The Design and Construction of an Intelligent Power Assist Jib Crane

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Abstract

Current material handling systems exhibit anisotropic behavior. That is, their two planar degrees of freedom require different amounts of force input from the operator. Movement of these devices is correspondingly difficult.

One of the three most prevalent material handling devices, the jib crane, is selected for research into creating isotropic motion. A platform for this research is created with two intended future directions in mind. The first direction creates isotropy by floating the boom over the load. The second direction adds a powered trolley and is capable of providing power assist to actually accelerate the load.

The thesis walks the reader through the design of the platform. Selection of the jib crane is followed by the design and selection of a servo motor and transmission capable of supporting both future directions. A cable angle sensor and ultrasonic distance sensor provide the intelligence necessary to achieve isotropy.

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1 Introduction

Entrance into the durable goods manufacturing plant of today is marked by the dominance of overhead material handling systems. The purpose of these devices is simply to help workers move objects, weighing from 30 pounds to 100 tons, from one point to another. The overhead material handling systems discussed here have three degrees of freedom (DOF). The two degrees of freedom in the horizontal plane are provided by motion of beams and trolleys, while the third, vertical degree of freedom is provided by a hoist. There exist three such types of systems.

The first 3-DOF material handling device and the one which is the subject of this thesis, is the jib crane, depicted in Figure 1.1. An anatomical description begins with the mast. Figure 1.1 shows a mast which is secured to the floor, but other types of masts can be mounted to a wall or ceiling. A rotational joint permits the boom (translucent in the figure) to rotate about the mast. While Figure 1.1 shows an axle and bushing type joint, joints with two or more hinges (like on a door) are common. There are two types of booms: enclosed track booms (shown in the figure) have a channel in which a trolley rides. This design keeps the rolling surface clean, significantly reducing rolling friction. The greater stiffness per unit mass of the I-beam boom provides increased capacity, greater span, and slightly lower inertia. A hybrid boom combines the stiffness of the I-beam with the low friction of the enclosed track by stacking two such beams on top of one another. To finish up the description, a trolley translates along this boom. A cable or chain, often in conjunction with a hoist, suspends a load from the trolley.

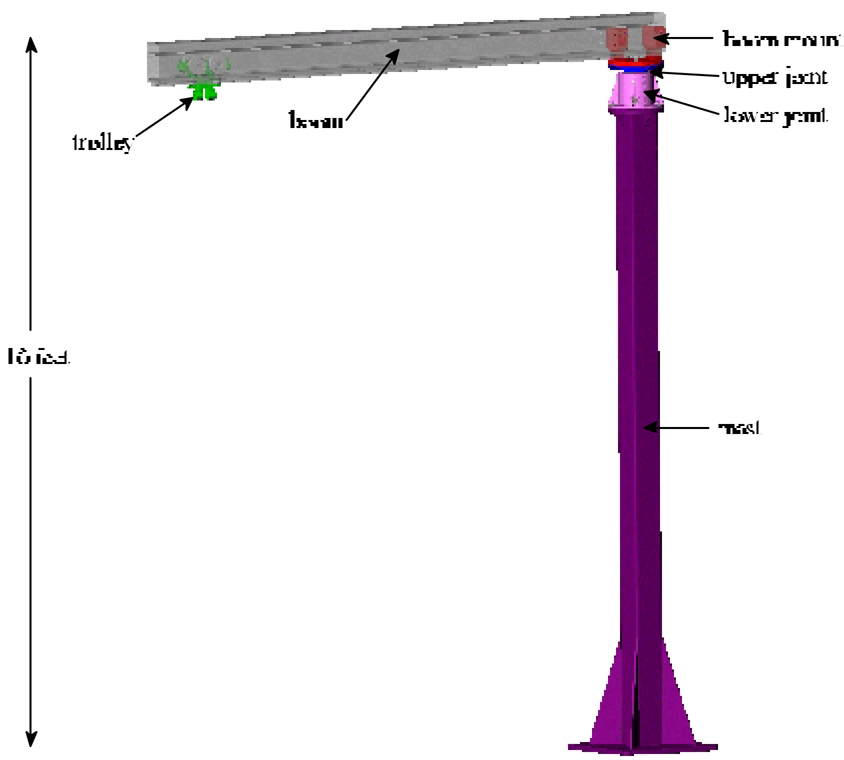


Figure 1.1 *Jib Crane Anatomy* This figure depicts the parts of a typical jib crane. The boom is shown translucent for clarity.

A variant of the standard jib crane is the articulating arm jib crane, shown in Figure 1.2. It has no trolley and thus no boom. Instead, it has two arms joined by a pivot joint, which allows up to 360 degrees of rotation of the second arm relative to the first. There is an attachment point at the end of the second arm for a chain, cable, or hoist. The advantage of the articulating arm over the standard jib is as follows: the moment of inertia of the two arms is generally less than that of a comparable boom. This means motions perpendicular to the boom's length on a standard jib crane require more force. Forces along the boom on a standard jib crane, however, often require less force due to the even lower inertia of the trolley.

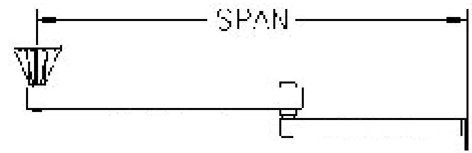


Figure 1.2 *Articulating Arm Jib Crane*

The two other types of overhead material handling machines are the gantry crane (Figure 1.3) and bridge crane (Figure 1.4). Both types have a rail (identical to a boom) and trolley similar to those on the jib crane. On the gantry crane, though, the rail is rigidly attached to two supports which roll along the ground, often in tracks. On the bridge crane, one or two rails form a bridge across two fixed rails. In Figure 1.4, the one rail which forms the bridge is shown in yellow. This bridge then has two or four trolleys at each end to allow movement along the fixed rails.

The three material handling machines exhibit a tradeoff between high load capacity and low space infringement. The cantilevered mount of the jib crane gives it a lower capacity than similarly sized gantry and bridge cranes, where the boom is supported at both ends. This same characteristic, however, allows the jib crane to take up significantly less floor space. The bridge crane necessitates the construction of a large frame around the perimeter and/or over the top of its rectangular work space. The gantry crane usually necessitates the mounting of track (for safety reasons) and the existence of the two mobile supporting columns along its rectangular workspace. In contrast, a jib crane merely requires the mounting of a slender mast at the center of a circular workspace or the attachment to a wall along a semicircular workspace. The large rolling mass of the gantry crane creates a safety problem and often necessitates powered motion; thus it is generally restricted to outdoor applications where the use of tracks allow for long

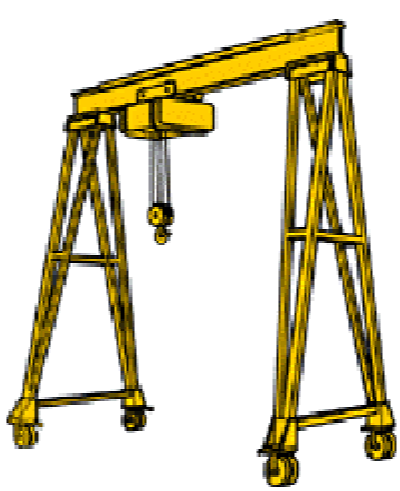


Figure 1.3 *Gantry Crane*

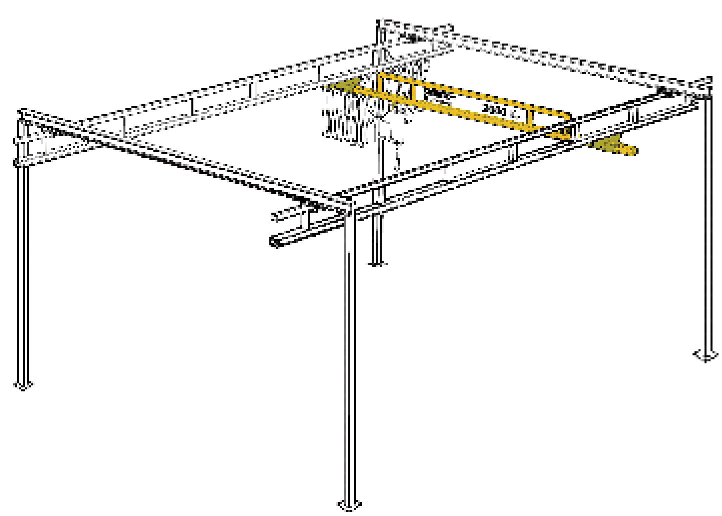


Figure 1.4 *Bridge Crane*

distance movements. For this reason, jib and bridge cranes are the most prevalent overhead material handling devices *inside* assembly plants.

For each handling device, the two horizontal DOF differ in feel. For the jib and gantry cranes, motion along the trolley's direction of travel requires relatively little force. Similarly, on the bridge crane, motion along the bridge is relatively easy. But for the jib crane, moving perpendicular to the boom's length is complicated by the boom's rotational inertia and the boom pivot's friction. For the gantry crane, this perpendicular motion is very difficult due to the large inertia of the crane, and as stated earlier, often requires power assist. Finally, for the bridge crane, it is the large inertia of the bridge and resulting increased rolling resistance in its trolleys that make the perpendicular motion more difficult.

Low speed power assist has been added to each of these difficult motions: rotation of the boom on a jib crane, translation of a gantry crane, and translation of the bridge along the fixed rails of a bridge crane. The control of this power has been limited to simple push button on/off switches and control algorithms which provide a slow start and stop to minimize load swing. The anisotropy exhibited in the different motions remains.

This lack of isotropy makes it difficult for workers to move an object from point to point. The worker must constantly vary the applied force as the desired direction of motion changes. To think of this phenomenon in different terms, consider a round stick half immersed in a pool of water. It is very easy to move this stick through a curved trajectory. The forces resisting the motion are independent of the motion's direction. Now consider a paddle (which cannot be rotated) immersed in the pool. It is considerably more difficult to move this paddle through the same curved trajectory. The human must continually adjust to the changing resistance forces. Increasing the viscosity of the fluid will exacerbate the problem. This is analogous to increasing the size and/or load of a material handling system.

Thus, ideally, overhead material handing devices would exhibit isotropy. That is, the feel of the device would be independent of the direction in which it was pushed. That leads into the objective of this thesis: to design and construct a platform for an intelligent power assist jib crane which can exhibit isotropic dynamic behavior throughout its workspace. The next section in this thesis will walk the reader through the design of this intelligent material handling system.

2 Design Analysis

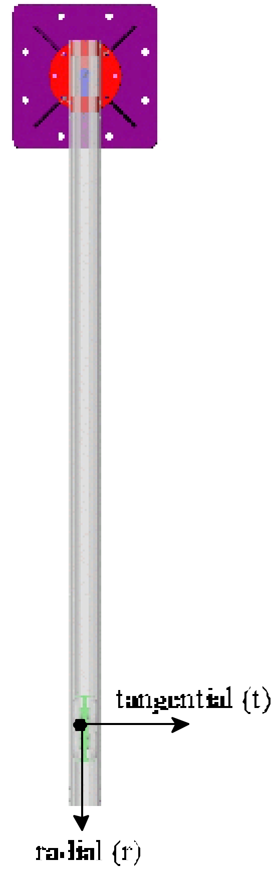


Figure 2.1
Coordinate System

To begin the design analysis, refer to the coordinate system defined in Figure 2.1. This is an overhead drawing of the same jib crane depicted in Figure 1.1. The origin of the coordinate system is attached to the trolley. The positive radial direction points from the trolley to the free end of the boom, while the positive tangential direction points in a perpendicular direction which would correspond to counterclockwise rotation of the boom about the mast. The positive vertical direction, not shown in the figure, is simply up, out of the page in Figure 2.1.

The unit system of choice is normally SI. For this design, though, the BG (British Gravitational) system is chosen for several reasons. First, the dimensions of most acquired components are in inches. Second, in America, the available English components far outnumber their metric counterparts. Finally, and most significantly, the machine shop available to the author uses primarily BG units. Though the author prefers SI, its selection would have unnecessarily complicated the design process. Keep in mind that the gravitational constant which complicates work in the EE (English Engineering) system does not arise in the BG system. Conversion factors from the BG system to SI are provided in Appendix C.

All currently available jib cranes have a passive trolley; the trolley is free to roll along the boom. In a low friction enclosed boom, radial movements feel almost as if the load is floating in air. To make the jib crane's motion isotropic, this floating in air feeling will have to be extended to the tangential direction. This can be accomplished by using power assist at the pivot to angularly accelerate the upper pivot, boom mount, boom, and trolley to provide a linear acceleration equal to the F/m acceleration the operator is imparting to the load. The boom and other components will thus follow the accelerating load and the operator will not feel their presence.

But current, parallel research into power assisted trolleys suggests different needs for an isotropic transformation. Though a floating in air feeling may sound nice, the user is still required to provide the force to accelerate the load, which can be very large. The addition of a power assist trolley allows some to all of this operator pushing force in the radial direction to be handled by the crane. To make a jib crane with a power assist trolley appear isotropic additional tangential acceleration is required because now the load also has to be accelerated. In fact, a lot more acceleration is required. Equation 2.12, an expression for the jib crane's moment of inertia (derived later), shows that the addition of a 200 lb load increases the moment of inertia that must be accelerated by a factor of 9.

The initial control strategy applied to the jib crane platform will be to simulate an isotropic feel with a passive trolley. This will only require acceleration of the boom, trolley, boom mount, and upper pivot. The platform will be designed, however, to be capable of also accelerating the suspended load and thus will have the potential to provide an isotropic feel to a jib crane with a power assist trolley.

Jib Crane Selection

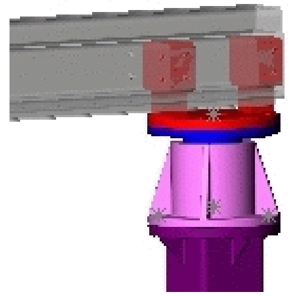


Figure 2.2a *Bushing Joint*

Figure 2.2b *Hinge Joint*

The first step in the design process is to choose the jib crane. Articulated arm jib cranes are not considered due to their lack of a rigid boom and trolley. Floor mounting is chosen over wall or ceiling mounting due to lab constraints. The fact that floor mounted jibs cranes are the most prevalent further supports this decision; research results will then be more widely applicable. The desire for low radial rolling resistance and the lack of need for a high load capacity or span, leads to the selection of an enclosed track boom. Finally, the type of joint is chosen based on the ease with which power assist can be applied. Referring to Figure 2.2a, a gear, pulley or sprocket could be inserted between the blue upper pivot and red boom mount. A motor could be attached to the mast and would allow 360 degrees of rotation of the boom. The hinge joint, shown in Figure 2.2b, complicates matters by moving the pivot away from the center of the mast and introduces the mast as an obstruction to the side hinge mounted boom. 360 degrees of rotation is impossible. A bushing joint is thus chosen.

