for ANSI #60 chain is rated for approximately 6.45 hp. Therefore, this design is acceptable given a factor of safety of 2. The design hp is therefore:

$$design\ hp = 2(1\ hp) = 2\ hp$$

Given the different speeds of the driving and driven sprocket, the required speed ratio is:

$$speed\ ratio = \frac{1725}{400} = 4.3125$$

Given a 10 tooth driving sprocket, the number of teeth on the driven sprocket can be calculated as:

$$(4.3125)(10) = 43.125\ tooth\ sprocket$$

Since sprockets are obviously not available with fractional numbers of teeth and suppliers only sell certain sprocket sizes, the sprocket size was rounded up to 45 teeth.

Therefore, the new speed ratio can be calculated as:

$$45 = (speed\ ratio)(10)$$

$$speed\ ratio = 4.5$$

This leads to a rotational speed of 383.33 rpm for the rollers and results in a truss speed of 6.69 ft/sec or 7.341 km/h. This truss speed is still below the running speed of a human and is slower than our original speed, which in this case is ideal.

B.1.4.2 Roller Shaft Key Sizing

Given that the existing roller shaft size is 1 in., the standard key sizing taken from Mott's text is ¼ in. square. This corresponds to the key sizing pre-machined in the selected driven sprocket. The torque on the shaft was calculated as:

$$T = \frac{63000(P)}{n} = \frac{63000(1)}{383.33} = 164.35\ in \cdot lb$$

This torque results in the following force on the shaft:

$$F = \frac{T}{\frac{D}{2}} = \frac{164.35}{\frac{1}{2}} = 328.7\ lb$$

Since the sprocket material is alloy steel, it is desired to have a key that will fail before the sprocket fails. AISI 1020 steel was chosen for a key material with a yield strength of 51000 psi. Therefore, the smallest required key size can be calculated as:

$$L = \frac{4TN}{DWS_y} = \frac{4(328.7)(3)}{(1)\left(\frac{1}{4}\right)(51000)} = 0.3094\ in.$$

This was calculated using a design factor of N = 3 for standard machine applications as recommended in Mott's Textbook [4].

The resulting key length of 0.3094 inches is much smaller than the sprocket width of 1.25 in. Therefore, length of the key will increased to 1 in. for further reliability.

The final key dimensions are:

Length= 1 in.

Width= ¼ in.

Depth= 1/8 in.

B.1.4.3 Motor Mount

In order to maintain simplicity in the final design, A36 steel was selected as the material for the motor mount. The properties of this material are as follows:

Yield strength σ_y = 36 ksi

Shear strength τ = 21 ksi

Endurance strength $s_n \cong$ 23 ksi

To demonstrate the motor mount's ability to handle the applied motor torque and weight loads, a simple finite element analysis was performed using Cosmos in SolidWorks. For the test, 200 lbs was applied on the upper mounting surface of the motor mount. This weight was selected as a good estimate of the static weight of the motor as well as the torsional forces applied during rotation. The following displacements and stress distribution resulted:

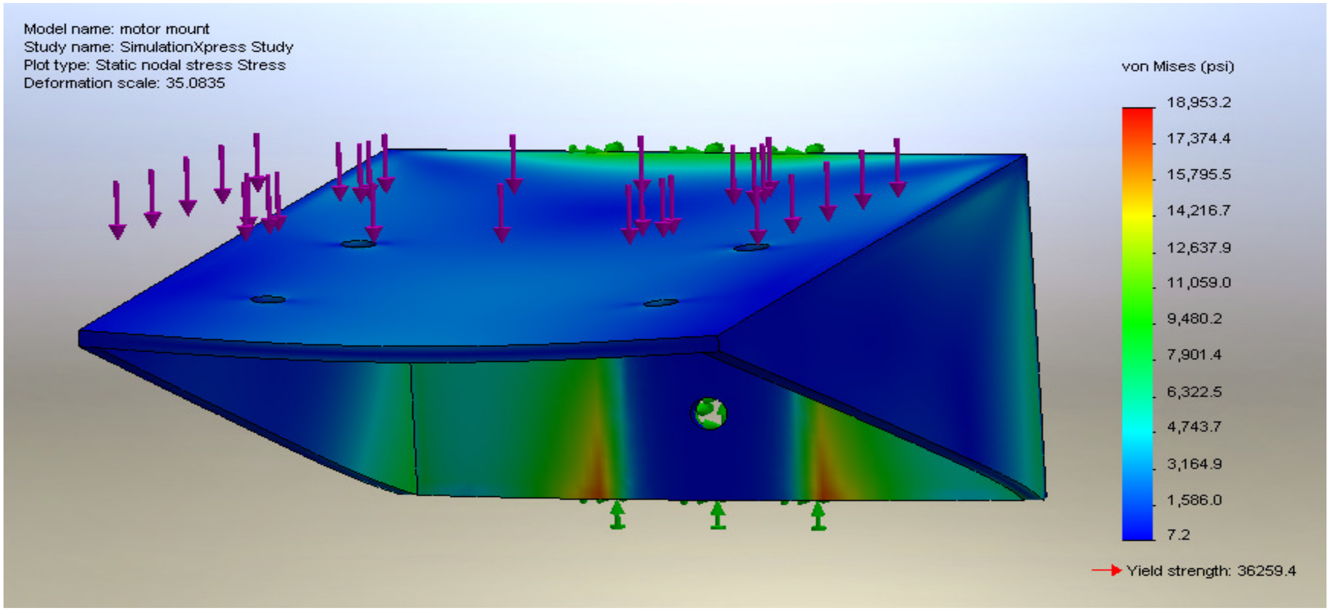


Figure 59: FEA von Mises (psi) Roller Platform - Motor Mount (Deformation scale 35.0835)

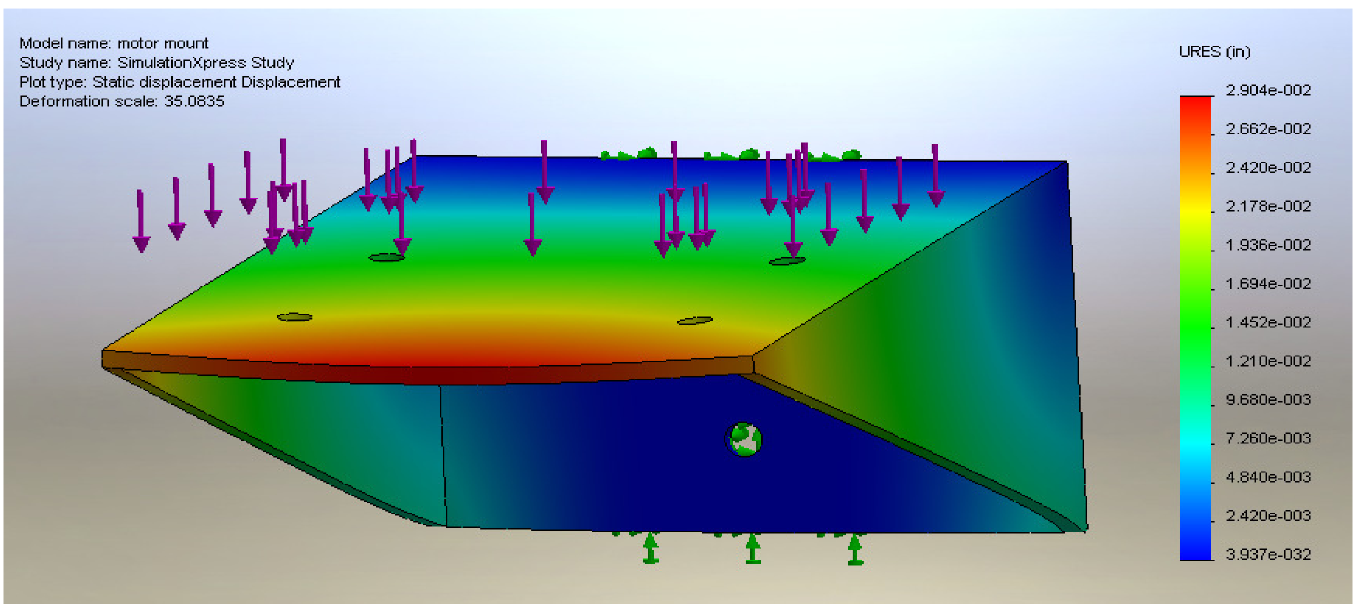


Figure 60: FEA Relative Displacement (in) Roller Platform - Motor Mount (Deformation scale 35.0835)

From the FEA test, we can see that the maximum Von Mises stress in the motor mount occurs at the portion of the mount which connects to the vertical roller stand. This stress was focused around the area where the stand no longer contacts the mount. However, this stress was found to be well below the acceptable yield strength of the A36 material chosen for the part and yielded a factor of safety of approximately 2.

The analysis of the displacement of the part yielded similarly acceptable results with a maximum displacement of approximately 0.029 in. This maximum displacement occurs at the far edge of the side of the motor mount facing the motor. This maximum displacement was found to be acceptable and part failure should not occur.

Appendix C: Technical Drawings

The following appendix is comprised of the technical drawings of the major components for our design. The technical drawings were completed using the SolidWorks program and examples from the CAD Drawings library available on McMaster-Carr [2].