The jib crane shown in Figures 1.1, 2.1, and 2.2a as a typical example, is the actual jib crane selected. This crane, model 900J, is obtained free of charge from Zimmerman Handling Systems, a division of Ingersoll Rand. It is 9' 7" tall, and has a boom length of 8'. Its maximum capacity is 200 pounds. Refer to Appendix A for further details.

Acceleration and Velocity Goals

To determine the desired torque and speed for the boom's angular rotation, experiments are performed. The jib crane had not yet arrived, so the tests were conducted on a fixed rail of an existing bridge crane, as shown in Figure 2.3. 150 pounds of weight are attached to a passive trolley similar to the one which will be first employed on the jib crane. An accelerometer is attached to the weights. The weights are accelerated back and forth at a rate which feels like the maximum that would be desired. Data acquired from the accelerometer shows the maximum acceleration to be 0.15 g's or 4.83 ft/s^2. This is set as the design goal for the maximum linear acceleration of the trolley in the tangential direction.

Figure 2.3 *Velocity and Acceleration Experimental Platform*

To determine the maximum velocity desired, the weights are moved along the rail at again what feels like the maximum desired speed. A stopwatch and marked off distance determine this velocity to be 3.3 ft/s.

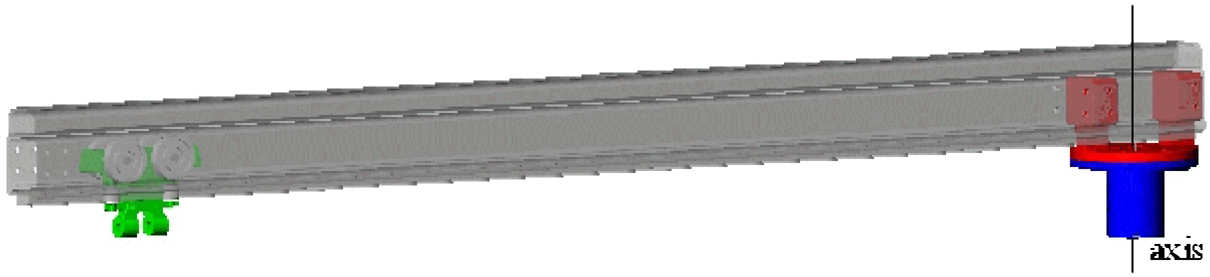


Figure 2.4 *Components of the Jib Crane which Add to the Rotational Moment of Inertia about the shown Axis*

The “what feels right” is based on trips to an automotive powertrain plant and Zimmerman Handling Systems. Multiple jib cranes were operated by the writer and the operation of several jibs by plant personnel was observed.

Mass Moment of Inertia Calculations

Recalling that T = I", to determine the torque that needs to be applied to the boom to accelerate the trolley at the desired linear acceleration, the moment of inertia of the jib crane’s moving parts about the axis shown in Figure 2.4, is determined. Since this value must be determined before the jib crane has arrived, it is done theoretically rather than experimentally. The parts which contribute significantly to the moment of inertia about the stationary mast include:

1. Moving part of pivot (blue)
2. Boom mount (red)
3. Boom (silver)
4. Trolley (green)
5. Load and Cable

The colors reference the parts shown in Figure 2.4. The 3-D drawing is done after the jib crane arrives, but is included here for clarity.

Mass Moment of Inertia of Upper Pivot

The geometry of the moving part of the pivot is not yet known. Employing a worst-case assumption, it is modeled as the combination of two solid cylinders:

cylinder 1 dimensions: 6.63 in x 2 in (H x R)
cylinder 2 dimensions: ½ in x 4.5 in (H x R)

Assuming a medium-carbon steel, the masses are...

$$m = p r^2 h r \tag{2.1a}$$

$$m_{c1} = p\left(2/12\,ft\right)^2\left(6.63/12\,ft\right)\left(15.2\frac{slug}{ft^3}\right) = 0.733 slug \tag{2.1b}$$

$$m_{c2} = p\left(4.5/12\,ft\right)^2\left(0.5/12\,ft\right)\left(15.2\frac{slug}{ft^3}\right) = 0.280 slug \tag{2.1c}$$

The mass moments of inertia are...

$$I = \frac{1}{2} m r^2 \tag{2.2a}$$

$$I_{c1} = \frac{1}{2}(0.733 slug)(2/12\,ft)^2 = 0.0102 slug \cdot ft^2 \tag{2.2b}$$

$$I_{c2} = \frac{1}{2}(0.280 slug)(4.5/12\,ft)^2 = 0.0197 slug \cdot ft^2 \tag{2.2c}$$

The combined moment of inertia is...

$$I_{pivot} = 0.0299 slug \cdot ft^2 \tag{2.3}$$

Mass Moment of Inertia of Boom Mount

The geometry of the boom mount is closely approximated by modeling it as the combination of the following shapes, progressing from the bottom to the top:

solid cylinder: 0.75 in x 4.5 in (H x R)

parallelepiped: 1.3 in x 1.4 in x 9 in (H x W x L)
negative[1] parallelepiped: 1.3 in x 1.4 in x 5 in (H x W x L)

parallelepiped: 3.5 in x 3.5 in x 9 in (H x W x L)
negative parallelepiped: 3.5 in x 3.5 in x 5 in (H x W x L)

The masses are...

$$m_{c3} = pr^2 h r = p(4.5/12\,ft)^2(0.75/12\,ft)\left(15.2\frac{slug}{ft^3}\right) = 0.420 slug \tag{2.4a}$$

$$m_{parallelepiped} = hwlr \tag{2.4b}$$

$$m_{p1} = (1.3/12\,ft)(1.4/12\,ft)(9/12\,ft)\left(15.2\frac{slug}{ft^3}\right) = 0.144 slug \tag{2.4c}$$

$$m_{np1} = (1.3/12\,ft)(1.4/12\,ft)(5/12\,ft)\left(15.2\frac{slug}{ft^3}\right) = 0.0800 slug \tag{2.4d}$$

[1]The moment of inertia of some shapes is most easily calculated by subtracting an easily calculated value from another.

$$m_{p2} = (3.5/12\ ft)(3.5/12\ ft)(9/12\ ft)\left(15.2\frac{slug}{ft^3}\right) = 0.970 slug \tag{2.4e}$$

$$m_{np2} = (3.5/12\ ft)(3.5/12\ ft)(5/12\ ft)\left(15.2\frac{slug}{ft^3}\right) = 0.539 slug \tag{2.4f}$$

The mass moments of inertia for the individual geometries are...

$$I_{c3} = \frac{1}{2}mr^2 = \frac{1}{2}(0.420 slug)(4.5/12\ ft)^2 = 0.0295 slug \cdot ft^2 \tag{2.5a}$$

$$\begin{aligned} I_{netp1} &= \frac{1}{12}m_{p1}\left(w_{p1}^2 + l_{p1}^2\right) - \frac{1}{12}m_{np1}\left(w_{np1}^2 + l_{np1}^2\right) = \frac{1}{12}(0.144 slug)\left[(1.4/12\ ft)^2 + (9/12\ ft)^2\right] \\ &- \frac{1}{12}(0.0800 slug)\left[(1.4/12\ ft)^2 + (5/12\ ft)^2\right] = 5.67x10^{-3} slug \cdot ft^2 \end{aligned} \tag{2.5b}$$

$$\begin{aligned} I_{netp2} &= \frac{1}{12}m_{p2}\left(w_{p2}^2 + l_{p2}^2\right) - \frac{1}{12}m_{np2}\left(w_{np2}^2 + l_{np2}^2\right) = \frac{1}{12}(0.970 slug)\left[(3.5/12\ ft)^2 + (9/12\ ft)^2\right] \\ &- \frac{1}{12}(0.539 slug)\left[(3.5/12\ ft)^2 + (5/12\ ft)^2\right] = 0.0407 slug \cdot ft^2 \end{aligned} \tag{2.5c}$$

The total mass moment of inertia for the boom mount is then simply the sum of these three values...

$$I_{mount} = 0.0759 slug \cdot ft^2 \tag{2.6}$$

Mass Moment of Inertia of Boom

The boom is modeled as a long slender rod of equal mass. The mass of the 8 foot boom is...

$$m_{boom} = (8\ ft)\left(7.50\frac{lb}{ft}\right)\left(\frac{1 slug}{32.2 lb}\right) = 1.86 slug \tag{2.7}$$

It's moment of inertia is...

$$I_{boom} = \frac{1}{3}mL^2 = \frac{1}{3}(1.86 slug)(8\ ft)^2 = 39.7 slug \cdot ft^2 \tag{2.8}$$

To approximate the error in the modeling assumption, the same procedure is carried out on the cross section shown in Figure 2.5, which is similar to the ZRA2 designated cross section of the Zimmerman boom. The exact and slender rod assumption mass moments of inertia are calculated and the percent error is only 1 percent, thus the calculated moment of inertia is deemed close enough.

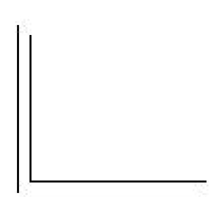


Figure 2.5 Cross section for error analysis

Mass Moment of Inertia of Trolley

The geometry of the trolley is quite complex. It is modeled as a thin rectangular plate with the following dimensions:

simplified trolley dimensions: 8 in W x 6 in H x 1 in D

Since the moment of inertia added by the parallel axis theorem (which is easily calculated from the known mass) far exceeds the moment of inertia about its own axis, the trolley geometry simplification should not affect results significantly. The mass moment of inertia of the trolley about the mast will be a function of r, the radial distance of the trolley from the mast axis...

$$I_{trolley} = \frac{1}{12}mw^2 + mr^2 \tag{2.9}$$

$$I_{trolley} = \frac{1}{12}(5.5lb)\left(\frac{1slug}{32.2lb}\right)(8/12ft)^2 + \left(5.5lb\frac{1slug}{32.2lb}\right)r^2 = \left(6.33x10^{-3} + 0.171r^2\right)slug\cdot ft^2 \tag{2.10}$$

Note that at the minimum radial distance of one foot, the mass moment of inertia of the trolley about its own axis makes up less than four percent of the total moment of inertia, thus validating the geometry assumption.

Mass Moment of Inertia of Load and Chain

The research jib crane will likely not include a hoist, therefore it is ignored in this analysis. Since the moment of inertia of the hoist and chain about their own vertical axis is negligible compared to the moment added by the radial distance, they could be incorporated into the mass of the trolley. Since the load is free to rotate about its own vertical axis, the mass moment of inertia about its own axis is ignored. The load's mass moment of inertia is...

$$I_{load} = m\cdot r^2 = (W/32.2)r^2 \quad \text{slug}\cdot\text{ft}^2 \tag{2.11}$$

Total Moment of Inertia

Addition of the previously calculated moments yields...

$$\begin{aligned} I_{total} &= I_{pivot} + I_{mount} + I_{boom} + I_{trolley} + I_{load} = 0.0299 + 0.0759 + 39.7 + \left(6.33x10^{-3} + 0.171r^2\right) \\ &+ (W/32.2)\cdot r^2 = \left(39.8 + r^2\left(0.171 + \frac{W}{32.2}\right)\right)slug\cdot ft^2 \end{aligned} \tag{2.12}$$

The mass moment of inertia of the jib crane's moving parts is thus a function of both the trolley's radial position and the weight of the load. As previously stated the inertia of the load is not accelerated in an isotropic jib crane with a passive trolley; the operator accelerates this load. The load *would* have to be accelerated, though, in the proposed isotropic jib crane with a power assist trolley in which the load *is* accelerated.

Design Torque

Recalling the linear acceleration goal of 0.15 g's or 4.83 ft/s^2, the desired angular acceleration is...

$$a = \frac{a}{r} = 4.83\frac{ft}{s^2}\cdot\frac{1}{r} = \frac{4.83}{r}\frac{1}{s^2} \tag{2.13}$$

Again recalling the familiar equation...

$$T = I \tag{2.14}$$

where: T = torque applied to the boom
I = the total mass moment inertia of the moving parts of the crane about the mast
" = angular acceleration of the boom about the mast

substitution of known values yields...

$$T = \left(39.8 + r^2\left(0.171 + \frac{W}{32.2}\right)\right)\left(\frac{4.83}{r}\right) lb\cdot ft \tag{2.15}$$

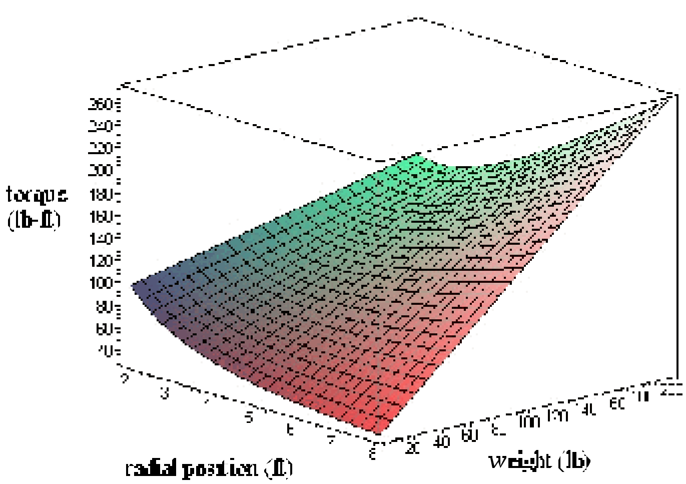


Figure 2.6 *Torque Necessary to Achieve Acceleration Goal as a Function of Trolley Radial Position and Load Weight*

Referring to Figure 2.6, this function reaches a maximum value of 271 lb-ft at the maximum radial position and maximum weight. This is the torque required to accelerate the load with no help from the operator. In a passive trolley system, the operator provides all the force to accelerate the load. In a powered trolley system the operator provides a fraction of this force. To ensure enough power is available, the 271 lb-ft value assuming no operator assist is chosen as the design goal.

Design Angular Speed

Recall that the linear speed goal is 3.3 ft/s. The desired angular speed, in rpm, of the boom is determined from...

$$= \frac{v}{r}\frac{60}{2p} \tag{2.16}$$

and is plotted in Figure 2.7. It is decided that movements inward of a 1 ft radial position are not necessary, so the maximum desired angular velocity of the boom, corresponding to a radial position of 1 ft, is 31.5 rpm.