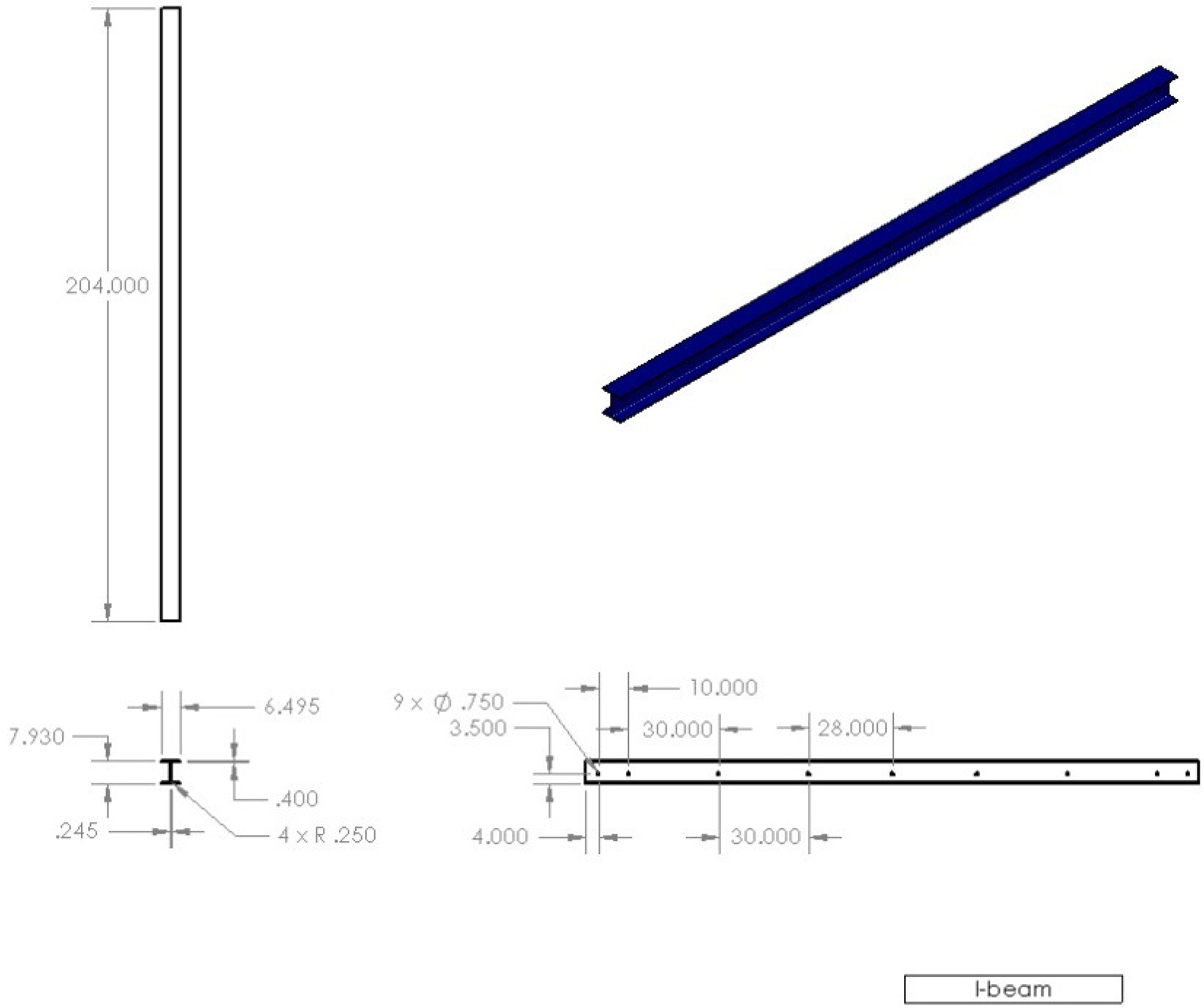


Figure 61: I-beam

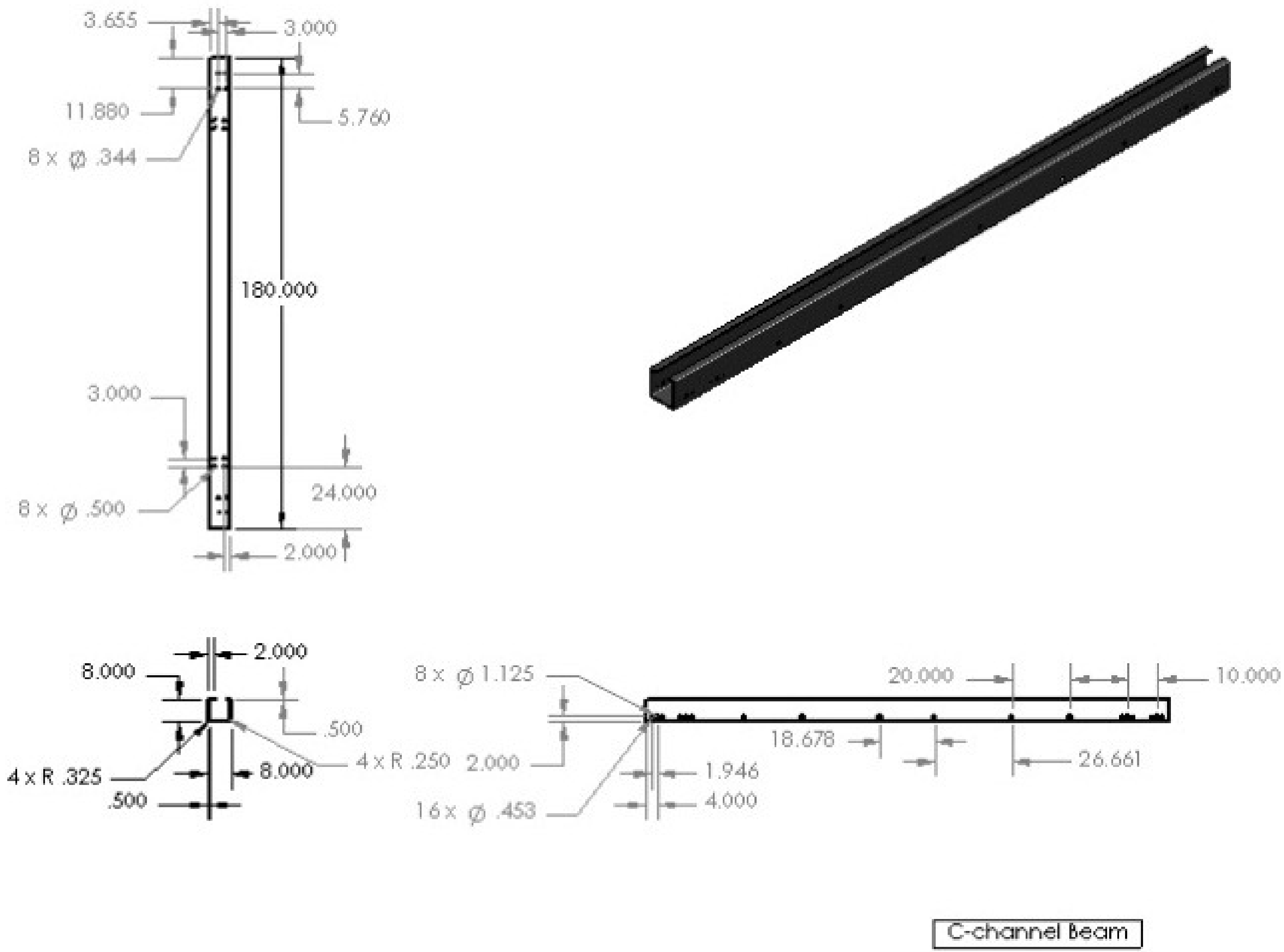


Figure 62: C-channel beam

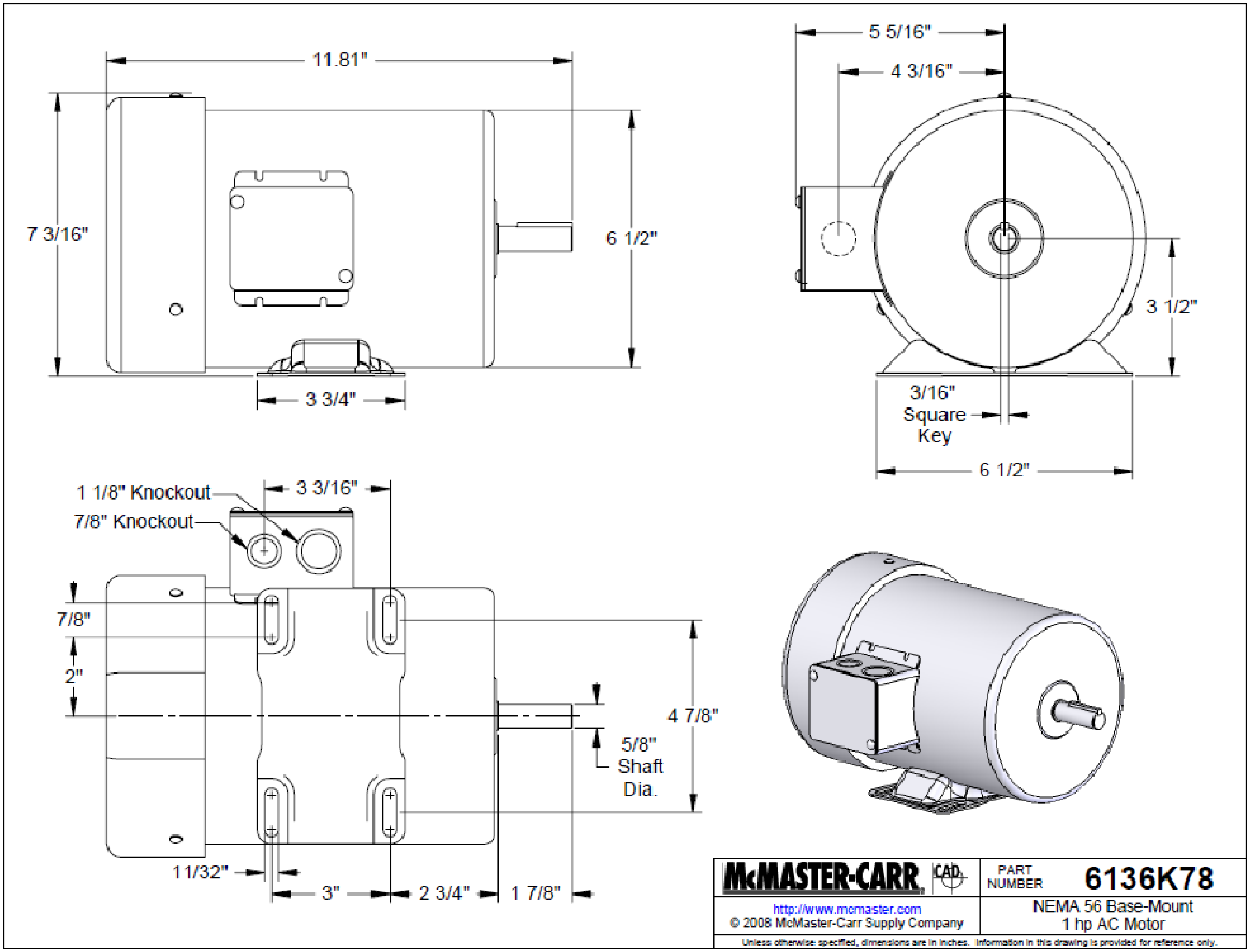


Figure 63: Horizontal drive motor [2]

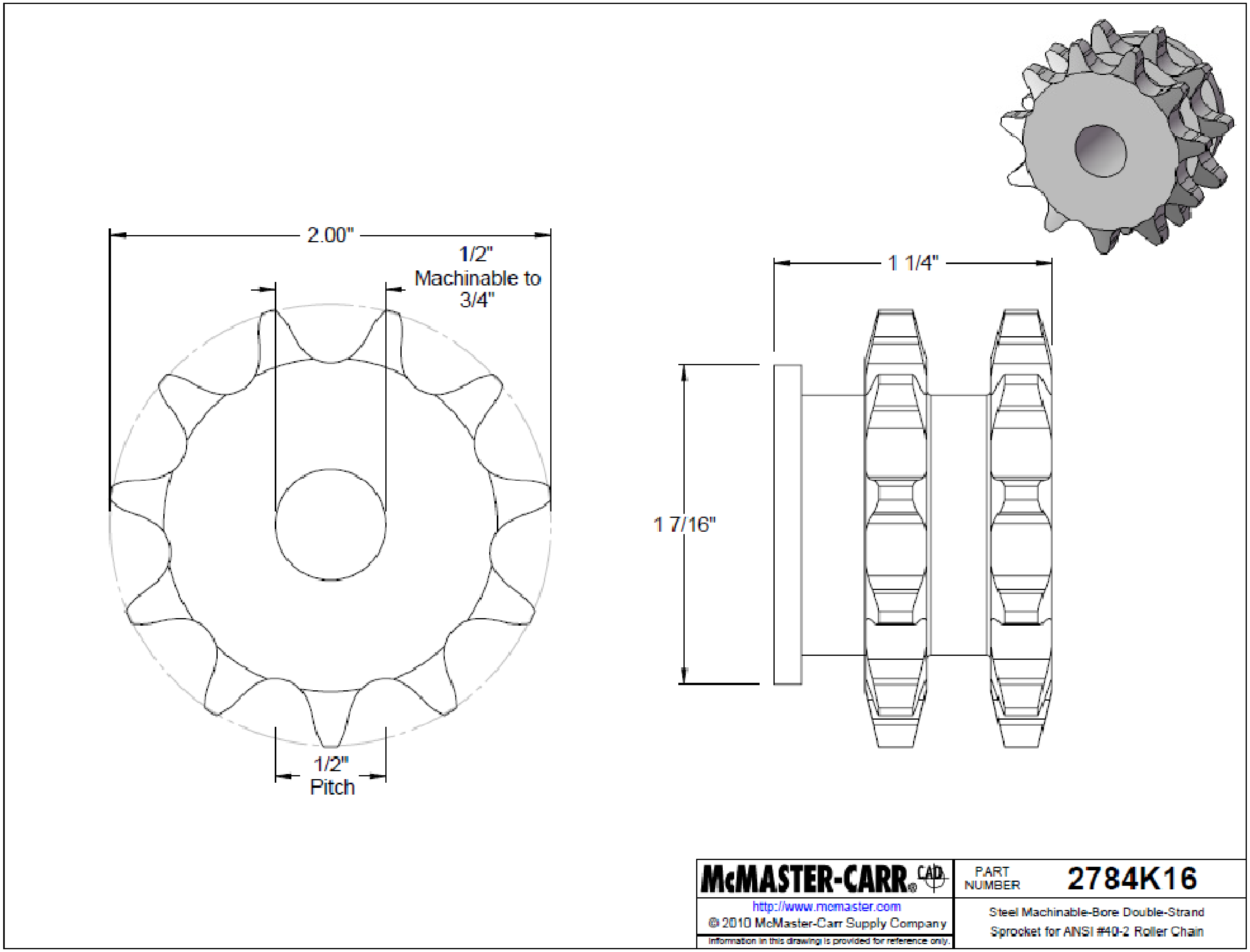


Figure 64: Roller drive sprocket [2]

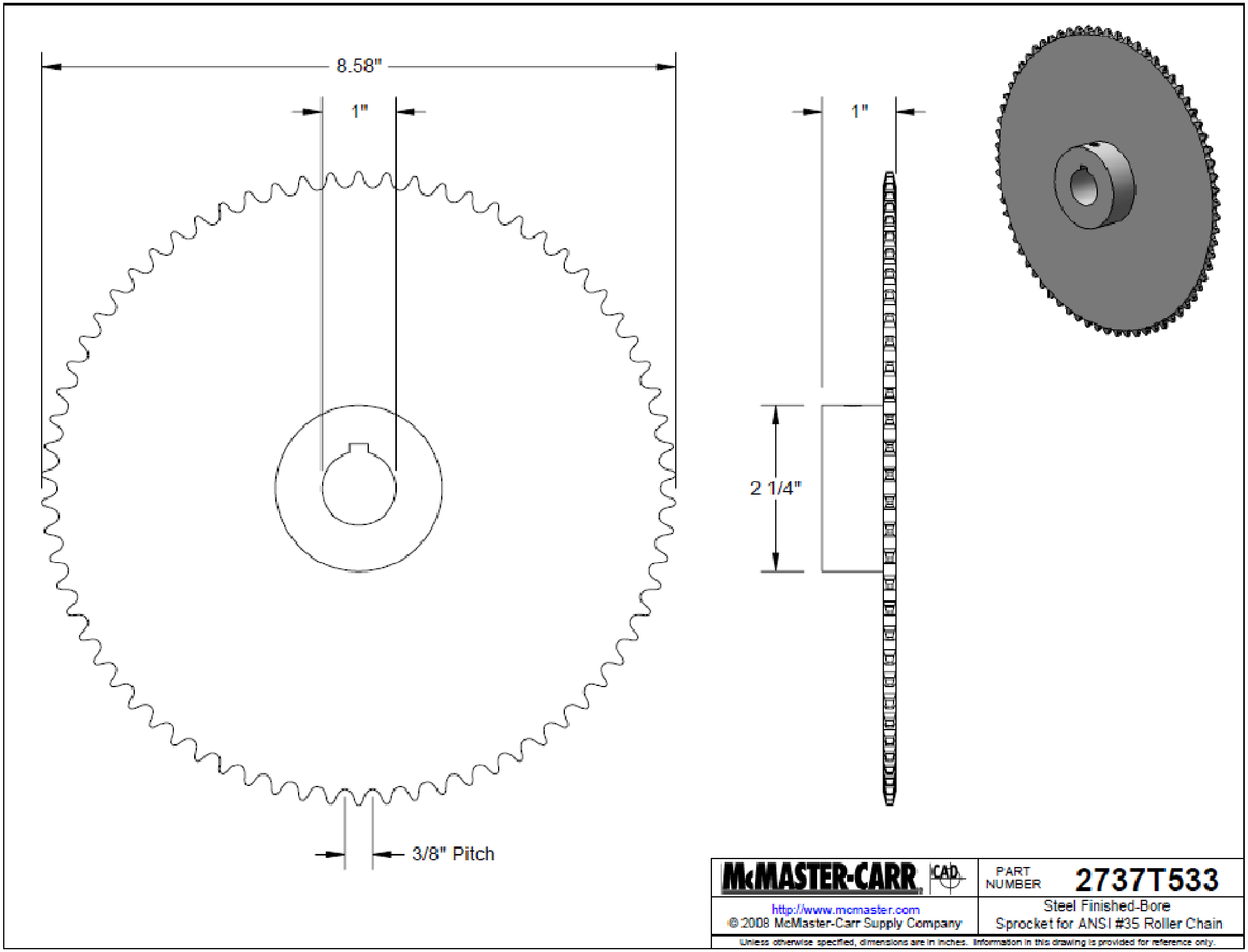


Figure 65: Roller driven sprocket [2]