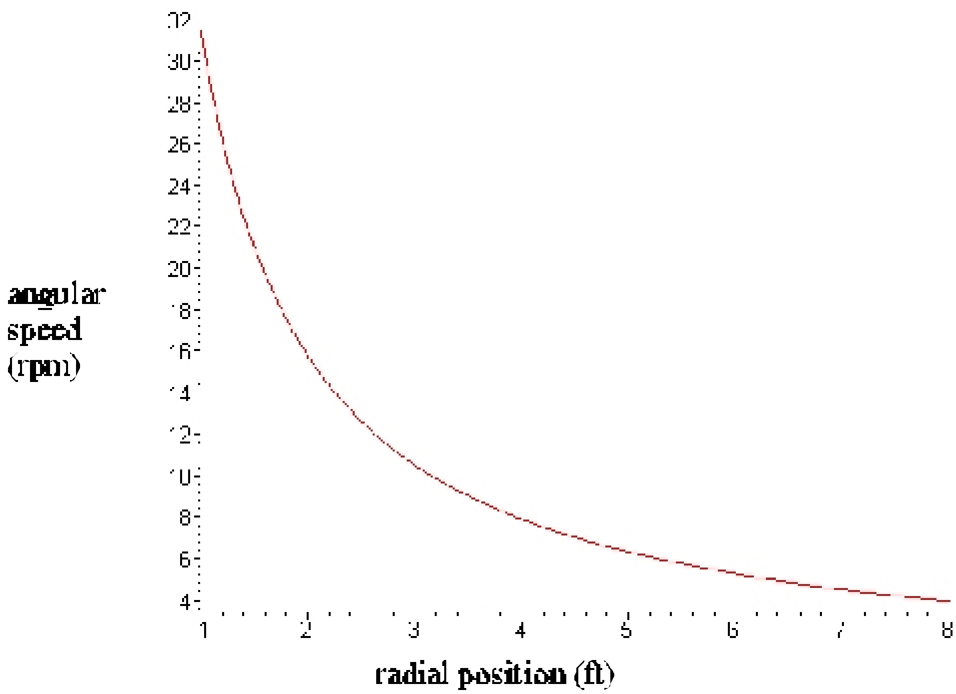


Figure 2.7 *Desired Angular Speed as a Function of the Trolley's Radial Position*

Transmission Type Selection

To transmit torque to the boom a transmission element will be inserted between the blue upper pivot and red boom mount, as shown in Figure 2.8. The power source to which this transmission connects, assumed to be a motor, is also shown in the figure. The manner in which this motor attaches to the mast is discussed later. Three key characteristics are sought from the transmission: high strength, ability to handle misalignment, and synchronization. The high strength requirement is obvious from the need to transmit 271 lb-ft of torque (that's more torque than the Ferrari F355's engine produces). Misalignment must be handled due to the inevitable flexure of the motor mount. Usually a motor is attached to a plate which also contains the bearing for the other end of the transmission, making alignment easy and flexure minimal. Notice in Figure 2.8, though, that this is not possible. In fact no simple method for attaching the motor is apparent. So flexure must be accommodated. Finally, a synchronous drive, one in which the rotational position of one transmission element relative to another is guaranteed, is desired. Slippage could result in poorly controlled performance and a very dangerous situation.

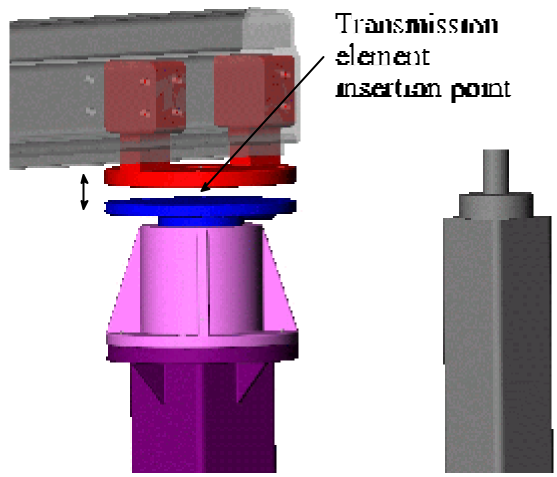


Figure 2.8 *Location of Transmission*

Four types of transmissions are considered: belt, cable, gear, and rollerchain. The spans of belts between their pulleys generally allows for small misalignment. Synchronization can be achieved by toothing the belt. But neither synchronous toothed flat belts, nor nonsynchronous untoothed flat belts or V-belts are capable of handling the torque requirement.

Cable transmissions are the most unusual of the four. Referring to Figure 2.9, two ends of a cable are secured to the ends of a section of pulley. The middle part of the cable is then wrapped around the smaller pulley several times. The power transmission is dependent upon the large frictional forces generated by the wrapping of the cable. Slippage is possible, though, so the transmission is nonsynchronous. The transmission also does not accommodate misalignment well. Little design information is available for cable drives. Though cable with enough strength is certainly available, the tight wrapping of high strength cable is not recommended. So employment of a cable drive would involve considerable uncertainty. Finally, while the workspace of this jib crane is only a semicircle, many industrial applications employ a full 360 degree workspace. A cable drive cannot provide 360 degrees of motion of the large pulley.

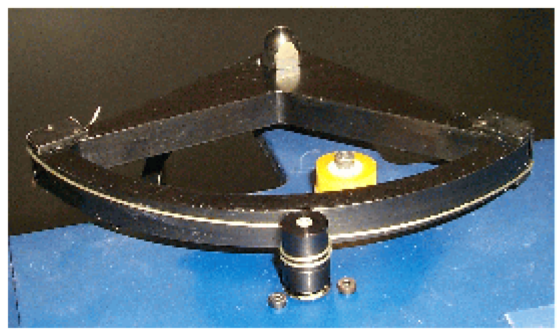


Figure 2.9 *Cable Transmission*

Gear transmissions, the most common in high load applications, provide the strength and synchronization attributes sought. The problem arises in the need for precise alignment. As mentioned earlier, the motor

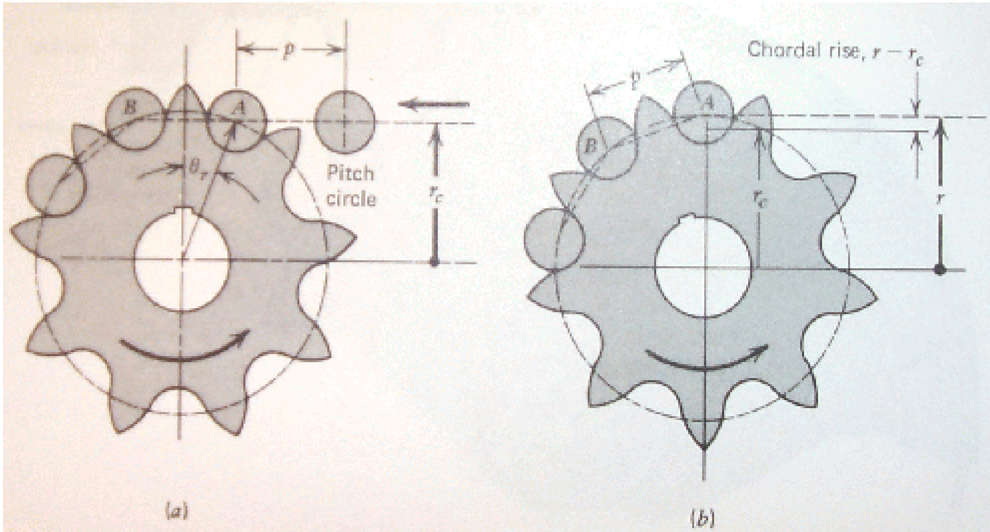


Figure 2.10 *Chordal Action* [1]

and transmission's other end cannot be attached to a common plate, as is usually done with gears. Some type of arm will have to be extended from the mast. Thus, the alignment required for gears would be very difficult to achieve. On the other hand, if a motor mount could be cast into the pink lower pivot, a gear transmission might be the best selection.

Many types of chain transmission exist, but rollerchain is by far the most widely used. It is best known for its employment on the bicycle. The steel chain and sprockets provide the necessary load capacities. Like on belt drives, misalignment is accommodated by the distance between sprockets and flexure of the chain. Finally, synchronization is provided by the toothed sprockets. Rollerchain has the three sought attributes and is thus the selected transmission.

Rollerchain does exhibit an undesirable attribute called chordal action. Referring to Figure 2.10a, when a roller chain link is first fully seated in a sprocket, the center of the chain is at a chordal radius, r_c. After the sprocket has rotated through an angle of 2_r, the center of the chain is now at the sprocket pitch radius, r (Figure 2.10b). The difference between these two radii is called the chordal rise...

$$\Delta r = r - r_c = r(1 - \cos q) = r\left[1 - \cos\left(\frac{180^o}{N}\right)\right] \tag{2.17}$$

where N is the number of teeth in a sprocket. The effect of chordal action is variation of the transmission speed ratio as the effective sprocket pitch radius varies between r and r_c. In a properly designed drive this fluctuation is for the most part absorbed by the elasticity of the chain. The phenomenon also becomes less significant with decreased speed. The boom's maximum angular speed of 31.5 rpm is considered low. The significance of chordal action is addressed later in the design analysis. The explanation is from [1].

Rollerchain Selection

The first step in selecting a rollerchain is the determination of a service factor. This value is multiplied by the normal operating power to get a design power. The factor accounts for the various types of loads and operating conditions. Different reference sources will give slightly different values, but for a reversing application with frequent starts and stops the suggested service factor is generally 1.5.

Next, the normal operating power is conservatively estimated by multiplying the maximum desired torque by the maximum speed...

$$p_{no} = (271\, lb \cdot ft)(31.5\, rpm)\left(\frac{2\pi\, rad}{rev}\right)\left(\frac{1\min}{60\sec}\right)\left(\frac{1hp}{550\frac{lb \cdot ft}{s}}\right) = 1.63hp \qquad (2.18)$$

Multiplication by the service factor gives the design power...

$$p_d = (1.5)(1.63hp) = 2.44hp \qquad (2.19)$$

It is important to note that this power value is very conservative. It is actually a peak value. It will never be seen for a significant amount of time. But, realizing that failure of the chain could have catastrophic results, conservatism is appropriate.

Rollerchain power rating tables, which give the design power as a function of small sprocket teeth number and small sprocket speed for each chain size, are then used to determine what size chain to select. Figure 2.11 shows the table for size 60. The smallest size which satisfies the design power is employed in order to be economical and minimize rotational inertia. Assuming

3/4" Pitch No. 60											
Teeth on Small Sprocket	Revolutions per Minute – Small Sprocket										
	50	100	200	500	700	900	1200	1400	1600	1800	2000
11	0.77	1.44	2.69	6.13	8.30	10.4	11.9	9.41	7.70	6.45	5.51
12	0.85	1.58	2.95	6.74	9.12	11.4	13.5	10.7	8.77	7.35	6.28
13	0.92	1.73	3.22	7.34	9.94	12.5	15.2	12.1	9.89	8.29	7.08
14	1.00	1.87	3.49	7.96	10.8	13.5	17.0	13.5	11.1	9.26	7.91
15	1.08	2.01	3.76	8.57	11.6	14.5	18.8	15.0	12.3	10.3	8.77
16	1.16	2.16	4.03	9.19	12.4	15.6	20.2	16.5	13.5	11.3	9.66
17	1.24	2.31	4.30	9.81	13.3	16.7	21.6	18.1	14.8	12.4	10.6
18	1.31	2.45	4.58	10.4	14.1	17.7	22.9	19.7	16.1	13.5	11.5
19	1.39	2.60	4.85	11.1	15.0	18.8	24.3	21.4	17.5	14.6	12.5
20	1.47	2.75	5.13	11.7	15.8	19.8	25.7	23.1	18.9	15.8	13.5
21	1.55	2.90	5.40	12.3	16.7	20.9	27.1	24.8	20.3	17.0	14.5
22	1.63	3.05	5.68	13.0	17.5	22.0	28.5	26.6	21.8	18.2	15.6
23	1.71	3.19	5.96	13.6	18.4	23.1	29.9	28.4	23.3	19.5	16.7
24	1.79	3.35	6.24	14.2	19.3	24.1	31.3	30.3	24.8	20.8	17.8
25	1.87	3.50	6.52	14.9	20.1	25.3	32.7	32.2	26.4	22.1	18.9
26	1.95	3.65	6.81	15.5	21.0	26.4	34.1	34.2	28.0	23.4	20.0
28	2.12	3.95	7.37	16.8	22.8	28.5	37.0	38.2	31.3	26.2	22.4
30	2.28	4.26	7.94	18.1	24.5	30.8	39.8	42.4	34.7	29.1	24.8
32	2.45	4.56	8.52	19.4	26.3	33.0	42.7	46.7	38.2	32.0	27.3
35	2.69	5.03	9.38	21.4	29.0	36.3	47.1	53.4	43.7	36.6	31.3
40	3.11	5.81	10.8	23.7	33.5	42.0	54.4	62.5	53.4	44.7	38.2
45	3.53	6.60	12.3	28.1	38.0	47.7	61.7	70.9	63.7	53.4	45.6
	Type A	Type B			Type C						

Figure 2.11 *Rollerchain Power Rating Table*

a 4:1 ratio (implying 126 rpm) and a minimum of 15 teeth on the small sprocket, no. 60 rollerchain is the correct selection. The pitch, 3/4 inch in this case, is simply the distance between roller axles. If it is found later that a lower transmission ratio is required, a larger size chain must be chosen.

With the chain size tentatively chosen, the next step is to choose the number of teeth on the sprockets. But the motor and possible motor gearhead selection depend on the rollerchain transmission ratio. Therefore, instead of prematurely deciding on sprocket sizes, the possible ratios will be explored.

Allowable Transmission Ratios

The large torque and relatively low speed requirements of this application indicate that the transmission should be a speed *reduction*. For low speed operation, it is recommended that the small sprocket, the one on the motor in this case, should have no fewer than 15 teeth. With fewer teeth, chordal action will become a problem. The large sprocket size must be larger than the 9 inch diameter boom mount and upper joint between which it is mounted. A minimum pitch diameter of 10 inches will avoid interference. Any size larger than 10 inches will infringe on the trolley's inner radial motion, so the size should be minimized. With No. 60 rollerchain, a 60 tooth sprocket has a diameter of 14.76 inches. This will eliminate about 3 inches (14.76/2-9/2) of the trolley's motion. This is decided as the maximum acceptable infringement, setting 60 teeth as the maximum size of the large sprocket. The maximum speed ratio is thus 4:1, obtained by selecting 15 and 60 tooth sprockets. Ratios down to 1:1 are obviously also possible.

Motor and Gearhead Selection

With the allowable transmission ratios now known, selection of the motor can begin. The type of motor is the first design decision. A servo motor is certainly required. A servo motor is differentiated from a regular motor simply by its feedback control of position, velocity, or torque. Servo motors are available in both AC and DC versions. This differentiation only refers to the current applied to the motor. In both cases, the current supplied to the motor's drive, which converts this current for application to the motor, is 115 VAC single phase or 230 VAC single or triple phase. AC servo motors are mechanically simpler than their DC counterparts and thus tend to be more durable. They also tend to offer a higher efficiency. For these reasons AC servo motors will likely be the motor of choice in the future. But at the present time, AC servo motors in the 2 hp range are considerably more difficult to control. This difficulty leads to higher priced drives. So for economy and ease of control, the DC servo motor is selected.

There are two main types of DC servo motors: brushed and brushless. The difference is in commutation, the control of motor phase currents or voltages in order to optimize motor performance. Brushed motors use mechanically contacting brushes for commutation, while brushless motors employ electrical feedback. This lack of mechanical contact makes brushless motors more durable. Brushless motors also tend to be more adaptable than brushed motors. Thus the DC brushless servo motor is selected for this power assist application.

To determine the manufacturer of the motor, resources such as ThomasRegister.com and GlobalSpec.com are utilized. The latter allows searches based on numerous criteria. Other than motor type, parameters including continuous output power, size, optional equipment, and feedback type may be entered. Once the manufacturers which offer suitable DC brushless servo motors are found, in depth information is available at the company's website or through a literature request. A company by the name of Kollmorgen, which supplies the most comprehensive online documentation and customer support is tentatively decided upon.

The first step in selecting the actual DC servo motor is to find models with continuous power ratings around the continuous operating power calculated previously...

$$p_{no} = (271 lb \cdot ft)(31.5 rpm)\left(\frac{2\pi rad}{rev}\right)\left(\frac{1\min}{60\sec}\right)\left(\frac{1hp}{550\frac{lb \cdot ft}{s}}\right) = 1.63hp \tag{2.20}$$

The DC brushless servo motors found with a continuous power rating around 2 hp are observed to have maximum speeds from 1400 rpm to 6000 rpm and stall torques from 1.6 lb-ft to 4.6 lb-ft. Recollection that the maximum desired angular speed of the boom is 31.5 rpm and the maximum applied torque is 271 lb-ft produces the following overall transmission ratios:

$$\frac{1400 rpm}{31.5 rpm} = 44.4 \tag{2.21a}$$

$$\frac{6000 rpm}{31.5 rpm} = 190 \tag{2.21b}$$

$$\frac{271 ft \cdot lb}{1.6} = 170 \tag{2.21c}$$

$$\frac{271 ft \cdot lb}{4.6} = 58.9 \tag{2.21d}$$

Thus the rollerchain's maximum 4:1 ratio must be supplemented with a gearhead on the motor. If the maximum 4:1 rollerchain ratio is chosen, the gearhead must have a ratio from 11:1 (44.4/4) to 48:1 (190/4). Thus the motor selection is put on hold, while gearheads are investigated.

There are three main types of gearheads: spur, planetary, and harmonic. Spur and planetary gearheads are differentiated by their extensive use of spur and planetary gear sets, respectively. Spur gearheads, though, are generally viewed as poor choices for servo motors due to their large backlashes and poor efficiencies. Planetary gearheads, on the other hand, have small single stage backlashes of 3 arc minutes and double stage values of 10 arc minutes. Their efficiencies are also relatively high, at 90% for single stage and 85 % for double stage units. They offer single stage reduction ratios up to 10:1 and double stage ratios up to 100:1, right in the desired range. They are far and away the most commonly used gearheads for servo motors. Harmonic drives are used when larger reduction ratios, 80:1 to 320:1 are desired. All ratios provide less than 3 arc minutes of backlash, but have poor efficiencies ranging from 50 to 80 percent. Harmonic and planetary gearheads are similarly expensive. The planetary gearhead is thus chosen for its high efficiency, relatively low backlash, and ideal reduction ratios.

A brief discussion on the significance of backlash is now carried out. A two stage planetary gearhead has a maximum backlash of 10 arc minutes. Backlash value for a rollerchain transmission cannot be found, but are assumed low due to the numerous amount of teeth engaged in the chain at any one time. Thus the rollerchain's backlash is not considered here. The movement at the end of the 8 foot boom corresponding to 10 arc minutes of backlash at the gearhead output is...

$$g = \sin\left(10 arc\min \frac{1\deg}{60 arc\min}\frac{1}{4}\right) 8\,ft \frac{12 in}{ft} = 0.070 in$$

which is a very small value.

Gearheads can be purchased from many manufacturers, but it makes the most sense to buy them from the motor manufacturer. That way the gearhead is designed for the actual motor purchased and it can be attached easily. An adapter plate may have to be made for a different manufacturer's gearhead. So Kollmorgen gearheads are investigated. Since a gearhead ratio of at least 11:1 is needed, a double stage unit is deemed necessary. These units are available in ratios of 20:1, 50:1, and 100:1. Their minimum efficiencies, as stated earlier are 85%. The rollerchain transmission has an efficiency of about 90%. To determine the true power needed at the motor, the normal operating power must be divided by these two efficiencies...

$$p_{motor} = \frac{p_{no}}{h_{transmission}} = \frac{1.63 hp}{(0.85)(0.90)} = 2.13 hp \qquad (2.22)$$

Now, to actually select a motor, the smallest model with a continuous power exceeding 2.13 hp is found. This motor, model B-206-B/M-207-B has the torque curve shown in Figure 2.12. The solid curve is for operation on 230 VAC, while the lower dotted line is for 115 VAC operation. The vertical line at a torque of 4.6 lb-ft separates the continuous duty zone from the intermittent one. Multiplication of its maximum speed by its continuous stall torque gives the continuous

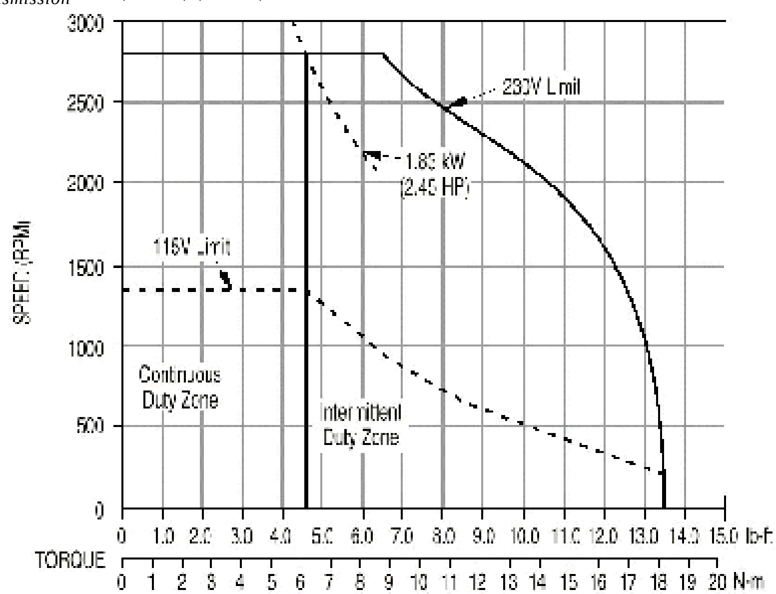


Figure 2.12 *Torque Curve for Selected Motor*

power rating of 2.45 hp. The stall torque, which is the amount of torque that can be developed when the rotor is locked in position, is the normal design parameter. Notice in Figure 2.12 that the stall torque of 4.6 lb-ft is available continuously at all rpms. This motor, the smallest sized model which supplies over 2.13 hp should provide the most economical choice, but several factors must be checked before it is chosen.

First, it is determined if an overall transmission ratio can be achieved which will supply the needed torque and speed. A 20:1 gearhead with a 4:1 rollerchain (60 and 15 tooth sprockets) transmission would supply the following values...

$$w_{boom} = \frac{1}{20} \cdot \frac{1}{4} \cdot 2800 rpm = 35 rpm \tag{2.23}$$

$$t_{boom} = \frac{20}{1} \cdot \frac{4}{1} \cdot 0.85 \cdot 0.90 \cdot 4.6 lb \cdot ft = 282 lb \cdot ft \tag{2.24}$$

which satisfy the design goals. A 50:1 gearhead with a 60:37 (60 and 37 tooth sprockets) rollerchain transmission would also work...

$$w_{boom} = \frac{1}{50} \cdot \frac{37}{60} \cdot 2800 rpm = 34.5 rpm \tag{2.25}$$

$$t_{boom} = \frac{50}{1} \cdot \frac{60}{37} \cdot 0.85 \cdot 0.90 \cdot 4.6 lb \cdot ft = 285 lb \cdot ft \tag{2.26}$$

But further investigation into the gearheads reveals that the 50:1 model is incapable of handling the motor's continuous torque. The amplifier must be compensated to limit the motor's performance. This compensation will deliver a torque of only 178 lb-ft to the boom. The continuous torque and maximum speed ratings on the 20:1 gearhead do not limit the motor's performance, meaning 35 rpm and 282 lb-ft of torque are attainable.

With the motor and gearhead selected, the next part of the servo system needed is the drive. It contains the power and control electronics. Like the gearhead, it is easiest to order the drive from the same manufacturer as the servo motor. Interface with the motor and feedback device are simple. Extensive control capabilities and safety devices are included. There is one drive for the selected motor and its capabilities match those of the motor. The drive is discussed further in a later section.

The final part of the system is the feedback device. Two types are available: resolver and encoder. A resolver acts like a rotating transformer to provide an analog signal with position and velocity information. An encoder uses an optical source and sensor to provide a digital signal with position and velocity information. A resolver tends to be more robust, but an encoder is decided on since the author's laboratory has more experience with this type of feedback device.

The size of the motor and gearhead is checked. The width of the square cross-sectioned gearhead and servo motor are both about four inches. Thus they will not complicate the mounting to the 5 inch wide mast.

Inertia Matching

Inertia matching is a process which optimizes a motor's acceleration capabilities. The goal is to get the motor's rotor inertia multiplied by the square of the overall transmission ratio as close as possible to the inertia of the driven load...

$$\frac{I_{load}}{I_{rotor} \cdot transmission\_ratio^2} \approx 1 \tag{2.27}$$

The gearhead should be included in the analysis, but this is made virtually impossible by numerous gears with various moments of inertia rotating with various angular speeds. Thus it is neglected. Foresight, with knowledge of the load's very high inertia, suggests that the procedure may not be successful.

The motor chosen comes in two versions. The B-206-B has a low rotor inertia of 1.85×10^{-4} slug-ft^2 The M-207-B has a large rotor inertia of 1.3×10^{-3} slug-ft^2. Knowing that the load's inertia is very large, the latter motor is considered. Recall the load's moment of inertia...

$$I_{total} = \left(39.8 + r^2\left(0.171 + \frac{W}{32.2}\right)\right) slug \cdot ft^2 \tag{2.28}$$

Insertion of the average radial distance of 4.5 feet and the average weight of 100 lb yields...

$$I_{total} = \left(39.8 + (4.5\,ft)^2\left(0.171 + \frac{100}{32.2}\right)\right) = 106 slug \cdot ft^2 \tag{2.29}$$

The inertia matching ratio becomes...

$$\frac{106 slug \cdot ft^2}{\left(1.3x10^{-3}\, slug \cdot ft^2\right)\left(\frac{80}{1}\right)^2} = 12.7 \tag{2.30}$$

Keep in mind that the inertia of the load changes with radial position and lifted weight. Repetition of the above calculation shows the ratio to vary from 5:1 to 54:1. Increasing the transmission ratio improves the inertia matching ratio dramatically, but any increase in the ratio will not allow the design velocity to be achieved. Though the values are far from ideal, the velocity goal is deemed more important than inertia matching. The motor will simply not operate optimally with respect to load acceleration.

Regeneration

The final topic addressed in the servo system sizing is the often overlooked need assessment for regeneration. The rotating jib crane has a lot of kinetic energy. When the motor decelerates the crane, this energy is pumped back to the drive. The bus module capacitors within the drive are capable of storing a certain amount of energy, but if the amount of energy sent back to the drive exceeds this capacity, the remaining energy must be dissipated by a regeneration resistor. If a regeneration resistor is not supplied when needed, the results will range from blown fuses or tripped relays to damaged bus capacitors and drive transistors. As is common practice, the electrical quantities are measured in SI units. The value 1.356/2 appears in place of the expected ½ for unit conversion. The procedure is from [2].

The amount of energy that the bus module can store is...

$$E_{BUS} = \frac{1}{2} C\left(V_{\max}^2 - V_{nom}^2\right) \tag{2.31}$$

where: C = bus module capacitance (F)
V_{max} = maximum bus voltage (V)
V_{nom} = nominal bus voltage (V)

Substituting supplied values...

$$E_{BUS} = \frac{1}{2}(0.00164F)\left[(390V)^2 - (325V)^2\right] = 38.1J \tag{2.32}$$

The amount of kinetic energy in the system minus system losses is...

$$E_m = \frac{1.356}{2} J_m w_m^2 + \frac{1.356}{2} J_l w_l^2 - \frac{3}{2} I_m^2 R_m t_d - \frac{1.356}{2} T_f w_m t_d \tag{2.33}$$

where: J_m = motor rotor inertia (slug-ft^2)
J_l = load inertia (slug-ft^2)
ω_m = motor speed before deceleration (rad/s)
ω_l = load speed before deceleration (rad/s)
I_m = motor current during deceleration (A)
R_m = motor resistance (Ω)
t_d = deceleration time (s)
T_f = friction torque (lb-ft)

The jib crane pivot joint exhibits very little resistance. Thus to be conservative, the friction torque is assumed zero and the last term drops out. Due to the presence of the squared angular speed term, the kinetic energy of the load is greatest when its speed is greatest. This occurs when the trolley is at its minimum radial position. Thus the speed of 31.5 rpm and corresponding moment of inertia of 46.2 slug-ft^2 are used. To determine the deceleration time, assume the boom is decelerated from its top desired speed at the maximum desired rate...

$$a = \frac{a_{\max,desired}}{r_{\min}} = \frac{4.8\,ft/s^2}{1\,ft} = 4.8\,rad/s^2 \tag{2.34}$$

$$t_{dec} = \frac{w}{a} = \frac{31.5rpm\dfrac{rad/s}{9.55rpm}}{4.8rad/s^2} = 0.687\,s \tag{2.35}$$

Substitution of these values gives...

$$E_m = \frac{1.356}{2}\left(0.0013slug\cdot ft^2\right)\left(31.5rpm\frac{80}{1}\frac{1rad/s}{9.55rpm}\right)^2 + \frac{1.356}{2}\left(46.2slug\cdot ft^2\right)\left(31.5rpm\frac{1rad/s}{9.55rpm}\right)^2 - \frac{3}{2}(6.0A)^2(2.32\Omega)(0.687\,s) = 61.4J + 340.8J - 86.1J = 316.1J \tag{2.36}$$

Because this energy is greater than the storage capacity of the bus module capacitors, a regeneration resistor is needed. There are two steps to select an appropriate resistor. The first step is to calculate the maximum resistance which will keep the bus module's voltage under its maximum...

$$R_{\max} = \frac{V_m^2}{V_B I_M \sqrt{3}} \tag{2.37}$$

where: V_b = motor back EMF less motor losses (V)

and

$$V_b = K_b N - \sqrt{3} I_m \frac{R_m}{2} \tag{2.38}$$

where: K_b = back EMF constant (V/krpm)
N = motor speed prior to deceleration

Substitution of values gives...

$$R_{\max} = \frac{(390V)^2}{\left(69.0V/krpm\cdot 2.52krpm - \sqrt{3}\cdot 6.0A\dfrac{2.32\Omega}{2}\right)6.0A\sqrt{3}} = 90.4\Omega \tag{2.39}$$

Thus the resistance of the regeneration resistor must be less than or equal to 90.4 S.