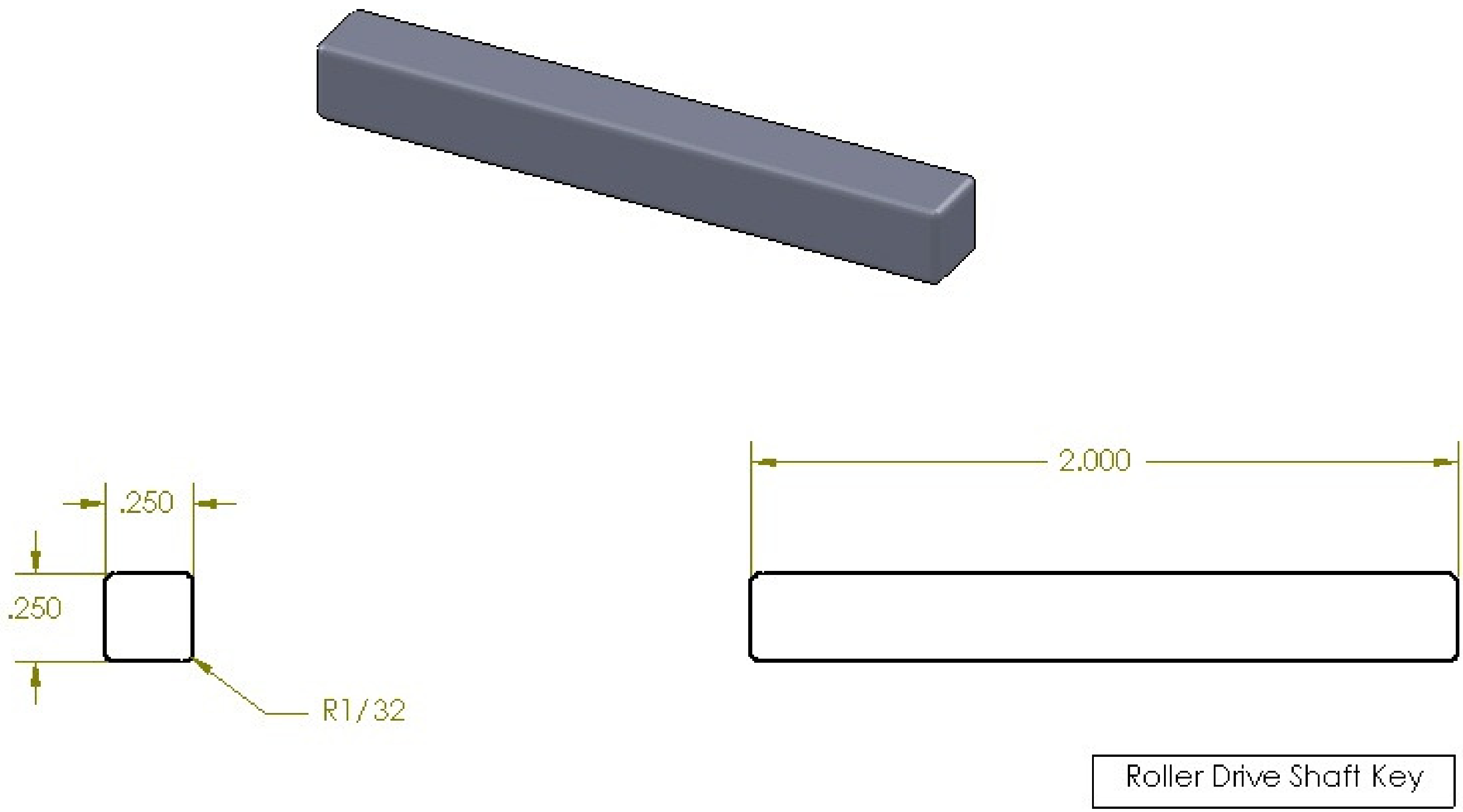


Figure 66: Drive shaft key

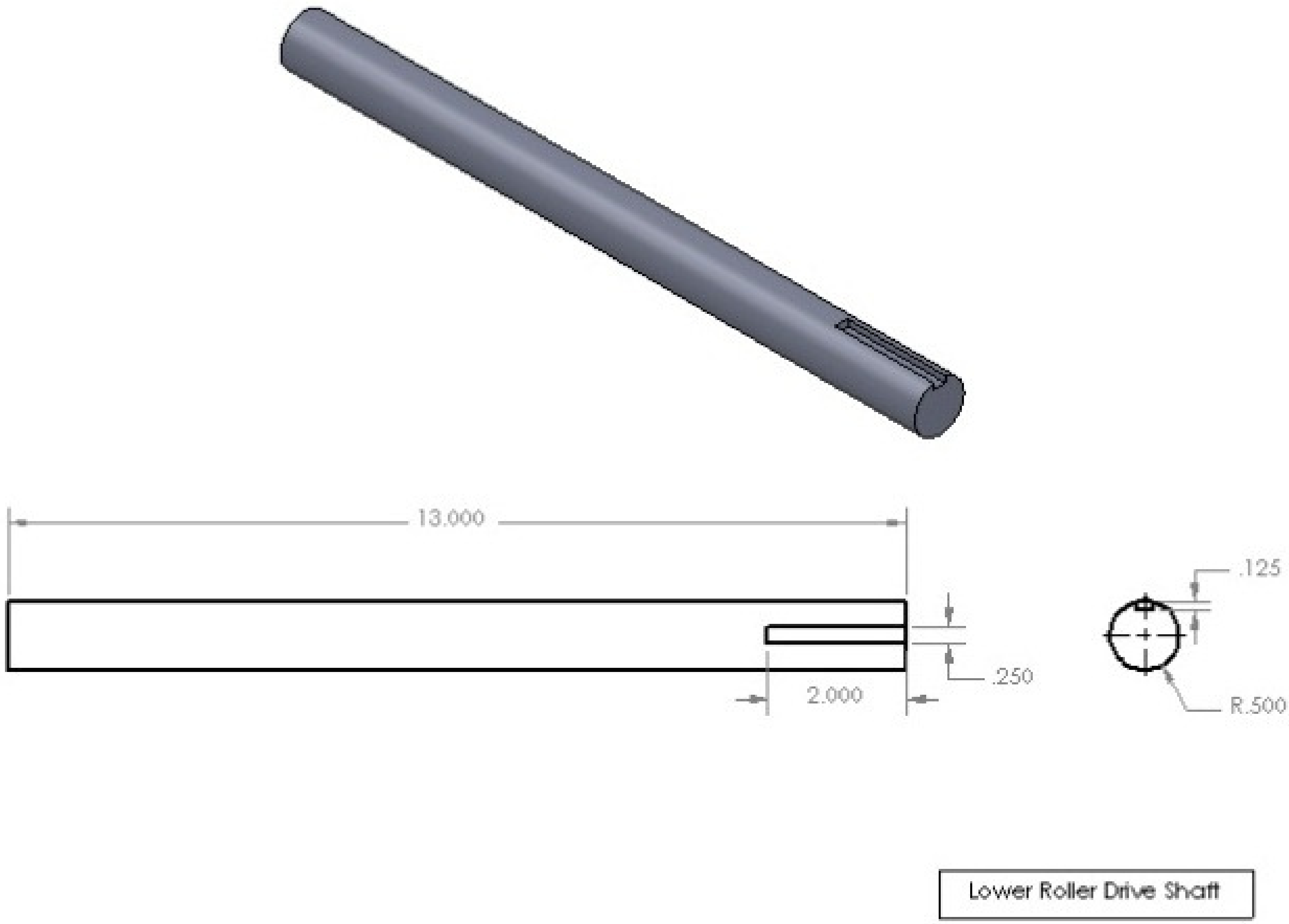


Figure 67: Lower roller drive shaft

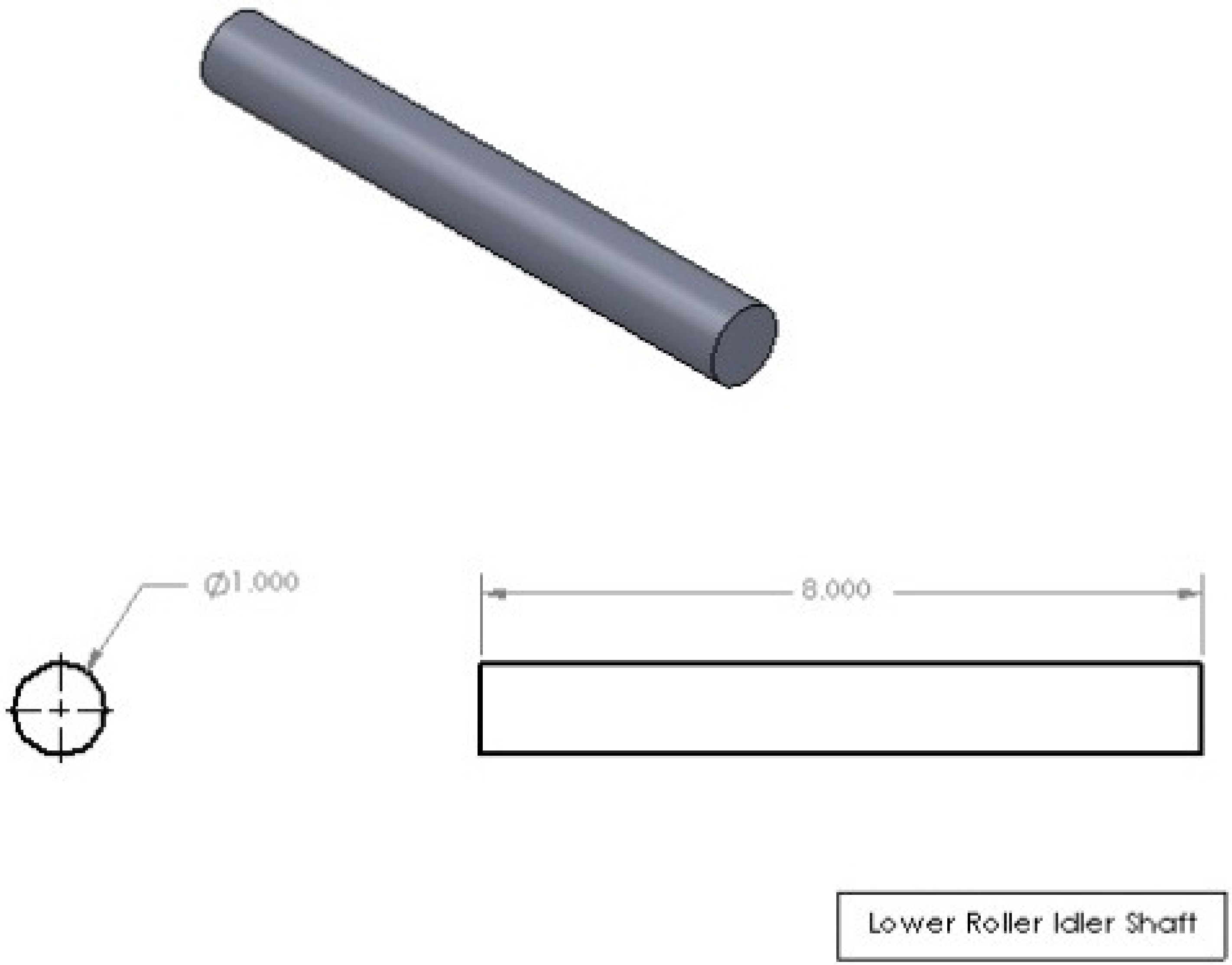


Figure 68: Lower roller idler shaft

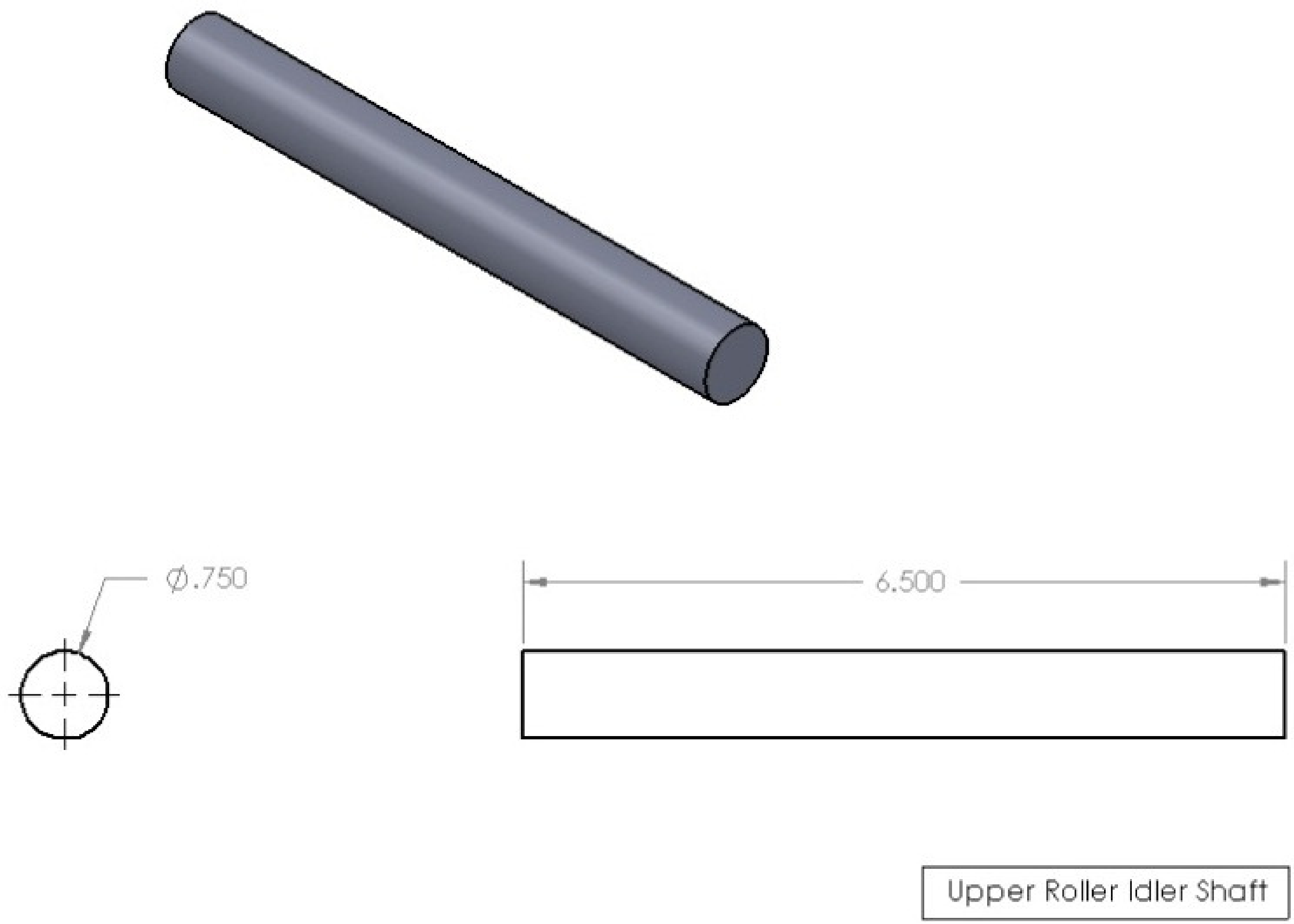


Figure 69: Upper roller idler shaft

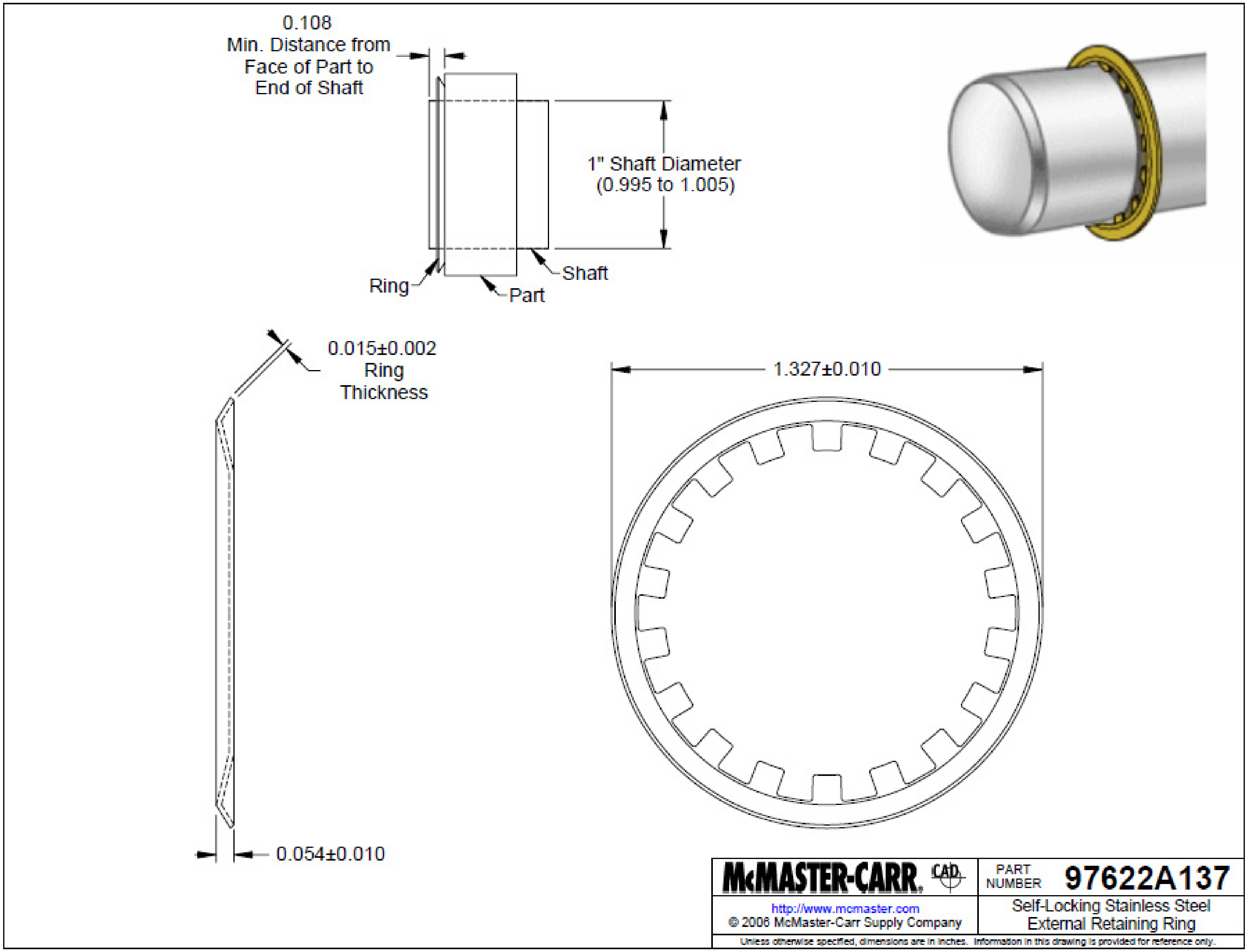


Figure 70: Lower wheel retaining ring [2]

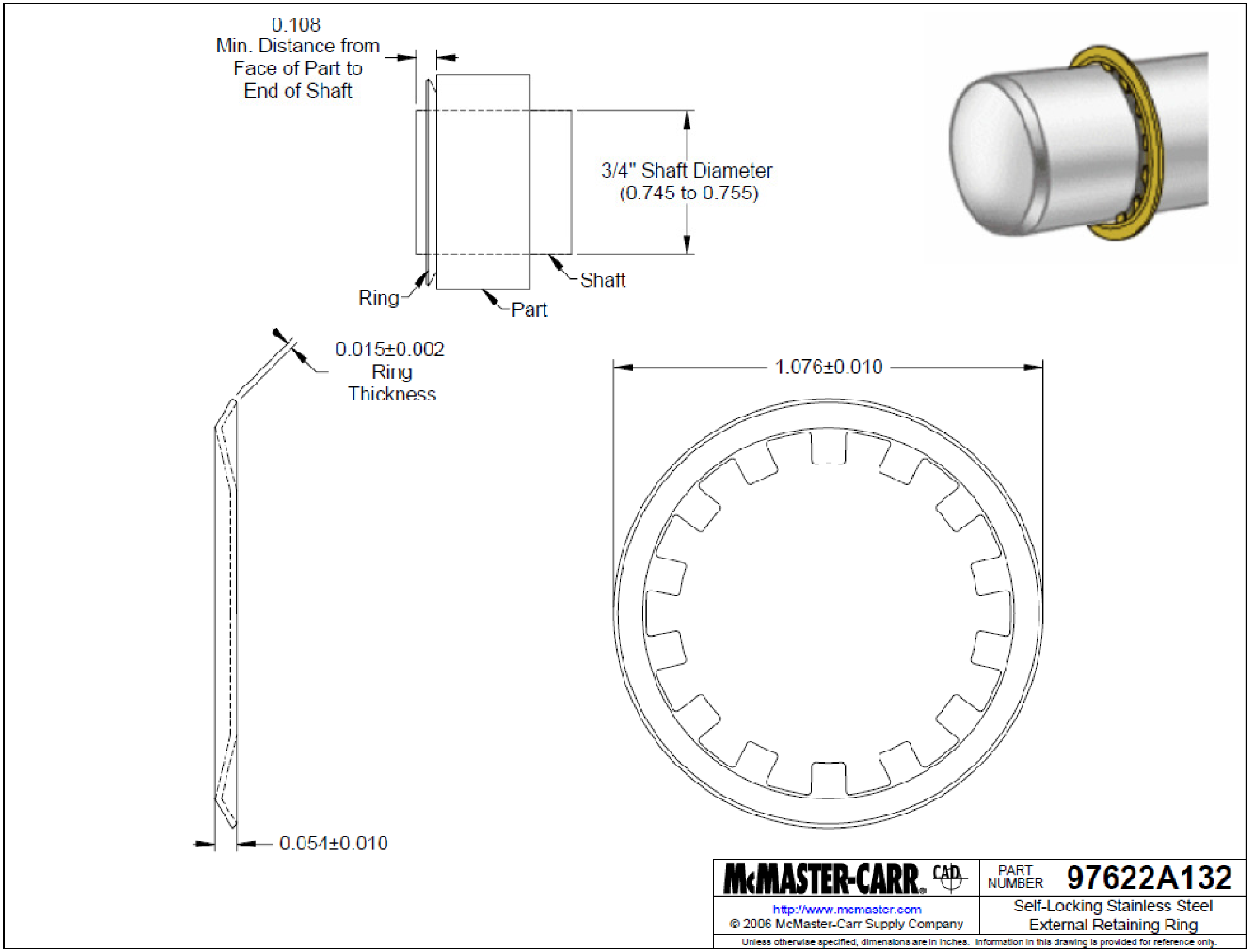


Figure 71: Upper wheel retaining ring [2]

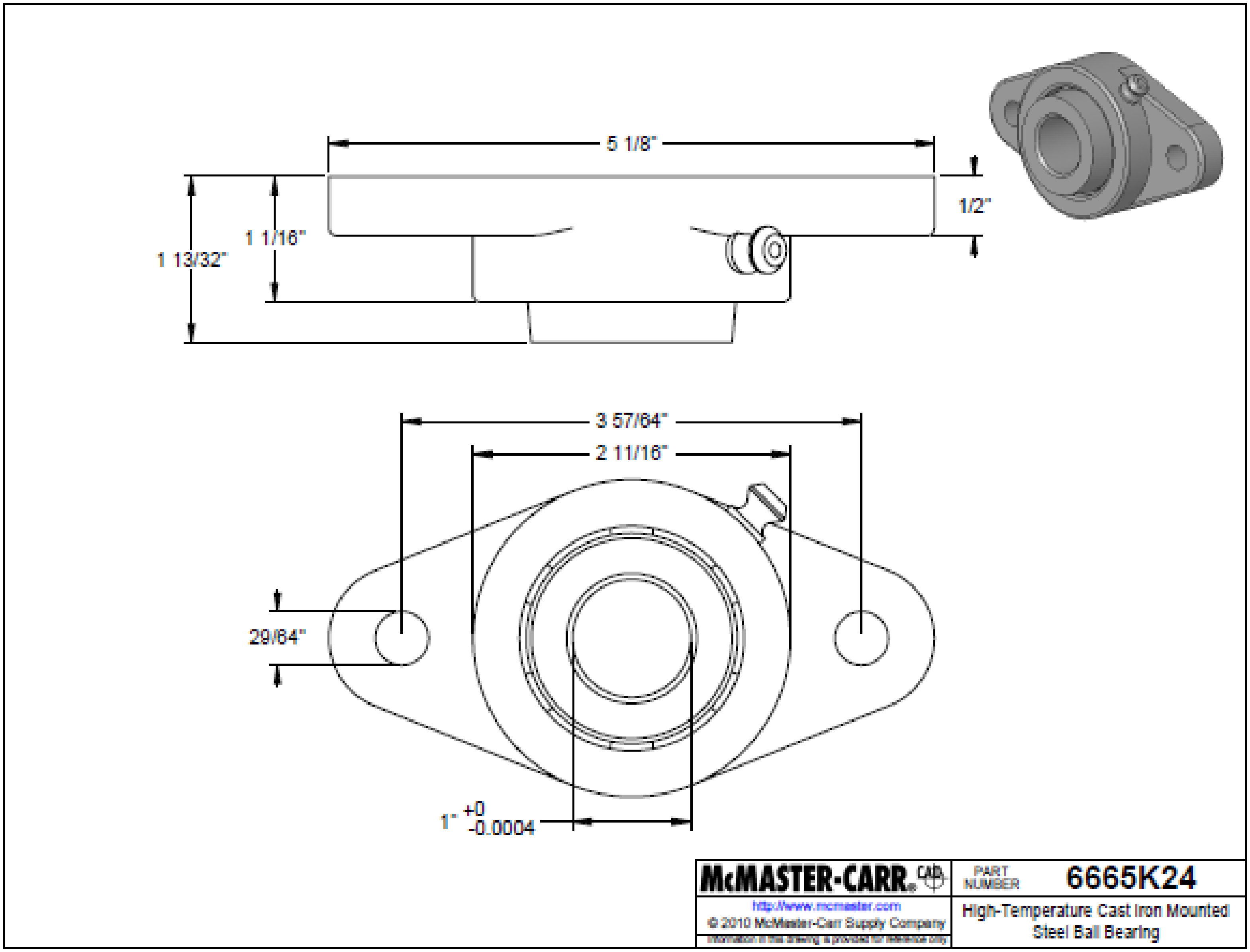


Figure 72: Drive roller bearing [2]