The second and final step in sizing a regeneration resistor is to determine the necessary wattage rating. This value is given by the following equation...

$$P_{avg} = \frac{E_m - \frac{1}{2}C\left(V_m^2 - V_{hys}^2\right)}{t_{cycle}} \tag{2.40}$$

where: P_{avg} = average power that must be dissipated (W)
V_{hys} = hysteresis point–point at which regeneration. circuit opens
t_{cycle} = time to decelerate plus time between decelerations

The cycle time is found by assuming the following worst case scenario. With the trolley at the innermost position, the boom is accelerated at the maximum desired rate until the top speed is reached. Then it is immediately decelerated at the maximum desired rate. The process then immediately repeats itself. This would result in a cycle time of 1.37 seconds, twice that calculated in Equation 2.35. The average power dissipated becomes...

$$P_{avg} = \frac{316.1J - \frac{1}{2}0.00164F\left[(390V)^2 - (370V)^2\right]}{1.37s} = 222W \tag{2.41}$$

Thus the regeneration resistor should have a power rating of 222 W and a resistance less than or equal to 90.4 S. But the largest regeneration resistor offered by Kollmorgen has a power rating of 200 W. Realize that adding only 0.15 seconds to the cycle time would make the 200 W rating acceptable. A human delay time of 0.15 seconds could easily be justified in the shift from maximum acceleration to maximum deceleration. Thus the 200 W regeneration resistor, with an acceptable resistance of only 20 S is selected.

Continuation of Rollerchain Transmission Design

With the servo motor, gearhead, and drive chosen, the remaining portion of the rollerchain transmission design is addressed. No. 60 sprockets of 60 and 15 teeth have been decided upon. It is verified that the 15 tooth sprocket can be bored out to accommodate the 32 mm shaft and 3mm. x 10 mm keyway of the servo motor. The metric system is reverted to here since the motor is designed with it and the diameter of the bored out sprocket must be extremely accurate. The small sprocket is specified with hardened teeth, as recommended for any sprocket with less than 25 teeth utilized in a heavily loaded system. Note that the 4:1 reduction ratio is less than the maximum recommended ratio of 7:1.

Rollerchain Verification

With the motor, gearhead, and rollerchain specifications now known, a second rollerchain selection process is carried out to verify the correct selection. Called a chain-pull analysis, it is often used on low speed transmissions. To be conservative, the peak stall torque is used in place of the continuous stall torque. This is the maximum torque that could possibly be transmitted to the boom. The motor's peak stall torque, which can be verified by viewing Figure 2.12, is 13.6 lb-ft. This corresponds to a torque at the small sprocket of...

$$t_{small\_spricket} = (0.85)\left(\frac{20}{1}\right)(13.6lb\cdot ft) = 231lb\cdot ft \tag{2.42}$$

Division of this torque by the small sprocket's pitch radius yields...

$$F_{chain} = \frac{231lb\cdot ft}{3.608\big/2\, in\left(\frac{1ft}{12in}\right)} = 1537lb \tag{2.43}$$

The average tensile strength of standard No. 60 rollerchain is 8500 lb. It is recommended that a safety factor of 6 be used. This would imply the following chain tensile strength is required...

$$S = 6\cdot 1537lb = 9222lb \tag{2.44}$$

Thus it is decided to choose a high strength version of No. 60 rollerchain which has a tensile strength of 10,000 lb.

Center Distance

The next step is determination of the center distance of the rollerchain drive. This is the distance between the sprocket centers. To avoid interference, this distance must be greater than the sum of the sprockets' radii, but to ensure proper chain wrap around the small sprocket, this distance must be significantly greater. It is recommended that for reduction ratios greater than 3:1, the center distance should be greater than the sum of the sprocket diameters...

$$d_c \geq \sum sprocket\_diameters = 3.98in + 14.76in = 18.74in \tag{2.45}$$

Division by the chain's pitch of 0.75 in. and upward rounding gives the minimum center distance as 25 pitch lengths. This value is needed in the following equation to determine chain length...

$$L = 2C + \frac{N+n}{2} + \frac{0.1013(N-n)^2}{4C} \tag{2.46}$$

where: L = chain length in pitches
C = center distance in pitches
N = # of teeth in large sprocket
n = # of teeth in small sprocket

Substitution of values gives...

$$L = 2 \cdot 25 + \frac{60+15}{2} + \frac{0.1013(60-15)^2}{4 \cdot 25} = 89.5 pitches \tag{2.47}$$

If the number of links is an odd number, an offset link, which reduces the load capacity of the chain by 30%, must be used. Thus, an even 90 pitch chain length is chosen. Due to the necessary rounding to a whole number of pitches, the new center distance must be calculated...

$$90 = 2C + \frac{60+15}{2} + 0.1013\frac{(60-15)^2}{4C} \tag{2.48}$$

Multiplying through by C and bringing everything to the same side yields...

$$2C^2 - 52.5C + 51.283 = 0 \tag{2.49}$$

Solving this quadratic equation gives a C value of 25.234 pitches, which corresponds to a center distance of...

$$d_c = 25.234 pitches\left(\frac{0.75in}{pitch}\right) = 18.925in \tag{2.50}$$

Motor Mount

With the motor, gearhead and rollerchain transmission now fully specified, the motor mount is designed. The design, shown in Figure 2.13 incorporates adjustability, accurate initial orientation, and good resistance to flexure. All parts are made out of aluminum for ease of machining. The motor attaches to the yellow plate, which slides in horizontal slots in the two diagonal arms. A movement of 3/4 inch, the pitch of the chain, is critical for two reasons. First, a mid-span movement of 2-3% of the distance spanned between sprockets is recommended for reversing rollerchain transmissions (the chain cannot be perfectly taught because of chordal action). This mid-span movement varies greatly with even small changes in center distances. Thus, adjustability is required to achieve the desired mid-span movement. Second, rollerchains stretch over time. Adjustability is required to account for this deformation of the chain. A movement of 3/4 inch, combined with the ability to remove pairs of 3/4 inch links, allows for infinite adjustability. Keep in mind that since the chain is at a slight angle to the motor plate's direction of travel, removal of two links will shorten the center distance by less than 3/4 inch. Four set screws in the diagonal arms fix the position of the motor plate. Bolts in the end of the plate prevent it from slipping out of the slots.

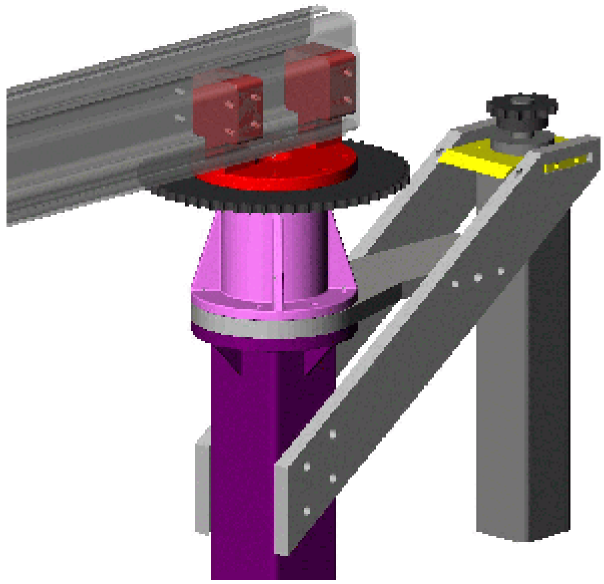


Figure 2.13 *Motor Mount*

The accurate orientation of the sprockets relative to one another is critical to prevent the unnecessary application of loads to the chain and to achieve the maximum efficiency of the transmission. Accurate machining of the slots in the diagonal arms is the first step. Second, a horizontal reference arm is inserted between the purple mast and pink lower pivot. It is simply attached through the six existing holes in the pivot and mast. This arm helps the orientation by supplying a surface guaranteed parallel to the large sprocket. Dowel pins accurately position the diagonal arms and attached motor plate relative to the horizontal reference arm. The attachment of the diagonal arms to the mast is simply for structural reasons, not for positioning.

Finally, the triangular structure formed by the mast, horizontal arm, and diagonal arms makes the motor mount extremely stiff. An accurate deflection analysis is not carried out, but simple experimental tests, involving forces similar to those generated by a maximum torque, generate no apparent deflection. The forces applied to the small sprocket by the chain under maximum continuous torque are found from a chain pull analysis and geometry of the transmission and are shown in Figure 2.14. Rough calculations and large safety factors are used to size the motor mount components and bolts.

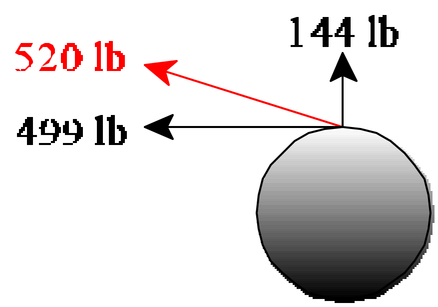


Figure 2.14 *Forces on Small Sprocket from Chain*

All components of the motor mount are made by the author via a combination of manual and computer numerically controlled machining. The collage of pictures in Figure 2.15 shows a few of the stages. The finished product, attached to the jib crane, is depicted in Figure 2.16

Figure 2.15 Collage of Pictures taken during Motor Mount Machining

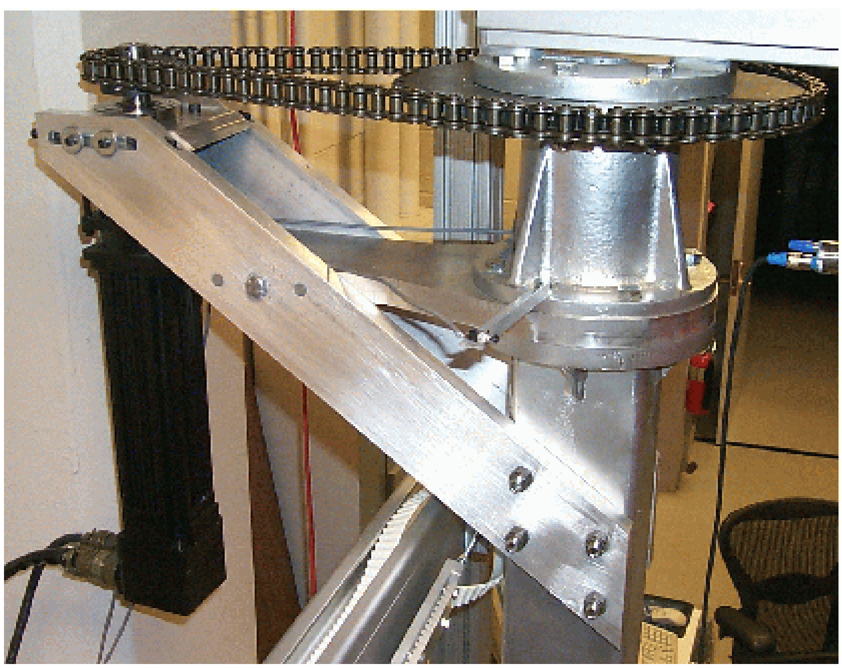


Figure 2.16 *Motor Mount, Servo Motor, and Rollerchain Transmission*

Electrical Enclosure

The drive and regeneration resistor are placed inside a sheet metal enclosure to protect people from the high voltages, to provide a suitable platform for grounding, and to protect the components from damage. Figure 2.18 shows the enclosure with all hardware mounted. The white panel in the back attaches to the enclosure and serves as a mount for the components. The description starts with the terminal block on the lower left part of the drive's front panel and progresses from bottom to top. The black wire attaches one of the drive's ground terminals to the panel's central grounding point. The brown and blue wires attach to the 40 A circuit breaker mounted to the side of the enclosure. A 115 VAC plug comes off the other end of the breaker and plugs into a standard outlet. The ground for this plug attaches to the back panel's ground. The two white wires attach the regeneration resistor to the appropriate terminals. The black, white, and blue wires are the three phases of high voltage current send to the motor. Finally, the top black wire is another grounding point.

Now progressing from top to bottom on the right side of the drive, the description continues. The LED displays system status information. The first cable attaches to the serial port of a computer. All system variables can be set and the drive controlled via this cable either through the supplied Windows software or through a dumb terminal. The plug below this cable is the motor feedback cable, supplying encoder information. A cable for the static holding brake, which can be used dynamically in emergencies, also runs through this cable. 24V DC must be applied to the two wires to release the brake. The next terminal block contains the

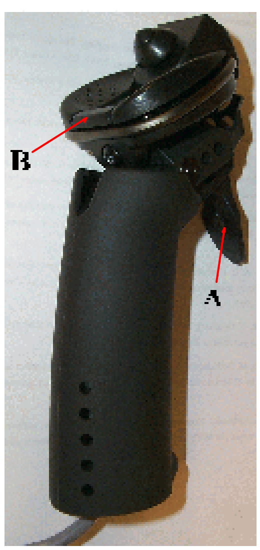


Figure 2.17 *Dead Man Switch*

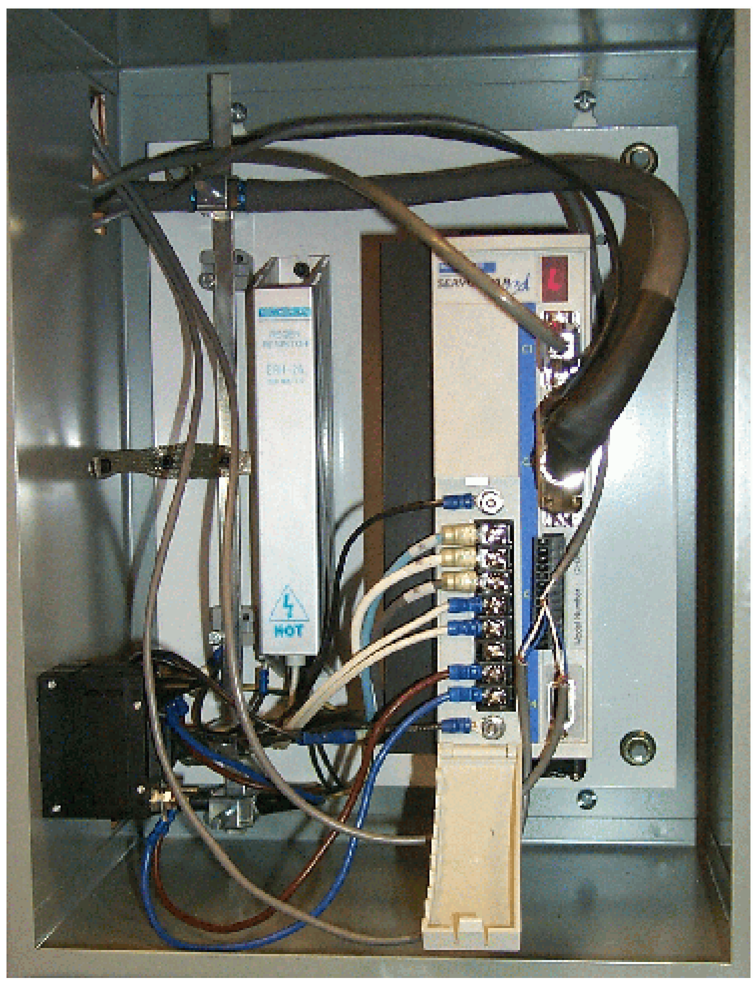


Figure 2.18 *Electrical Enclosure Housing Drive and Regeneration Resistor*

configurable inputs and outputs for the drive. The position, velocity, or torque control variable can be read from one of these analog inputs. An enable circuit, which must have 24 VDC applied across it is also located here. Two of the inputs are configured for clockwise and counterclockwise limit switches. The switches are normally closed and have a 24 VDC potential applied across them. The enabling current is run through switch A in the dead man's switch depicted in Figure 2.17. Both of the switches in the joystick are normally open. The current for the holding brake runs through switch B. If either of the limit switches is opened or switch A is opened (trigger released), even momentarily, the drive disables and decelerates the load to a stop using the maximum continuous torque. In an emergency situation, switch B can also be opened to provide an even faster stop.