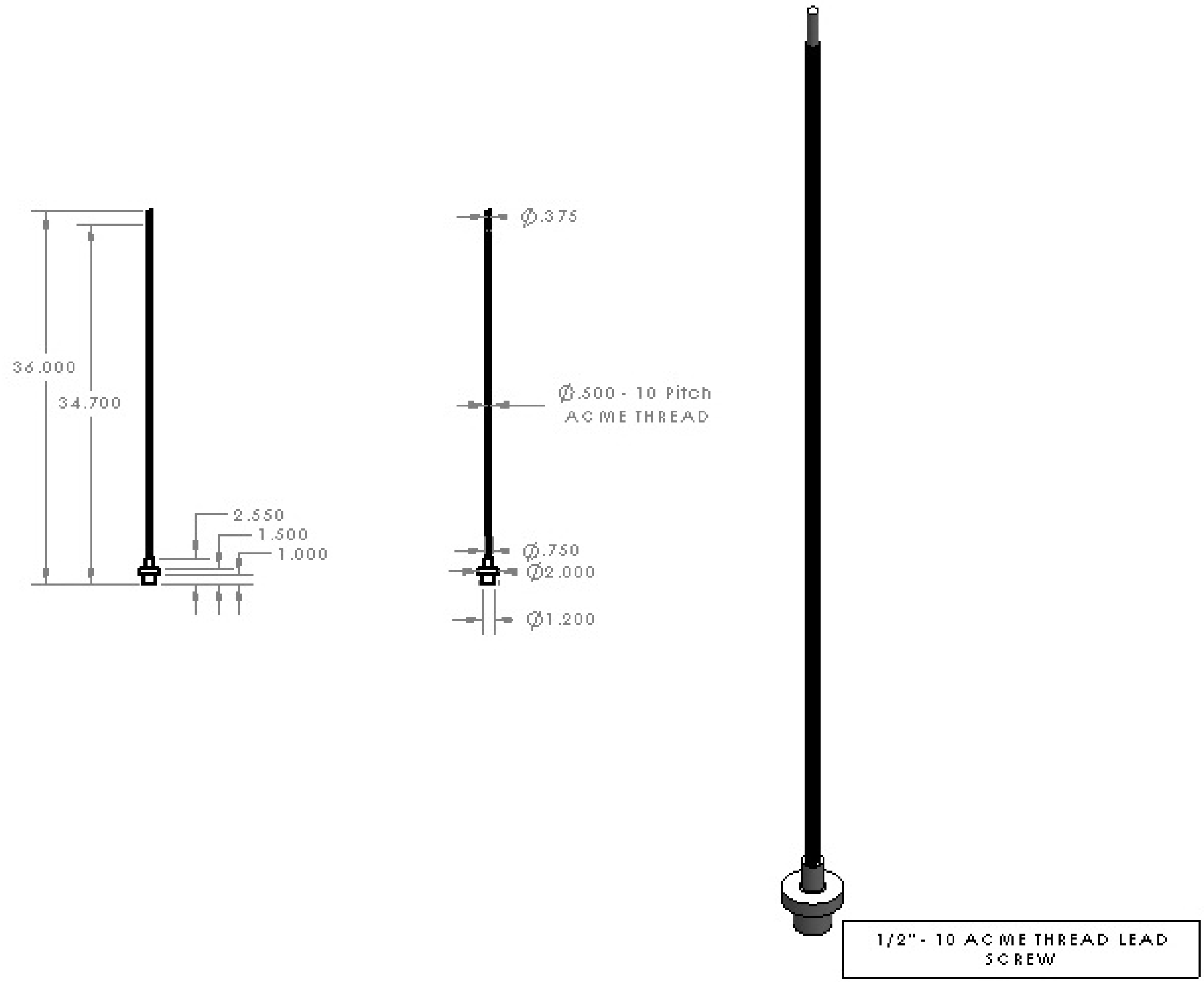


Figure 73: Lead screw

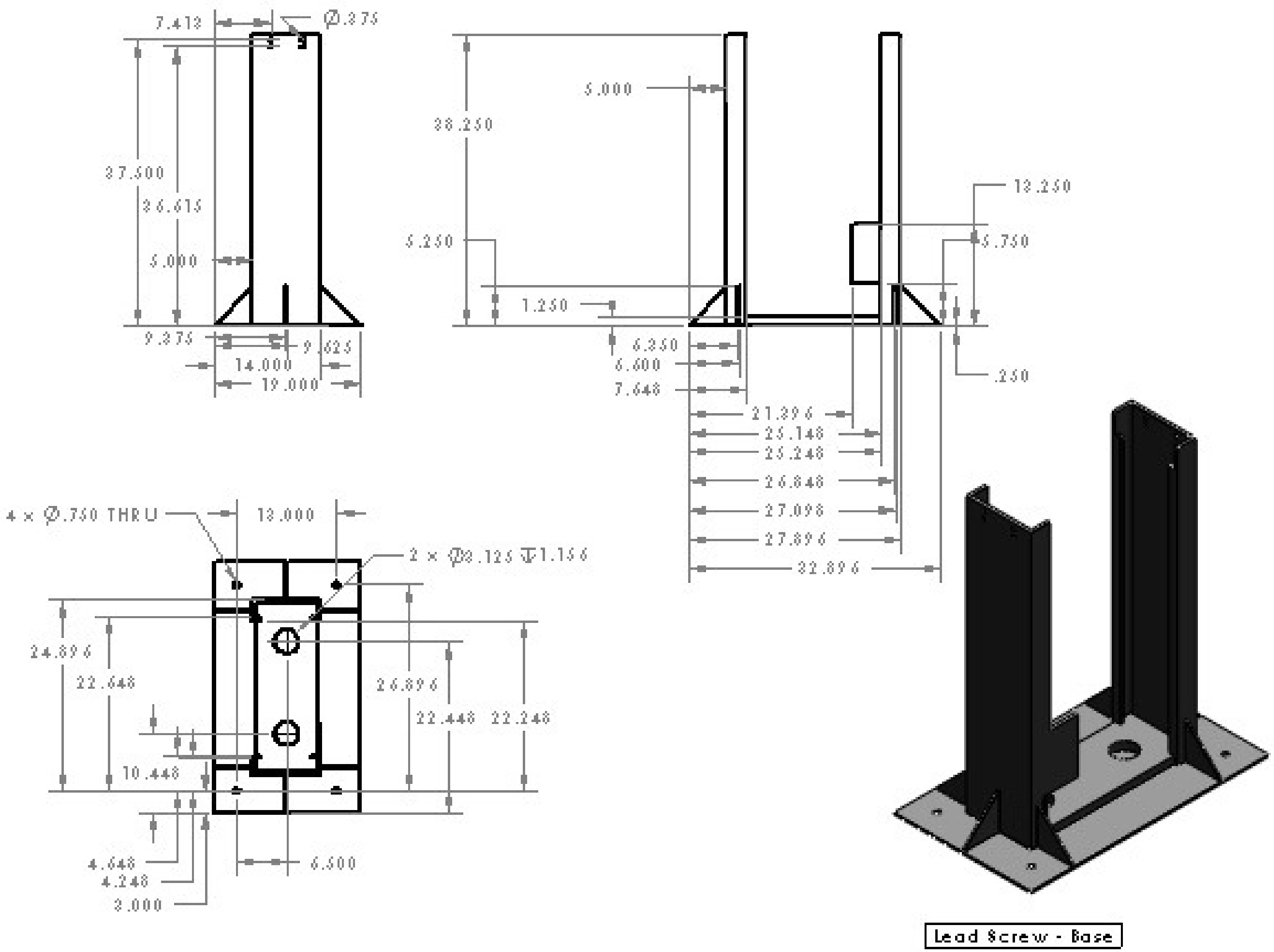


Figure 74: Lead screw - base

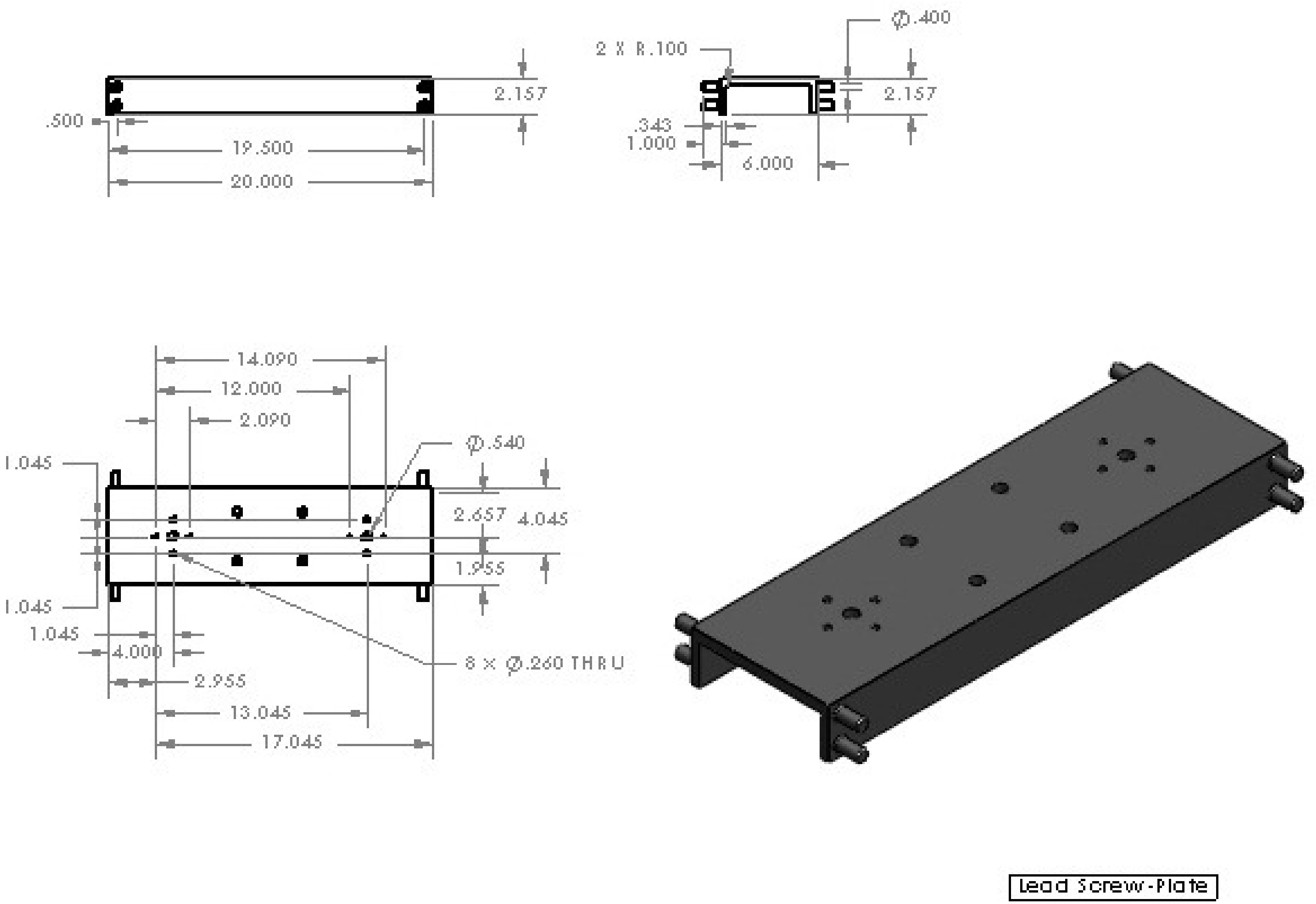


Figure 75: Lead screw - plate

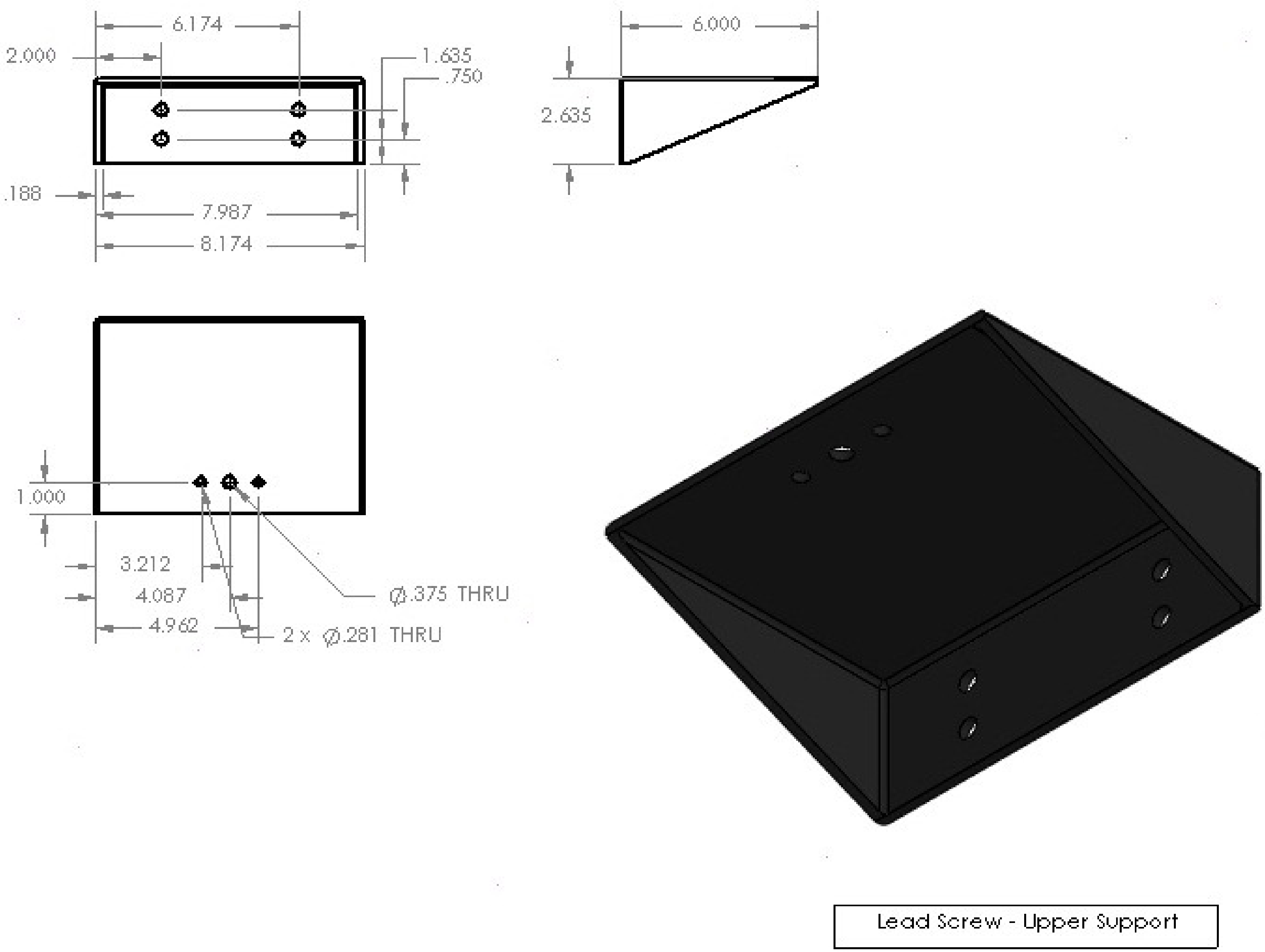


Figure 76: Lead screw - upper support

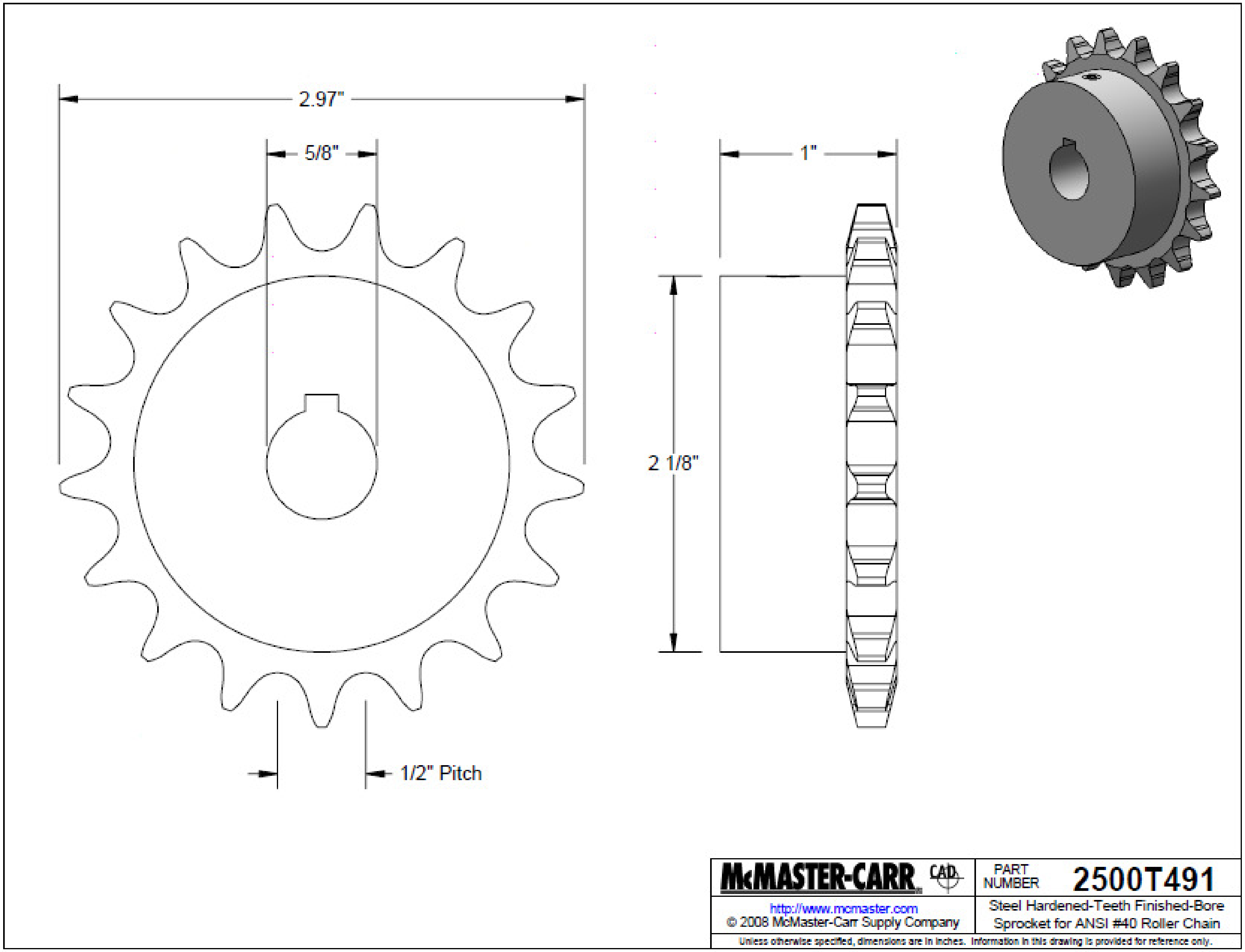


Figure 77: Lead screw - motor sprocket [2]

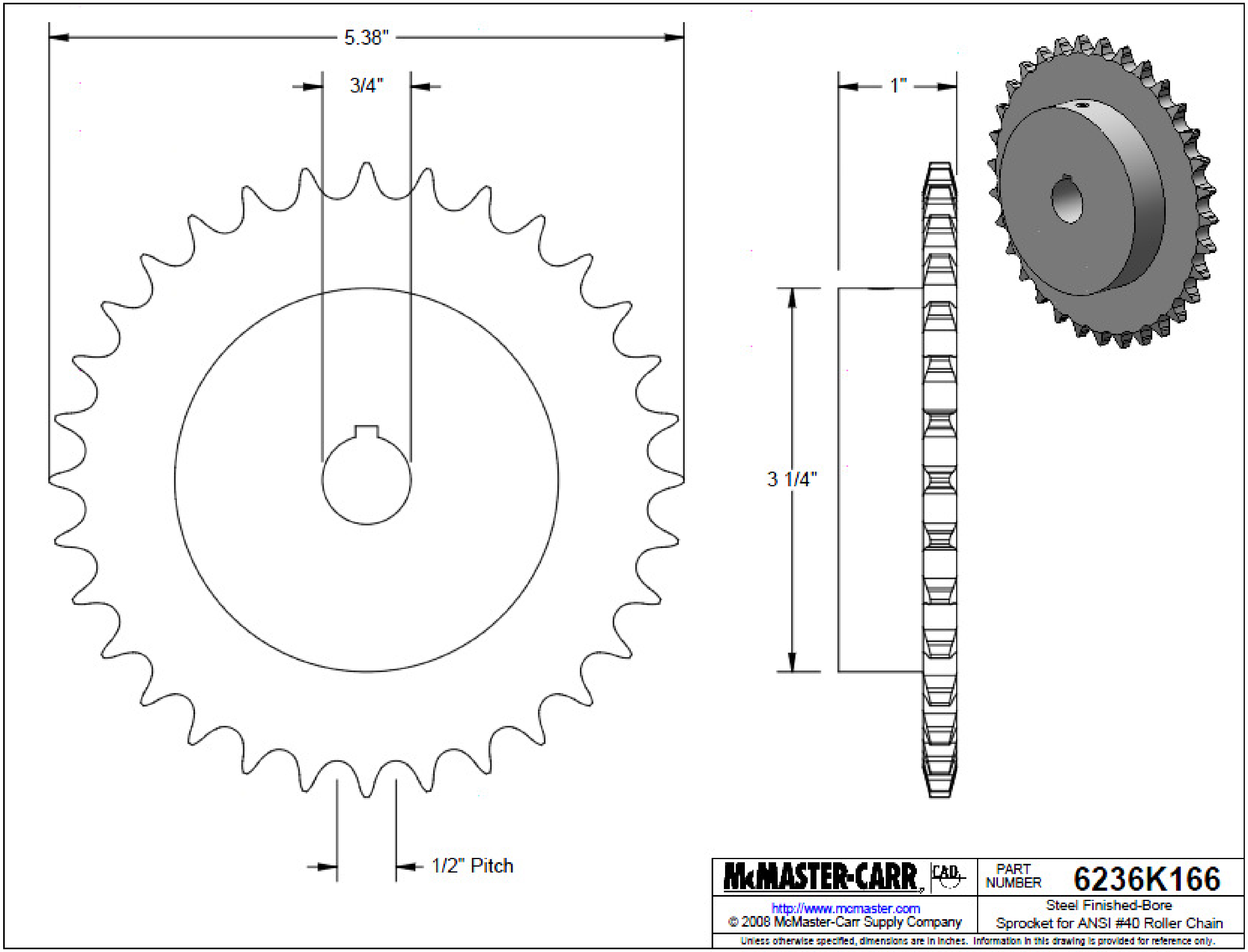


Figure 78: Lead screw - sprocket [2]

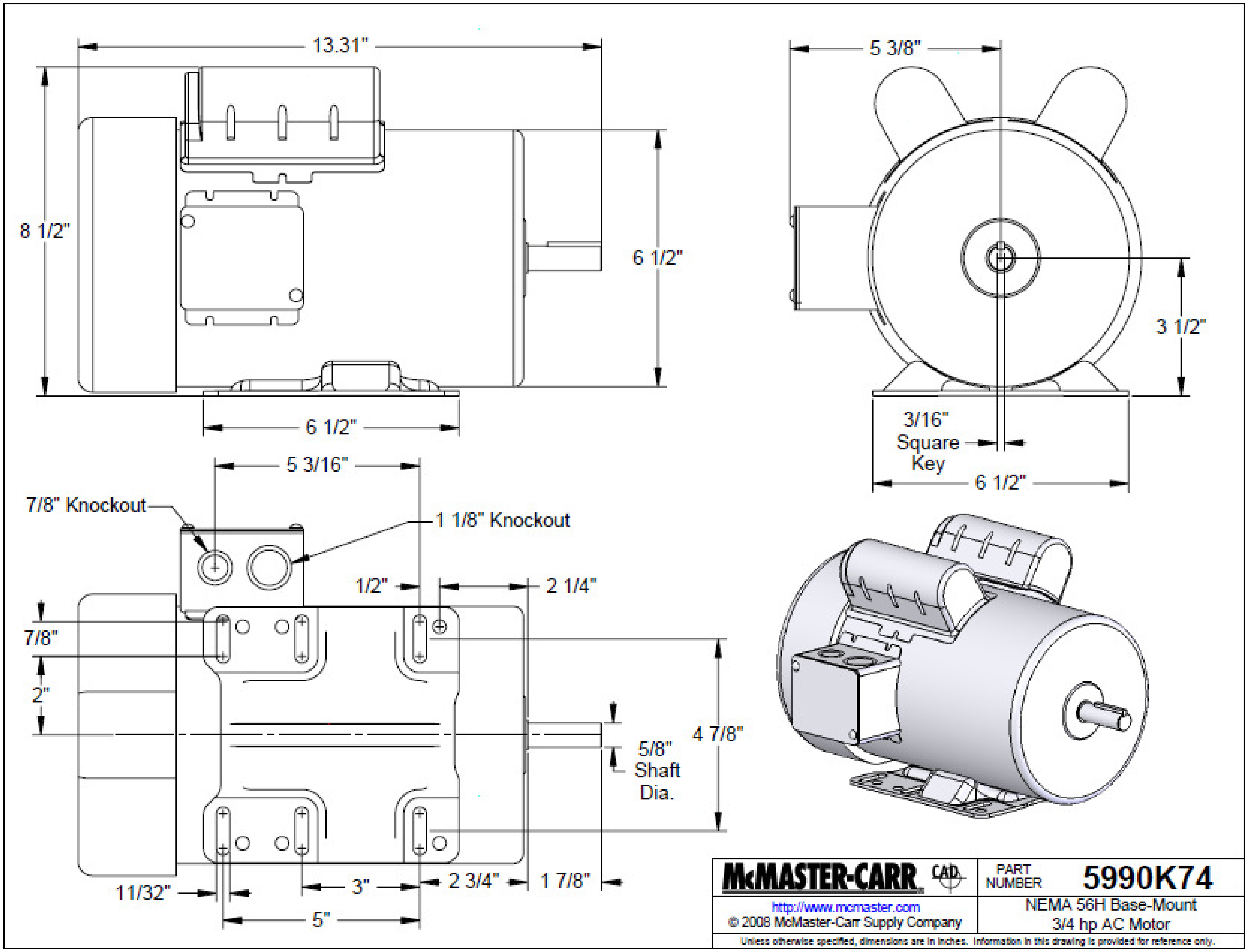


Figure 79: Lead screw - drive motor [2]

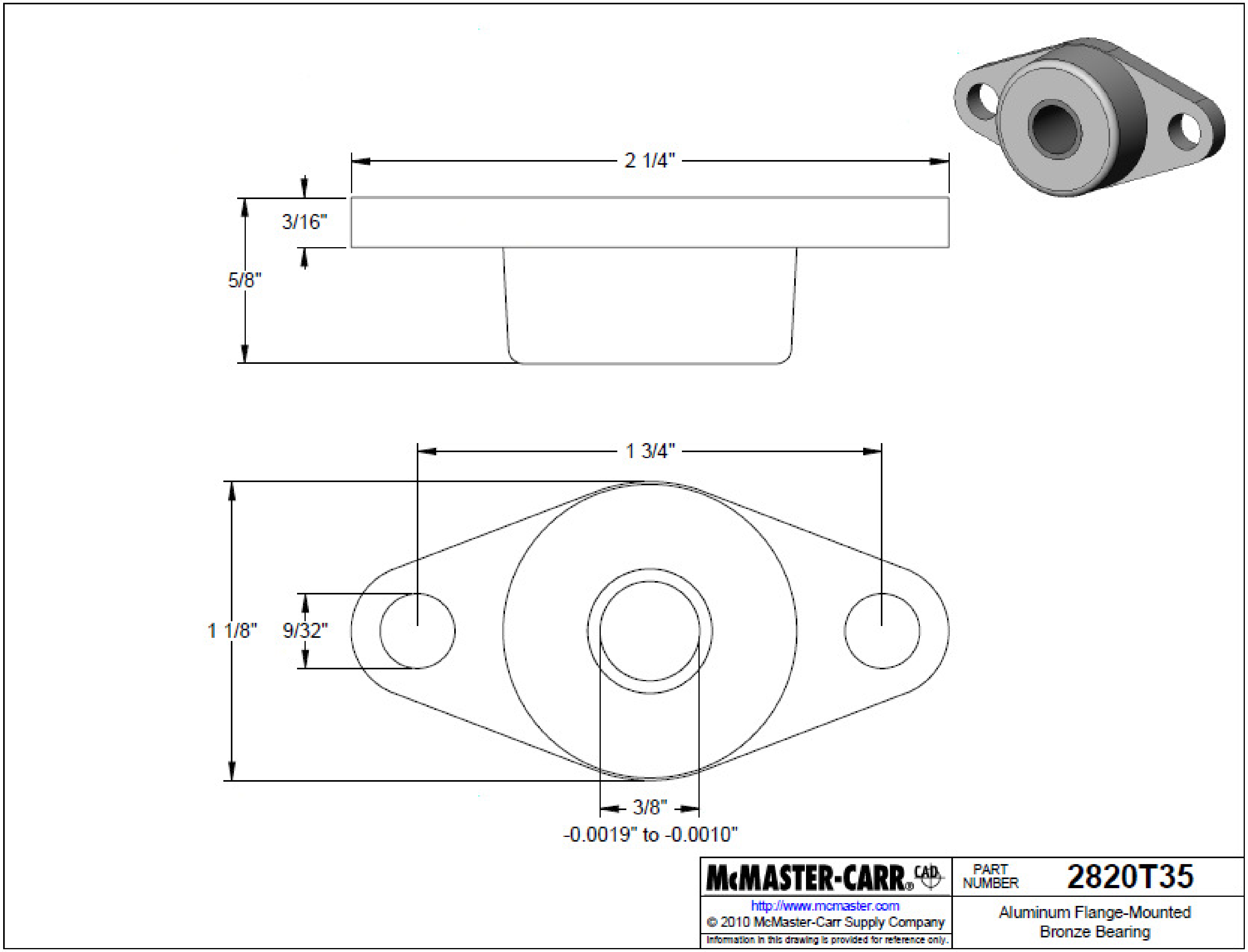


Figure 80: Lead screw - upper bearing [2]