The thin bar attached to the left side of the back panel is called a BUS bar. The power cable, motor current cable and motor feedback cable have their outer sheath removed at the point where they cross this bar and have their shields attached to the bar via clamps. At the center of the bar, a flat braid of copper electrically attaches the bar to the back panel and enclosure. The purpose of this system is to provide a high frequency ground. Normal wire grounding will not provide a sufficient ground for these high frequencies. Finally, care is taken to separate the signal cables from the high voltage power cables. The former are run out the top left of the enclosure, while the latter leave the box at the bottom left.

Sensors to Add Intelligence

The servo motor, transmission, and drive give the jib crane the capability of power assist in the tangential direction. Now sensors are added to supply the intelligence which will give the passive trolley system the potential for isotropic behavior. To know the torque that should be applied to the boom, two variables must be known: the intent of the operator and the value of the variable moment of inertia.

Cable Angle Sensor

To sense the intent of the operator, the angle of the cable suspending the load will be measured. The cable angle sensor and enclosure are depicted in Figure 2.19. Two blue arms extend down from the green Zimmerman trolley. They attach to a gray enclosure shown translucent in the main drawing on the right and opaque in the rotated view on the left of Figure 2.19. The two green plates represent the cable angle sensor circuit boards. The top board contains the sensor's circuitry, while the lower board contains four sets of coiled inductors. A current from the top board runs through the suspending cable. If the cable is tilted from its vertical orientation, a voltage is induced in the coiled inductors. The top board collects this

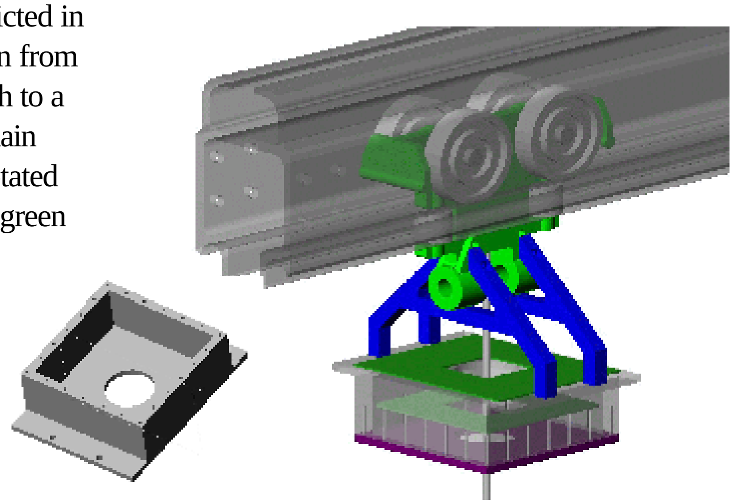


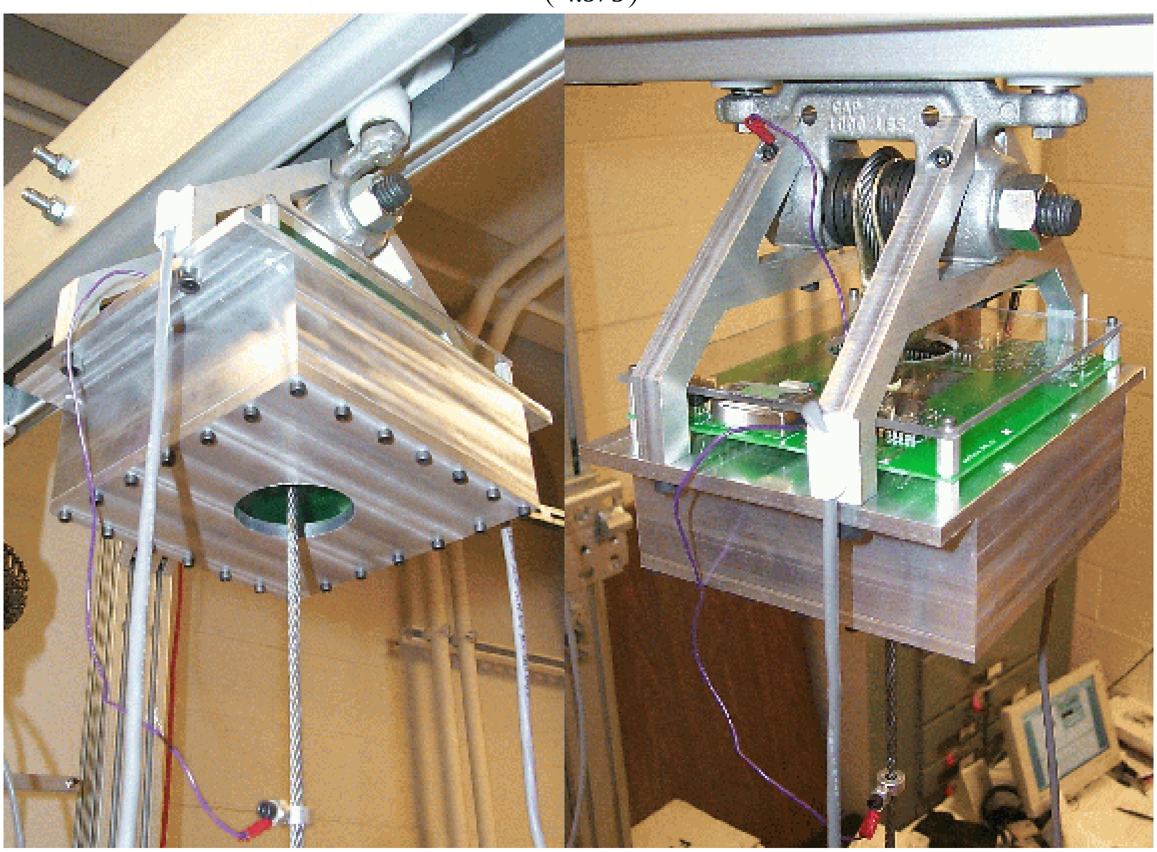
Figure 2.19 *Cable Angle Sensor and Enclosure*

signal and outputs a voltage directly proportional to the cable's angle. The purple top is secured tightly to the enclosure with 24 screws to prevent flux leakage from the coiled inductors. Further information on this sensor can be found in [3]

As with the motor mount, the cable angle sensor bracket and enclosure components are CNC machined by the author The finished product is shown in Figure 2.20a from below and Figure 2.20b from above. In the latter figure, notice the excitation current (purple wire) attachment points directly to the cable below and to the frame above. A Plexiglas shield is inserted above the circuit board to protect it from possible damage. One electrical cable houses the wires for the 24 VDC used to power the sensor and the other contains the four wires carrying the two axis sensor's output. Figure 2.20b also shows the combination thimble and clamp used to attach the high strength hoist cable to the trolley's bolt. While the trolley and cable have a capacity of 1000 lb, the thimble reduces this to 600 lb, still well above the jib crane's 200 lb limit. Washers are used to ensure the thimble does not slide along the bolt.

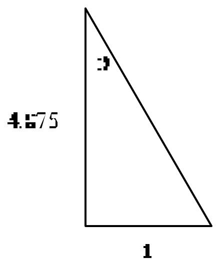
The cutouts in the enclosure are 2 in diameter holes, while those in the circuit boards are squares with a 2 in side. Leaving space for the low profile thimble and clamp, the enclosure is mounted as close to the trolley as possible. This maximizes the allowable cable angle before the cover is hit. Illustrated in Figure 2.21, the maximum allowed angle from vertical is...

$$q = \tan^{-1}\left(\frac{1}{4.875}\right) = 11.6^{\circ} \tag{2.51}$$



a b

Figure 2.20 *Cable Angle Sensor and Fabricated Enclosure*



With a 200 lb load, this corresponds to a maximum push of...

$$P_{\max} = W\tan\theta = 200lb\tan\left(11.6^{\circ}\right) = 41.0lb \tag{2.52}$$

.

which should be more than adequate.

Figure 2.21
Cable Angle Calculation

Radial Position Sensor

The cable angle provides the intent of the operator. To determine the appropriate torque to apply to the boom, the moment of inertia is also needed. Recalling Equation 2.12, this moment of inertia varies with radial trolley position. Conversion from desired linear acceleration to boom angular acceleration also requires the radial position. Thus a sensor must be added to measure the trolley's radial position.

The desired attributes of the radial position sensor are lack of interference with the trolley's motion, low cost, and robustness. Accuracy of less than a tenth of an inch is deemed unnecessary. Three types of position sensors are considered: reeled cord, laser, and ultrasonic.

A reeled cord position sensor physically attaches a cord to the device being measured. The device is both inexpensive and robust. However, like on a tape measure, a retractile force is necessary to draw the cord back into the reel. Thus, this type of sensor applies a force, which generally increases with measured distance, to the measured object. Because of the very low rolling friction in the radial direction, this type of sensor is deemed unacceptable.

Laser position sensors send out a narrow pulse of light and determine the distance to the reflecting object by measuring the time for the pulse to return. The motion of the trolley is thus not interfered with. The measured object must reflect light, but the shiny metallic surface of the cable angle sensor enclosure is perfect. Laser sensors are also robust, but at a price of a few thousand dollars, are judged too expensive.

Finally, the ultrasonic sensor is similar to its laser counterpart, except is sends out a sound wave instead of a light wave. This change in technology allows a more reasonable price of $200 to $500. Like the laser sensor, an ultrasonic sensor does not interfere with the motion of the measured object. Two possible problems exist in the area of robustness. The first deals with the variation in the speed of sound. In contrast to light, sound needs a medium to travel in and the speed of propagation varies with the medium's (air's) temperature. This potential problem is solved by selecting an ultrasonic sensor with temperature compensation. The second potential problem is that air forced through a nozzle creates a whistling sound with harmonics in the ultrasonic range [4]. This could be a problem with the prevalence of pneumatic devices, such as hoists, in material handling environments. The effect is minimized by selecting an ultrasonic sensor which emits a high frequency (low wavelength) sound wave. Thus the two uncertainties in the area of robustness can be dealt with and the ultrasonic sensor is chosen to measure the trolley's radial position.

Numerous manufacturers of ultrasonic sensors are found at ThomasRegister.com. Every sensor investigated has a beam angle of 8 degrees or more. This presents a problem. Referring to Figure 2.22, the beam angle is the total included angle of the conical pattern formed by the emitted sound wave. The angle of the beam in Figure 2.22 is 5 degrees and produces a circular pattern on the target with the following diameter...

$$d = 2 \cdot target\_distance \cdot tan\left(\frac{beam\_angle}{2}\right) = 2 \cdot 67in \cdot \tan\left(\frac{5^\circ}{2}\right) = 5.85in \tag{2.53}$$

A larger beam angle would produce a conical pattern which could include other objects, particularly, the boom. If any of these objects includes a surface, even a small nick, perpendicular to the sound

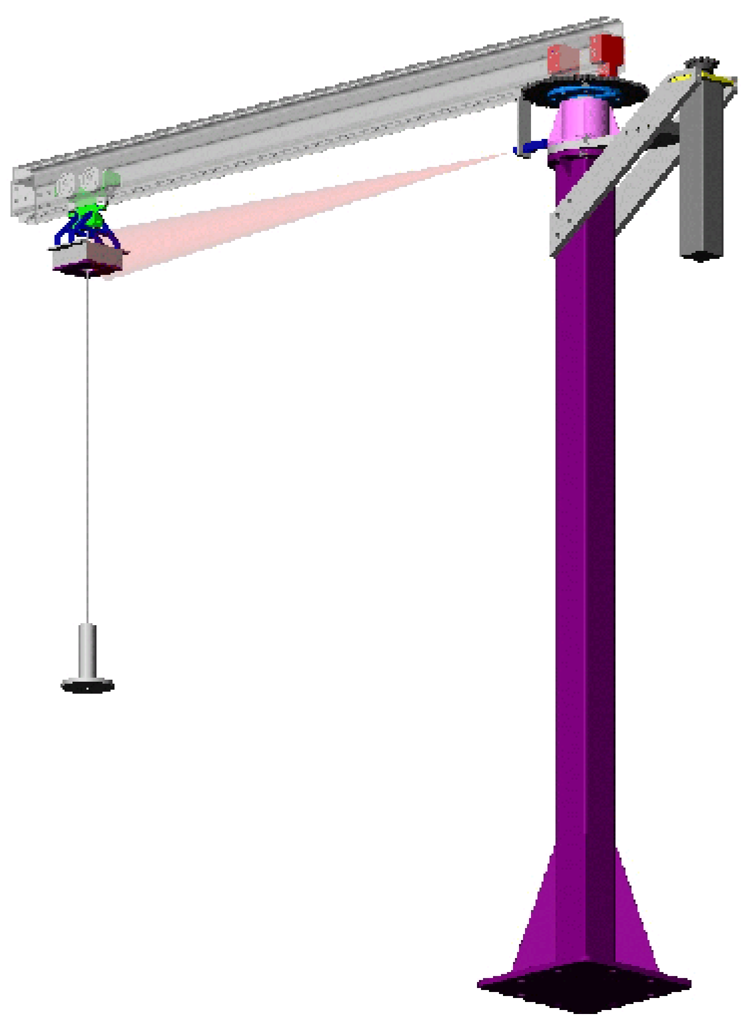


Figure 2.22 *Conical Beam Pattern of Emitted Sound Wave*

wave, it is the distance to this false target which will be measured. Thus a beam angle of 5 degrees is desired. The sensing range must be around zero to 75 in. An extensive search finds an ultrasonic sensor made by Honeywell which has a beam angle of 5 degrees and a sensing range from 7.87 in to 78.7 in. Though it is at the upper end of the price range, it is chosen for these optimal characteristics. By selecting the smallest beam angle available, another problem is also solved. The small beam angle is obtained by using a higher than normal frequency (low wavelength) sound wave. As mentioned earlier, this is the type of sound wave which will experience the least amount of interference from sound waves created by pneumatic devices.

10 to 30 VDC powers the sensor and the output can be read as either a 4-20 mA or a 0-10 V signal. The resolution of the device, 0.017 in, is more than adequate for this application.

The most convenient place to mount the ultrasonic sensor is directly under the boom. At this position, the cable angle sensor enclosure presents itself as an excellent target. The ultrasonic sensor is shown mounted in Figure 2.23. The casing is simply a threaded cylinder. The mount is designed to ensure the sensor points directly at the cable angle sensor enclosure. Figure 2.24 depicts the sensor's mount. Accurate machining of surfaces A and B and the fact that the boom is extruded, ensure that the sensor points in the desired direction.

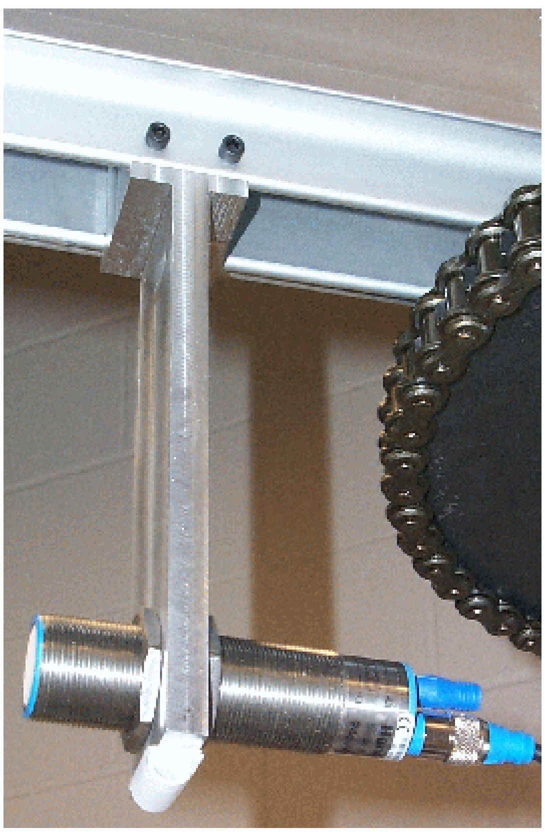


Figure 2.23 *Ultrasonic Sensor Attached to Boom*

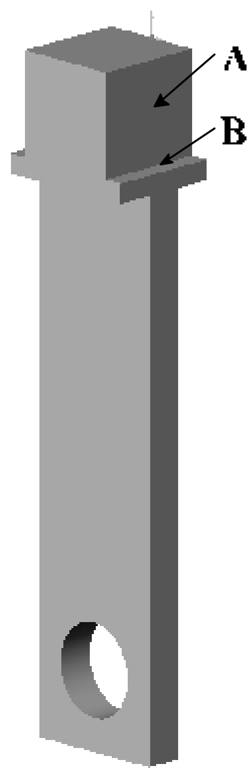


Figure 2.24 *Ultrasonic Sensor Mount*

The final design consideration is the mounting of the CW and CCW limit switches. Recall that these switches are normally closed. Even momentary contact with either switch disables the drive, decelerating the load to a stop as quickly as possible. The ultrasonic sensor mount is viewed as the best contact element. As shown in Figure 2.25, the two switches are mounted in aluminum right angle parts at a position where the ultrasonic sensor mount will contact them. If they are pushed, they will not break because they will rotate about their mounting bolts on the lower pivot. Contact with these switches will only occur in an emergency situation.

Figure 2.25 *CW and CCW Limit Switches*

The design of the platform for an intelligent power assist jib crane is thus completed. The finished device is shown in Figure 2.26. Note the weight holding system suspended from the cable via a hook. It is designed to hold up to 200 lb in Olympic free weights. Also note how the two electrical cables from the cable angle sensor are suspended from the ultrasonic sensor mount. They must be draped in this manner to avoid interference with the sound wave's conical beam. Stops placed at the free end and just in front of the ultrasonic sensor mount prevent the trolley from traveling too far. Finally note that the mast is mounted to the concrete floor with 8 3/4 in bolts screwed into drop-in anchors.

Figure 2.26 *Completed Platform for an Intelligent Power Assist Jib Crane*

3 Experimental Results

With the platform built, proper operation of the mechanical and electrical components is verified. First, with the enable and holding brake disable buttons depressed, a step velocity command is sent to the drive from a computer via the serial port. The step commands the motor angular speed to 100 rpm for 5 seconds. The recorded response is shown in Figure 2.27. The high frequency oscillation in the current corresponds to an oscillation in the actual velocity. Note that the amplitude of these high frequency current inputs dies out quickly as the load reaches the commanded speed. The low frequency oscillation (a peek occurs at 3 seconds) seen in the current is believed due to swinging of the 50 lb load.

The commanded motor speed of 100 rpm is only 3.6% of its 2800 rpm maximum speed. Higher velocity commands are attempted, but step commands over 100 rpm result in large impact forces on the trolley. There is significant play in the trolley's tangential position within the boom. The problem with a velocity step input is the immediate onset of large torques. To get a smoother response, a different input mode is attempted.

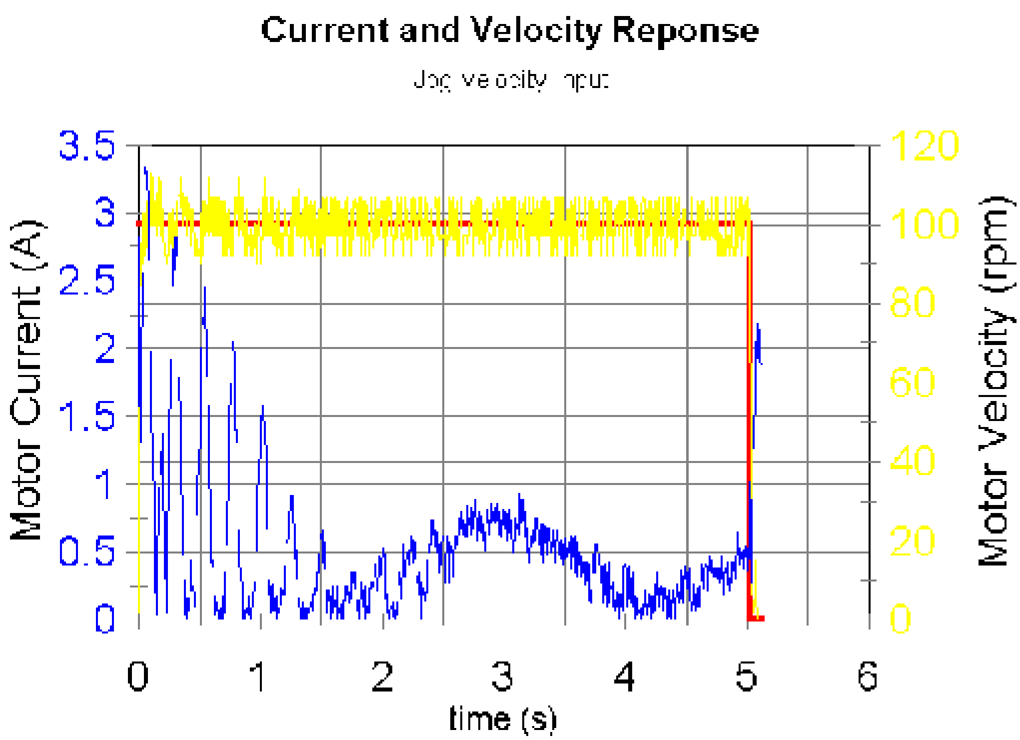


Figure 2.27 *Motor Current and Motor Angular Speed in Response to a Velocity Step Input of 100 RPM*

A step torque command of 0.72 A, or about 1/9 of the motor's continuous torque is applied to the motor for five seconds. This is the maximum torque which will likely be applied in the passive trolley isotropic application. The response is shown in Figure 2.28. The response exhibits the same low frequency oscillation found in Figure 2.27. This is again attributed to swinging of the load. The observed response to the torque command is a very smooth acceleration, as expected. The servo motor is very quiet and the disabling dead man switch works very well. Release of the enabling trigger brings the boom to a stop quickly. Release of the holding brake button brings the boom to a stop almost instantaneously. The correct operation of both limit switches is observed at low speed. Backlash results in about 1/4 inch of movement at the end of the 8 ft boom. The gearhead's 10 arcmin of backlash must be supplemented by 0.1 degrees of rollerchain backlash. The servo motor, gearhead, transmission and drive appear to work as designed. The tremendous power potential, which would be applied in the powered trolley isotropic application, is evident from these low power tests.

The software used to transmit the velocity and torque commands is provided by the motor manufacturer. The interface is very intuitive and allows control of the entire several hundred variable list. Numerous operations, including data acquisition are made possible by the application As mentioned, the drive and motor can also be controlled through a dumb terminal, where knowledge of the entire variable set is required.

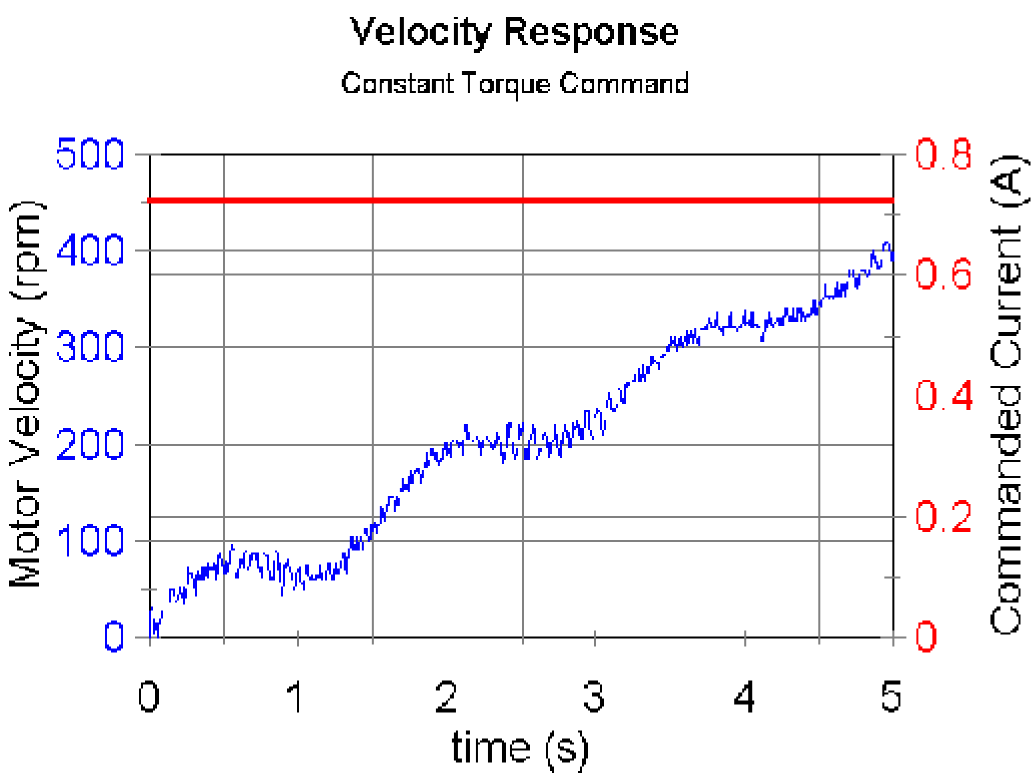


Figure 2.28 *Velocity Response to a Constant Torque (Current) Command*

With correct operation of the motor, drive, and transmission verified, the cable angle sensor is tested. A two channel data acquisition module is attached to the four signal wires and 24 VDC to the power cable. With the rollerchain removed, a 50 lb load is suspended from the trolley and the following motion carried out. The load is grasped by the operator, moved in the radial direction, stabilized, then moved in the tangential direction and stabilized. In Figure 2.29, the output of the radial angle sensor is shown in red, while that of the tangential angle sensor is shown in blue. Both curves show the expected oscillatory response. The greater amplitude of the tangential curve is indicative of the larger force necessary for motion in this direction. Also, notice that the tangential curve dies out very quickly, showing evidence of the pivot's damping effect. In contrast, the radial curve exhibits much oscillation due to the very low rolling resistance in the trolley. The measured tangential angle from 1.5 to 2.5 seconds is simply showing the inability of the operator to apply radial motion only. The same can be said of some of the random radial motion seen from 3 to 4 seconds. The non zero offsets at the end are caused by the operator stabilizing the load at a non zero angle position. All evidence shows that the cable angle sensor works well.