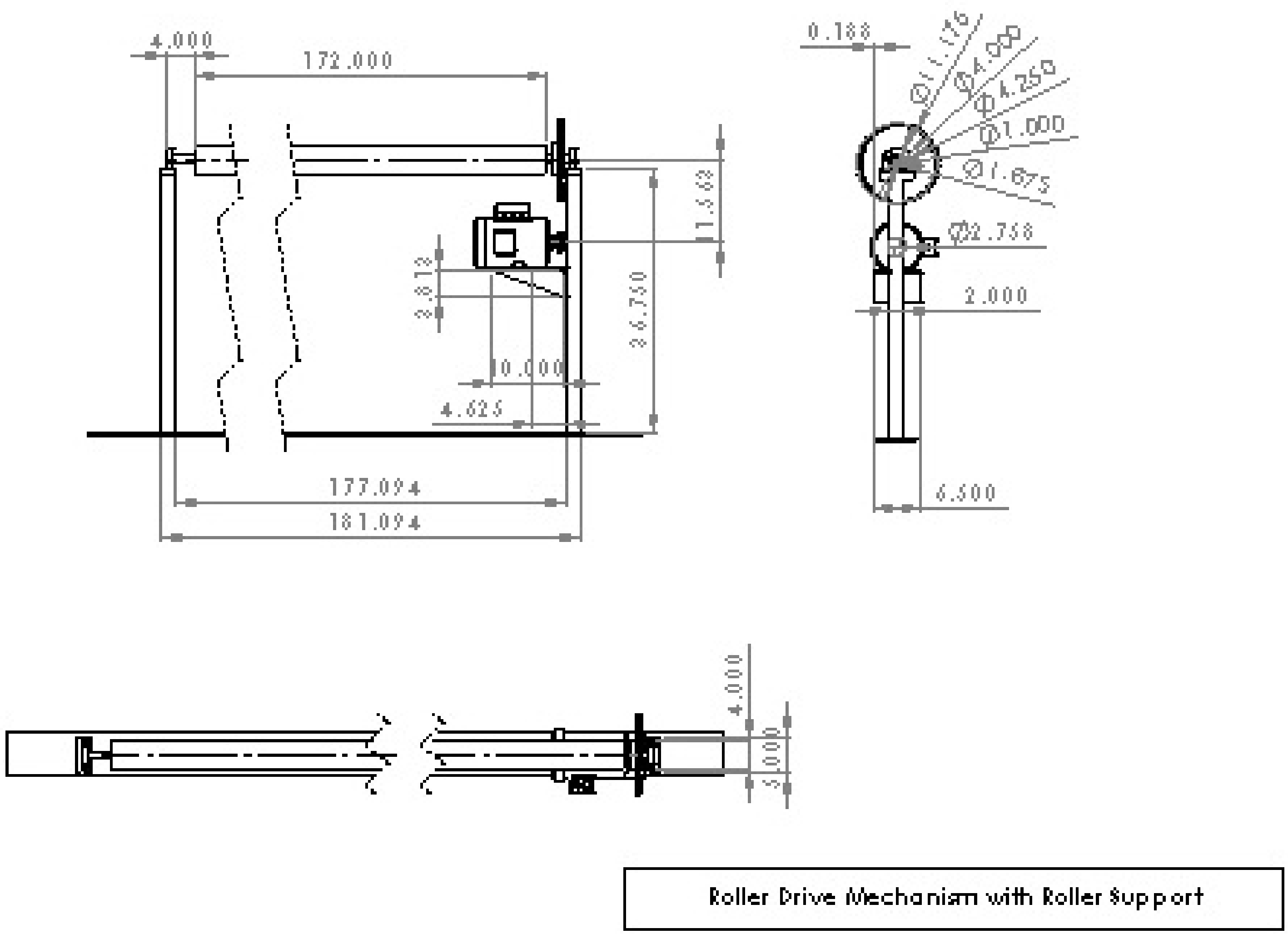


Figure 81: Roller drive mechanism with main roller support

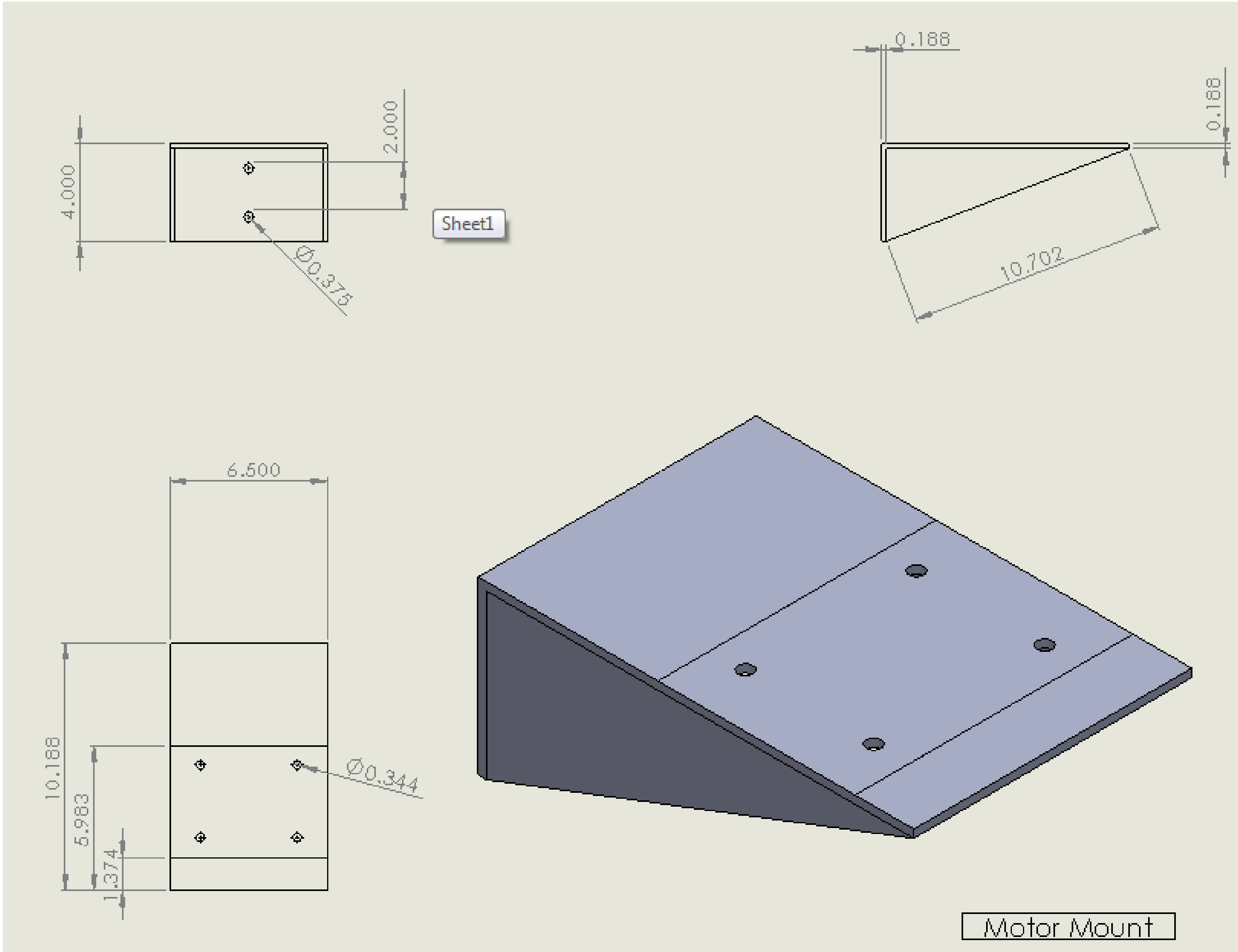


Figure 82: Motor mount

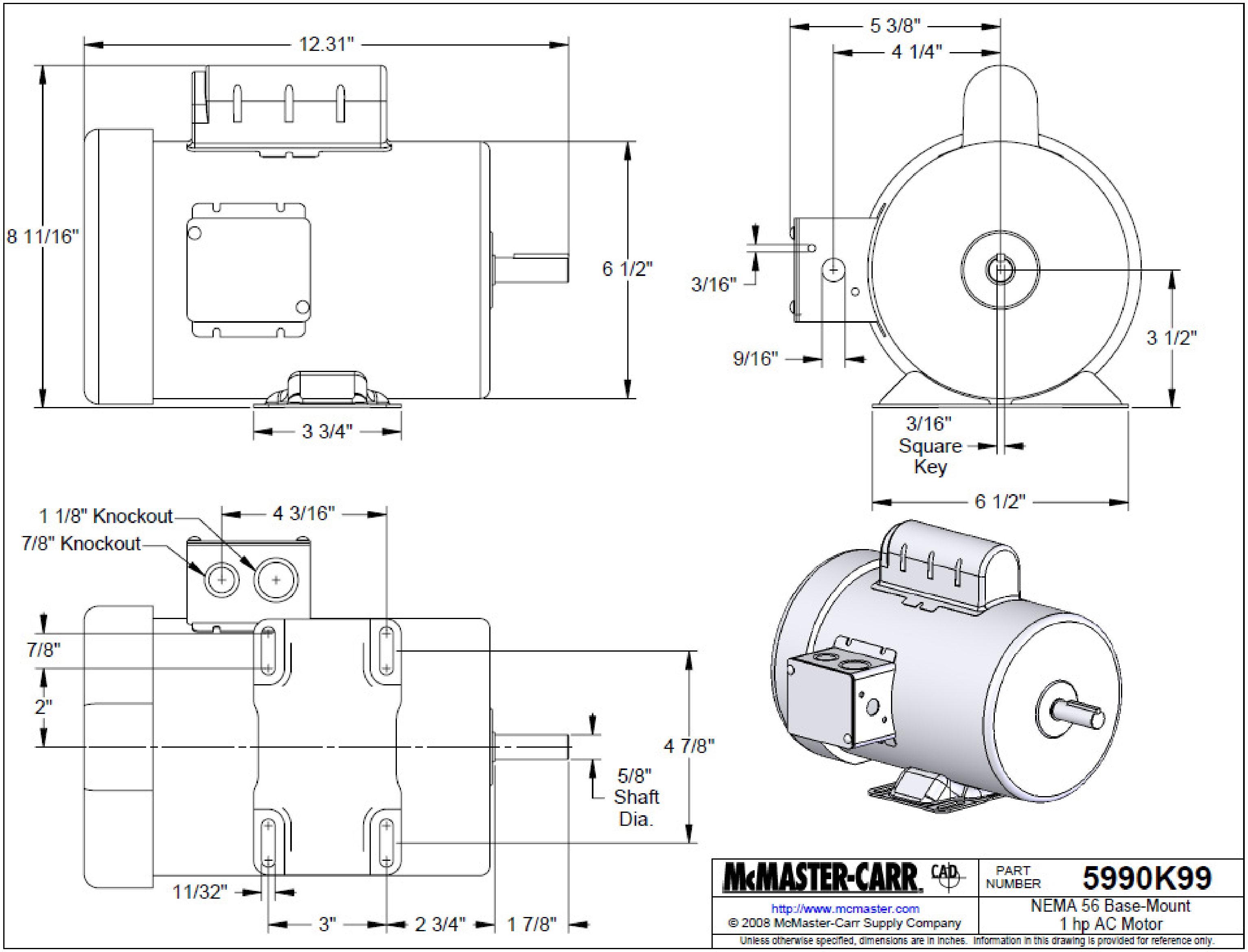


Figure 83: Roller drive motor [2]

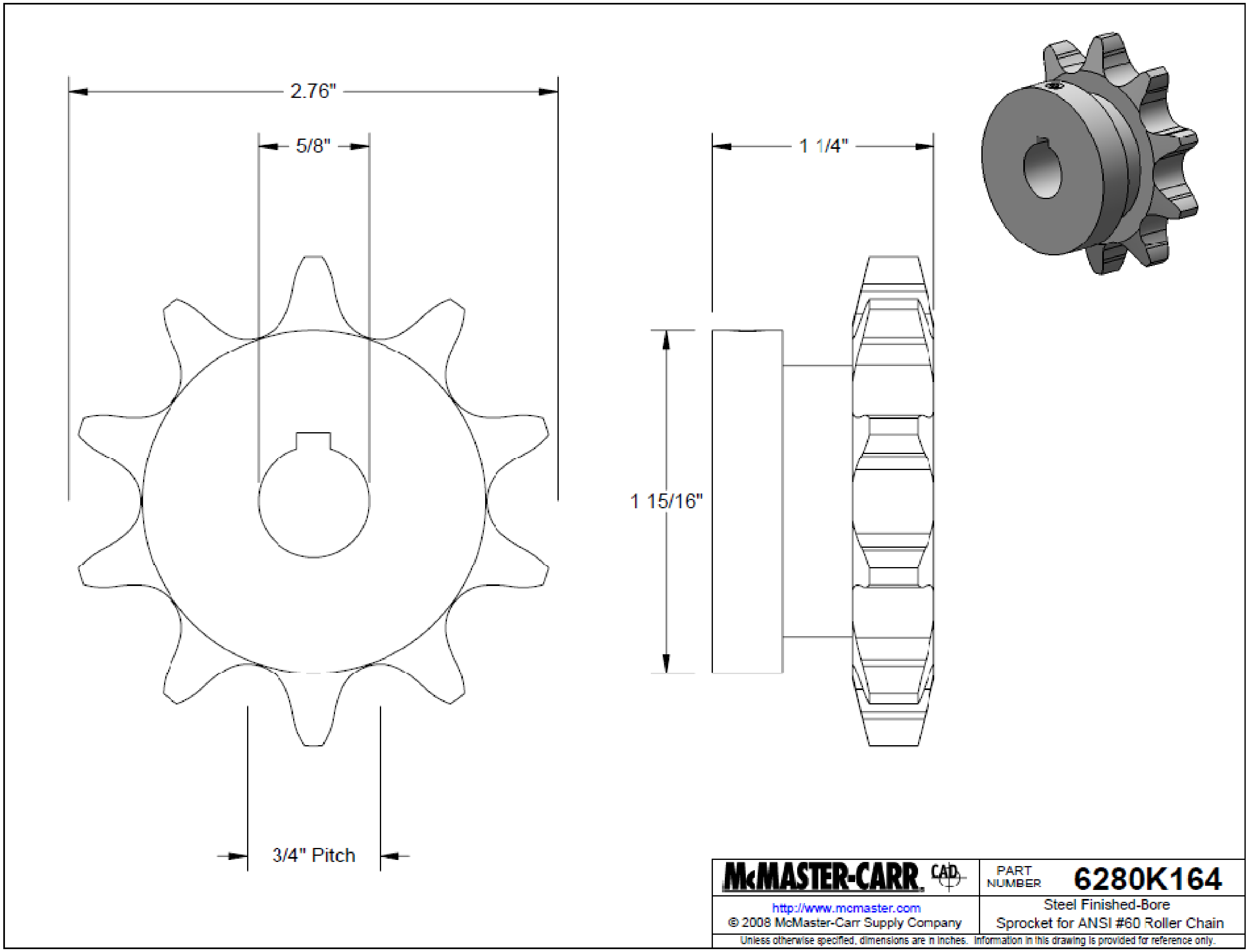


Figure 84- Roller drive mechanism- driving sprocket [2]

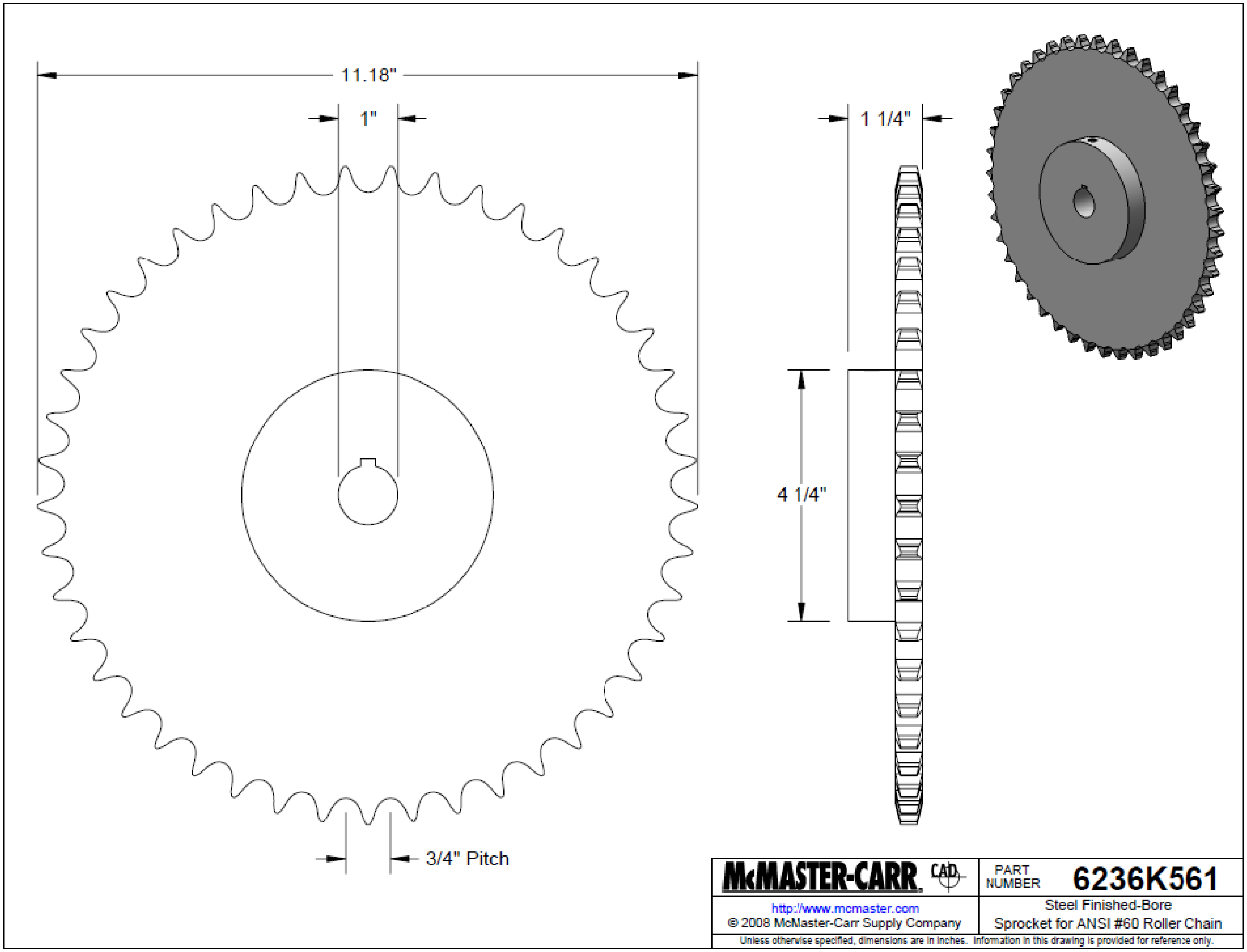


Figure 85- Roller drive mechanism- driven sprocket [2]

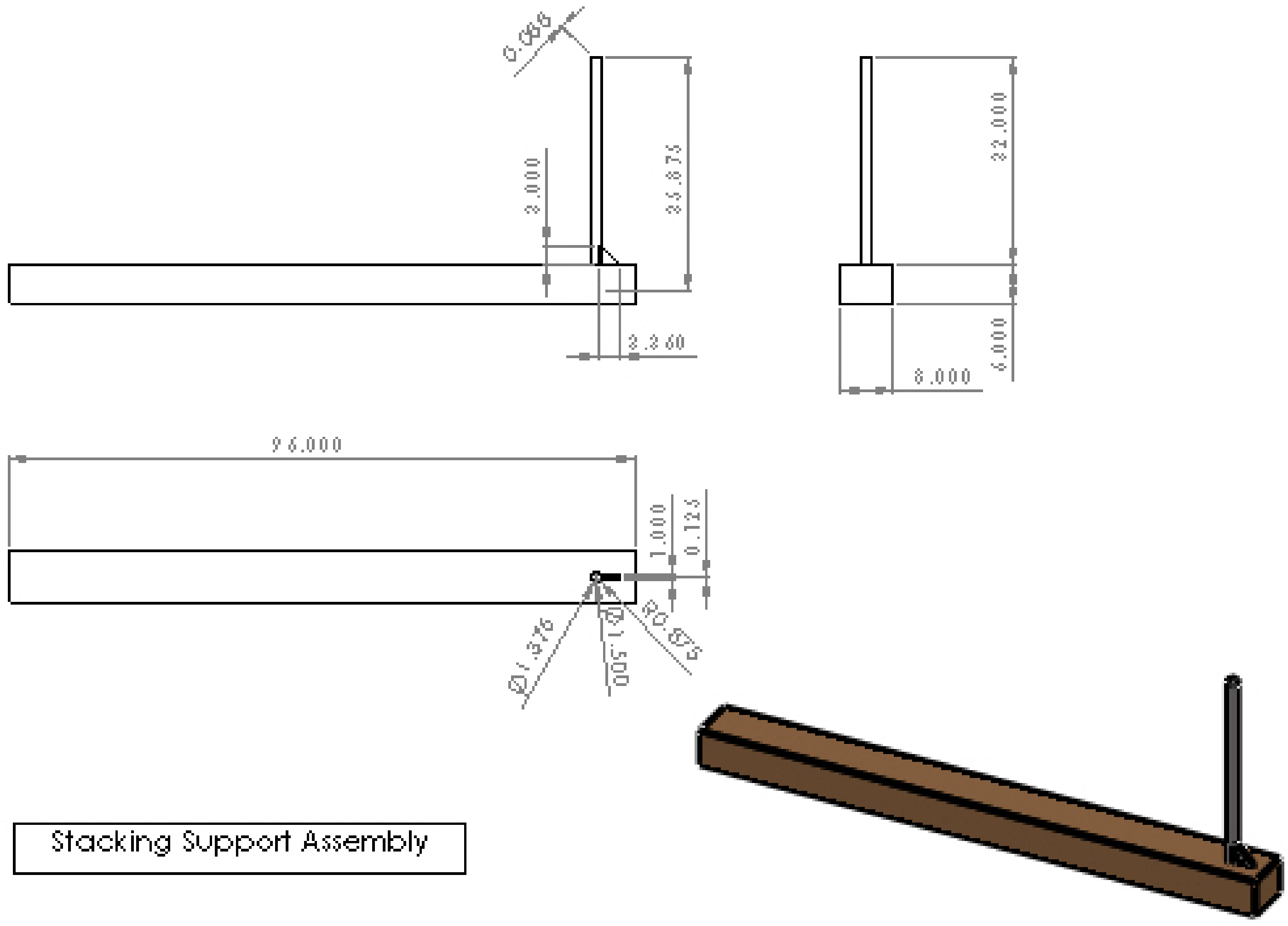


Figure 86- Stacking support assembly

Appendix D: Discussion of Assembly and Manufacturing

The following sections describe the basic assembly and manufacturing procedures required to build the truss stacking system. Grade 5 fasteners are used in all mounting locations where large stresses and/or torsional loading will be experienced. These locations include all motor mounts and motors. Grade 8 fasteners will be used in locations that will not experience high stresses or torsional loads. These include pillow block mounting locations and the lead screw mounts to the vertical platform.

All welded joints are to be completed by a skilled welder using clean, dry steel welding rods.

The stacker assembly consists of several components shown in the following exploded view:

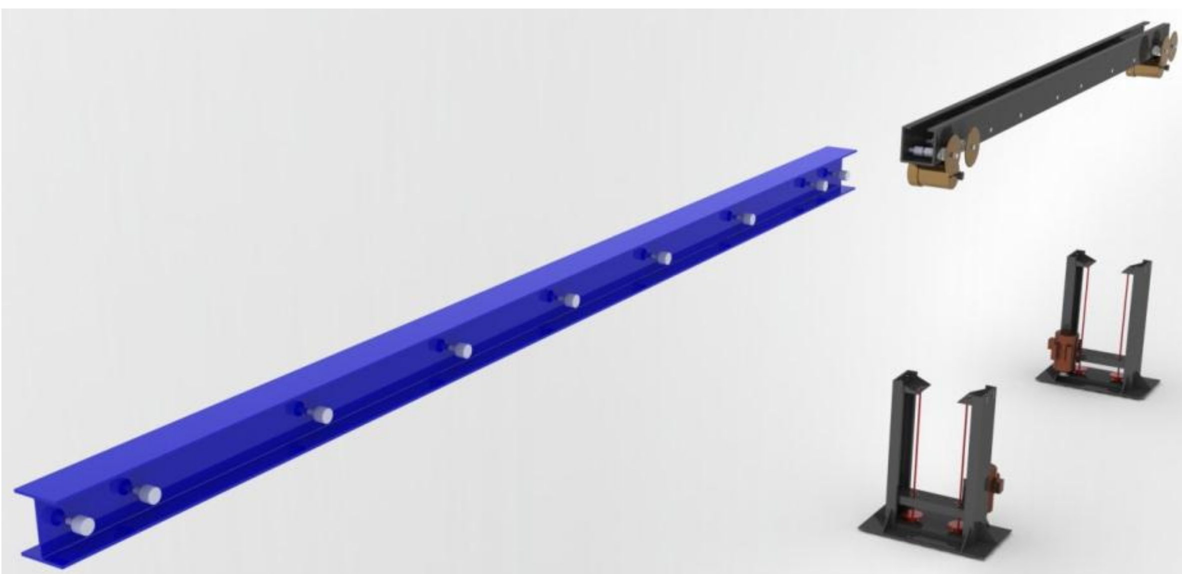


Figure 87: Stacker assembly - exploded view

D.1 Translating Assembly

The assembly and manufacture of the translating is broken down into three sections: I-beam and upper rollers, C-channel Beam and lower rollers, and horizontal drive system.

D.1.1 I-beam and Upper Rollers

The I-beam assembly consists of several components each shown in the following exploded view:

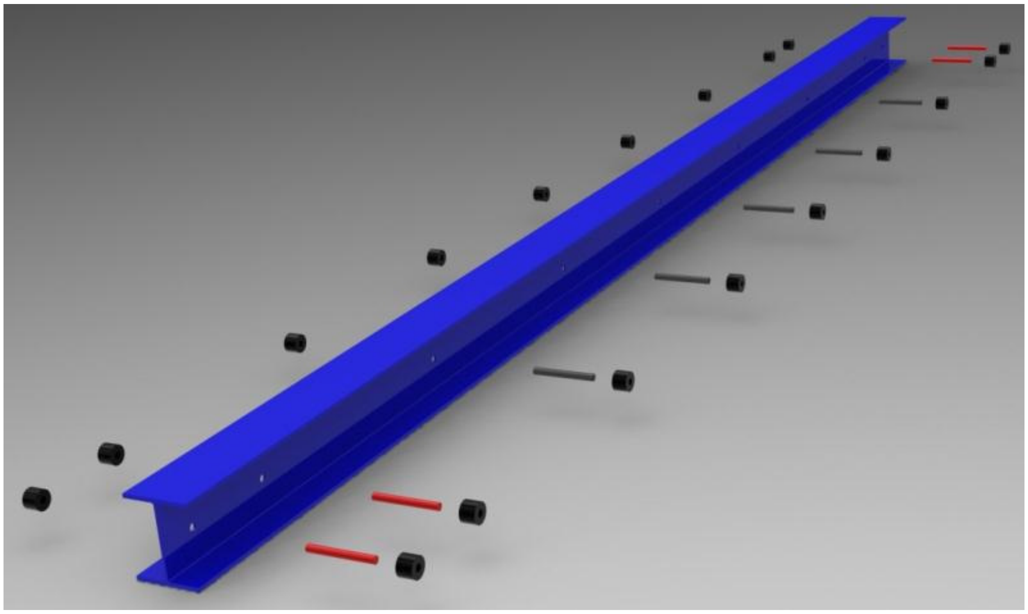


Figure 88: Exploded view of I-beam assembly

The I-beam requires nine ¾ in. diameter holes to be drilled through the center web to accommodate the nine upper idler shafts. The idler shafts are not required to rotate and can, therefore, be welded in place. The upper idler wheels will then be placed onto either end of the upper idler shaft, along with the ¾ in. retaining rings.

D.1.2 C-Channel Beam and Lower Rollers

The C-channel beam assembly consists of several components. Each component is shown in the following exploded view:

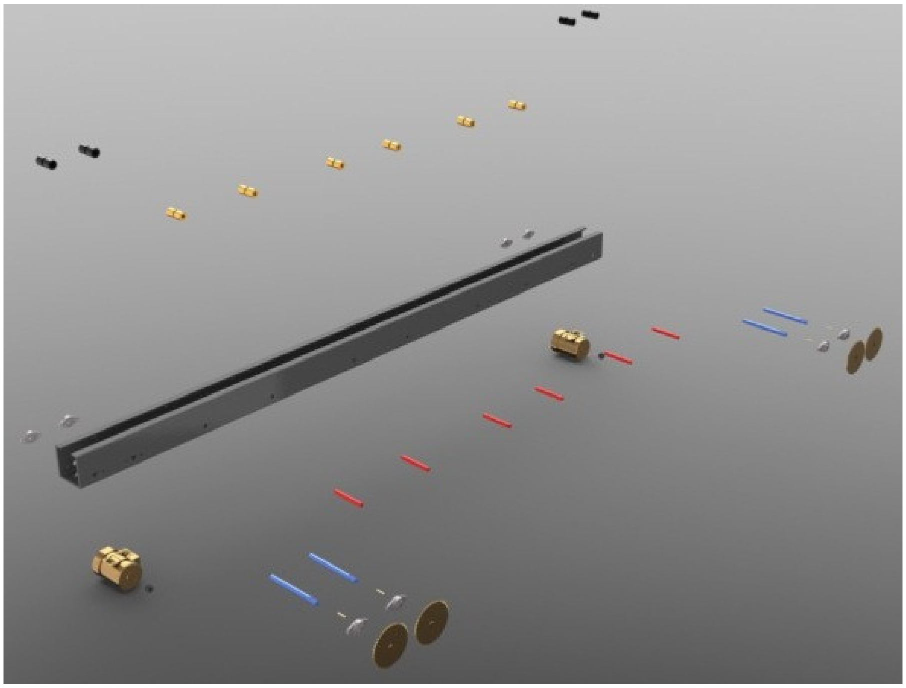


Figure 89: Exploded view of C-channel beam and horizontal drive system

The C-channel beam requires a 4 in. strip to be cut from the center of the top flange to allow the I-beam to move horizontally. There needs to be ten 1 in. diameter holes drilled on both sides to accommodate the ten lower roller shafts. There also needs to be two 29/64 in. holes drilled for the mounting bolts on each of the eight drive shaft bearings. On the bottom of the beam, eight ½ in. holes (four on each side) need to be drilled to bolt the translating assembly to the lift assembly. Finally, an additional eight 11/32 in. holes (four on each side) need to be drilled to attach the horizontal drive motors to the bottom of the C-channel beam.

The six lower idler shafts are inserted into the six center holes that were drilled into the side of the C-channel beam. After going through the first set of holes, the lower roller idler wheels can be positioned in the shafts, using the retaining rings to hold them in place (this is done inside the C-channel beam). Once this is done, the other end of the idler shaft can be inserted into the opposite hole and welded in place.

The eight drive shaft bearings are bolted onto the outside of the channel beam. The four drive shafts (with the 2 in. long ¼ in. keyway machined into the one end) are then inserted through the bearings on one side of the C-channel beam and the drive wheels are positioned in the drive shaft. The drive shaft is then inserted into the other bearings, and the bearing and drive wheel set screws are tightened to lock the drive shaft into place. The drive shaft should be positioned so that the end without the keyway sits flush with the bearing.

The I-beam is then inserted into the C-channel beam, on top of the lower rollers and the whole translating assembly is bolted onto the lift assembly.

D.1.3 Horizontal Drive System

The horizontal drive motors are bolted to the bottom of the C-channel beam making sure to include ½ in. spacers in order to achieve the proper centered distance between the sprockets. The motor drive shaft should be facing outward in the same direction as the key end of the roller drive shaft. The roller drive sprocket is then placed on the motor drive shaft so that the tooth end is flush with the end of the drive shaft. The 3/32 in. key is inserted into the keyway and the set screws are tightened to mate the two components.

The roller driven sprockets are then placed on the drive shafts making sure to offset the inner-most sprocket, so that it lines up with the second set of teeth on the roller drive sprocket. The ¼ in. key is inserted into the keyway and the set screws are tightened to mate the sprockets to the drive shafts. The chains are then applied to the horizontal drive system.

D.2 Lift Assembly

The lift assembly consists of several components, each shown in the following exploded view:

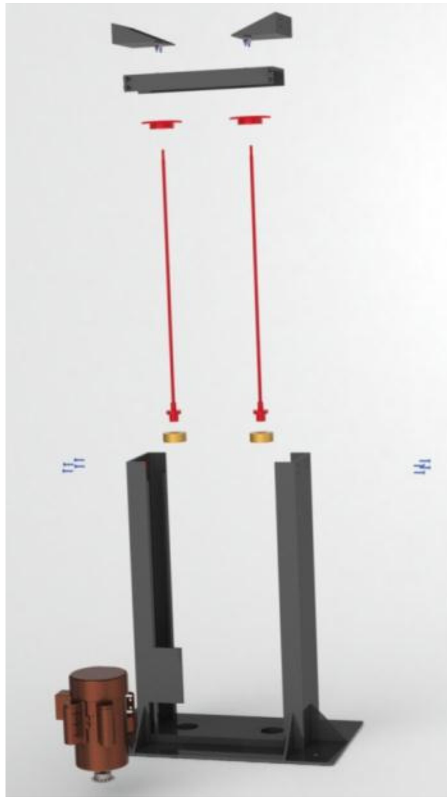


Figure 90: Lift assembly- exploded view

The first step in building the lift assembly is to manufacture all of the steel components. By following the technical drawings the proper components will be cut from steel sheets. Each section of components will be welded together into the configuration shown in the technical drawings. The resulting components will be: The upright posts with base, the horizontal platform, and the two lead screw mounts.

Next, the lead screw bearings are press-fit into the base of the upright post assembly. The upright post section is now ready to receive the other components.

Connect the sprockets to the lead screw and the drive motor using the set screws and key provided. Next, machine off the threads of a section of the lead screw then slip on a hub and weld it on one side. Grind the welds smooth. Machine a key groove into this hub and insert the sprocket key. Finally, slide the sprocket over this key and secure using retaining rings and the provided set screws.

To attach the motor, slide the sprocket over the existing key and secured using the set screws provided.

Next, insert the lead screws into the lower bearings while sliding the upper bearings over the upper portion of the lead screw.

The lead screw support sections are then secured to the side of the uprights using grade 8 bolts. The lead screw upper bearings should line up with the bearing mount holes on the lead screw support. Mount the bearings through these holes using grade 8 bolts.

Finally, mount the motor assembly upright on the motor mount using grade 8 hardware. Wind the ANSI #60 chain around the sprockets on the lead screws and motor. Set the tension of the chain by adjusting the chain length and distance of the motor from the sprockets using the slotted holes. Close the chain using the provided locking link.

D.3 Stacking Platform Assembly

The stacking platform requires a 1.5 in. diameter hole to be drilled 4 in. deep into a railway tie. The stopping rod requires a gusset to be welded 4 in. from the bottom of the rod so that when the rod is inserted into the railway tie, the gusset sits on top of it and provides stability.

D.4 Roller Drive Mechanism Assembly

The roller drive assembly makes use of the existing rollers and roller stands at the All-Fab facility. The first step in manufacturing the roller drive is to drill the specified holes in the roller stand. Next, the components of the motor mount assembly are cut from the A36 steel plate and welded together in the proper configuration. The specified holes are drilled in the motor mount, enabling it to be mounted on the roller stand using grade 8 hardware. Next, the drive sprocket is attached to the motor mount using the key and set screws provided. To ensure proper motor alignment, the position of the sprocket on the shaft should be carefully measured. Next, the motor is attached to the mount using grade 8 hardware.

Prior to mounting the driven sprocket, detach the roller and roller shaft assembly from the roller stand. This is done by unbolting the pillow block bearing and sliding it off of the shaft. A properly sized key slot should be machined into this shaft. If this is not possible the set screws provided may be used instead. Next, slide the driven sprocket onto the shaft and secure it in the proper location using retaining rings. Replace and re-secure the pillow block bearing to the stand. Finally, wrap the roller chain section around the two sprockets and secure with the locking link. Adjust the chain by loosening the mount bolts and sliding the mount up or down to achieve the proper tension.

Appendix E: Detailed Cost Analysis

The following is a detailed analysis of the materials and manufacturing costs of each component in the final design. The analysis is broken down into material cost and manufacturing costs per unit.

TABLE IX: DETAILED COST ANALYSIS

Horizontal Stacker			
Part:	Number of Parts Per Stacker:	Material Cost Per Part:	Manufacturing Cost Per Part:
Translating Assembly:			
I-Beam	1	$ 408.00	$ 100.00
C-Channel Beam	1	$ 750.00	$ 300.00
Drive Wheel	8	$ 24.85	$ 0.50
Lower Roller Idler Wheel	12	$ 25.00	$ 0.50
Upper Roller Idler Wheel	18	$ 20.00	$ 0.50
Lower Wheel Retaining Ring	24	$ 2.31	$ 0.10
Upper Wheel Retaining Ring	36	$ 1.76	$ 0.10
Lower Roller Drive Shaft	4	$ 6.50	$ 200.00
Lower Roller Idler Shaft	6	$ 4.00	$ 50.00
Upper Roller Idler Shaft	9	$ 2.27	$ 50.00
Roller Drive Shaft Bearing	8	$ 30.10	$ 5.00
Horizontal Drive Motor	2	$ 226.23	$ 15.00
Roller Drive Sprocket	2	$ 34.36	$ 5.00
Roller Driven Sprocket	4	$ 31.32	$ 5.00
Drive Shaft Key	4	$ 0.45	
Horizontal Drive Chain	4	$ 33.75	$ 5.00
Fasteners	16	$ 2.00	$ 0.10
Lift Assembly:			
Lead Screw	4	$ 22.59	$ 300.00
Lead Screw Nut	4	$ 31.45	$ 2.00
Lead Screw Flange	4	$ 44.78	$ 2.00
Lead Screw Plate	2	$ 30.00	$ 300.00
Upper Support	4	$ 200.00	$ 1,500.00
Lead Screw Top Support	4	$ 15.00	$ 200.00
Lower Lead Screw Bearing	4	$ 200.00	$ 100.00
Upper Lead Screw Bearing	4	$ 11.38	$ 5.00
Lead Screw Motor	2	$ 355.12	$ 15.00
Lead Screw Drive Sprocket	2	$ 19.70	$ 5.00
Lead Screw Driven Sprocket	4	$ 39.52	$ 5.00

Lead Screw Drive Chain	2	$ 19.42	$ 5.00
Fasteners	64	$ 2.00	$ 0.10
Total: Single Stacker		$ 6,497.45	$ 11,209.00
Total:		$ 38,984.70	$ 89,672.00
			$ 128,656.70
Main Roller Platform			
Part:	Number of Parts Per Roller:	Material Cost Per Part:	Manufacturing Cost Per Part:
Drive Motor	1	$ 269.06	$ 15.00
Driven Sprocket	1	$ 74.91	$ 5.00
Driving Sprocket	1	$ 13.79	$ 5.00
Roller Drive Chain	1	$ 22.12	$ 5.00
Motor Mount	1	$ 50.00	$ 200.00
Total: Single Roller		$ 429.88	$ 230.00
Total:		$ 2,149.40	$ 1,150.00
			$ 3,299.40
Stacking Support			
Part:	Number of Parts Per Support:	Material Cost Per Part:	Manufacturing Cost Per Part:
Stopping Rod	1	$ 36.00	$ 10.00
Railway Tie	1	$ --	$ 5.00
Gusset	1	$ 10.00	$ 50.00
Total: Single Support		$ 46.00	$ 65.00
Total:		$ 368.00	$ 520.00
			$ 888.00
Total Cost of System	$ 132,844.10		