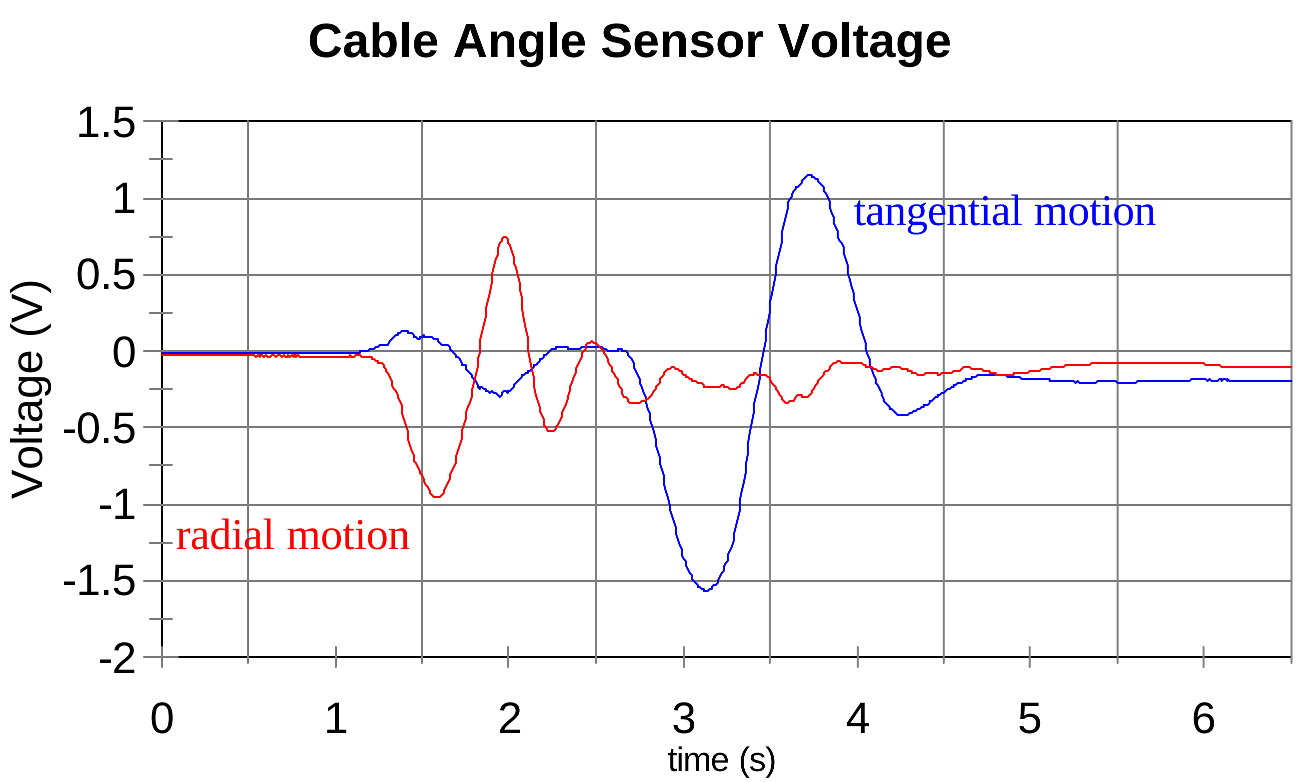


Figure 2.29 *Cable Angle Sensor Response to Jib Crane Motion* The above curve represents a motion in the radial direction, which is then stabilized, followed by motion in the tangential direction and stabilization.

The final verification involves the ultrasonic sensor. 24 VDC is hooked up to its power leads and the voltage output is connected to the data acquisition system. Data is collected as the trolley, initially at the maximum radial position, is moved radially inward at an approximately constant speed. The stop is contacted fairly hard, motion is paused for one second and then the trolley is moved to a middle radial position and held. The collected data is shown in Figure 2.30. The sensor appears extremely linear, as desired. This sensor also appears to function well.

Thus the mechanical and electrical components of the platform function as desired. The thesis now moves toward conclusion.

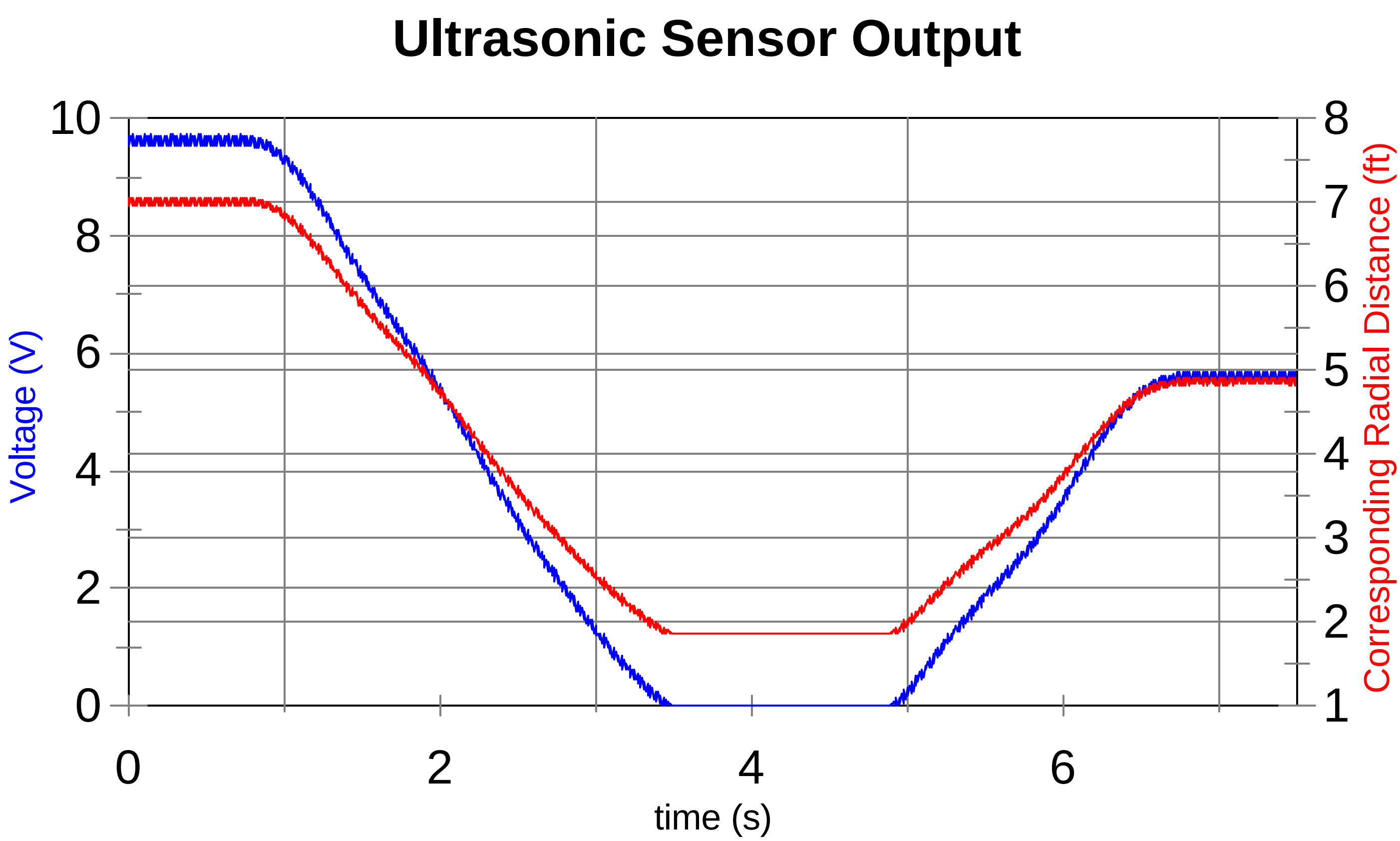


Figure 2.30 *Voltage Output and Calculated Proportional Radial Distance from Ultrasonic Sensor*

4 Conclusion

The design and construction of a platform for an intelligent power assist jib crane is completed. The brushless DC servo motor, planetary gearhead, motor drive, and rollerchain transmission are capable of tangentially accelerating a 200 lb load at 0.15 g's up to a maximum speed of 3.3 ft/s. Limit switches and a dead man's joystick to disable the motor and/or apply the emergency stopping brake bring safety to this potentially dangerous material handling system. A sensor mounted to the trolley measures the angle of the suspending cable. The radial position of the trolley is measured by an ultrasonic sensor. The summation of these components provides a platform capable of giving the jib crane an isotropic feel throughout its workspace.

Future Work

Work left to be done includes first the application of a control system to make the jib crane with a passive trolley exhibit isotropic behavior. A possible control logic is described. First, when the operator pushes on the load, the angle of the cable is read. With the mass of the load known, the force applied to the mass can be determined. With the mass and operator applied force now known, the intended linear acceleration of the load can be found from Newton's second law. The measured radial position of the trolley can then be used to convert this desired linear acceleration to an angular acceleration of the boom. Knowing the moment of inertia of the upper pivot, boom mount, and boom, along with that of the trolley (which is a function of the radial position) allows determination of the torque that should be applied by the motor. An attempt in the control system to try to minimize the cable angle would likely result in a stable system robust to cable swing.

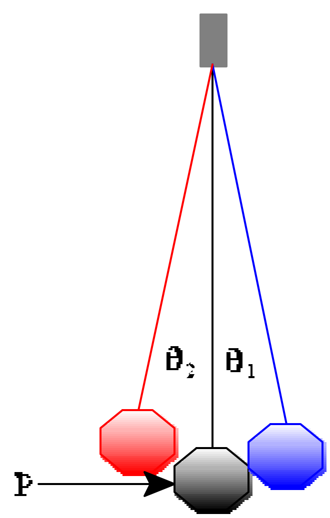


Figure 2.31 *Diagram Showing Inability of Cable Angle Sensor to Measure Intent when Load must be Accelerated*

Another area of future work would be an attempt to create isotropic behavior with a powered trolley. With both degrees of freedom powered, the inertia of the load can be accelerated by the crane. With the ability to take the load's inertia out of the hands of the operator, this setup could exploit the full potential of the servo motor. This would obviously require the addition of a new trolley. Less obvious is the need for a new intent sensor. The problem is that for the crane to accelerate a load, a cable angle opposite that created by the operator's push is required. Referring to Figure 2.31, a force P is applied by the operator creating an angle 2_1. But in an attempt to accelerate the load, the crane will create an angle of the opposite sign, such as 2_2. The measure of intent must thus be detached from the angle of the cable. A force sensing joystick located near the load attachment point could supply this detachment. The intended linear acceleration can then be found from this measured force and the mass of the load and the above control logic implemented. A portion of the moment of inertia of the load, maybe 90%, would be included in the determination of the applied torque. Algorithms for minimizing cable swing would have to be implemented; the measured cable angle should be helpful. Another way to minimize cable swing would be implementation of a multiple cable system.

Finally, a third area of exploration is full remote control. The motor and drive are sized to accelerate the full inertia of a 200 lb load. The addition of a powered trolley would make quick motions with no operator intervention possible. In this application, the cable angle sensor would once again be valuable for minimizing cable swing and the ultrasonic sensor would be useful for calculating the moment of inertia and determining the jib crane's kinematics.

[1] R. Juvinall and K. Marshek. Fundamentals of Machine Component Design. 1991.

[2] Kollmorgen. Regeneration Requirements.
http://kmtg.kollmorgen.com/Service/Documentation/ASU001H.pdf

[3] A, Lorenz. Force Sensors for Human/Robot Interaction. Aug. 1999.

[4] P. Shirley. An Introduction to Ultrasonic Sensing. *Sensors*. Nov. 1989.

Appendix A

The drawings of all fabricated components are given here   They are referenced to Figure A.1 below.

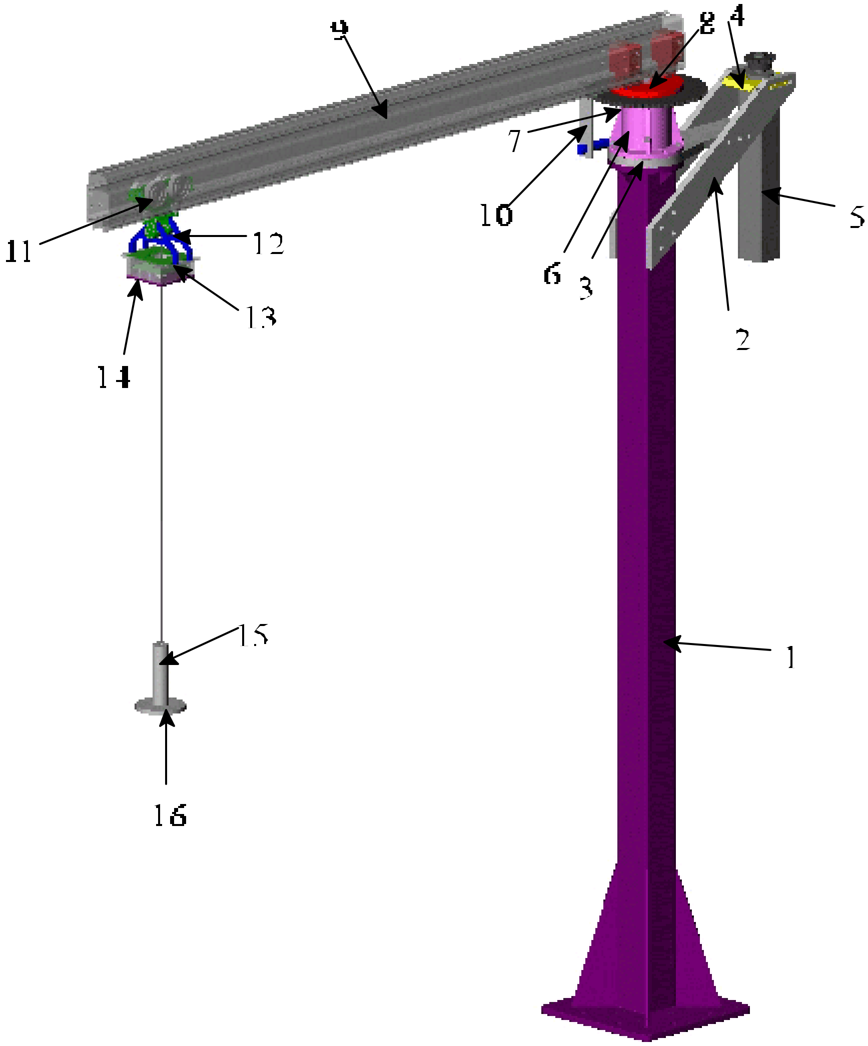
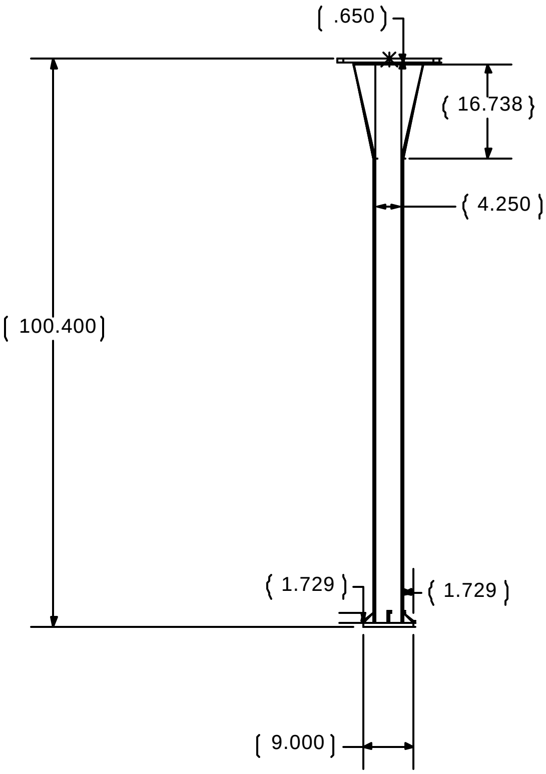
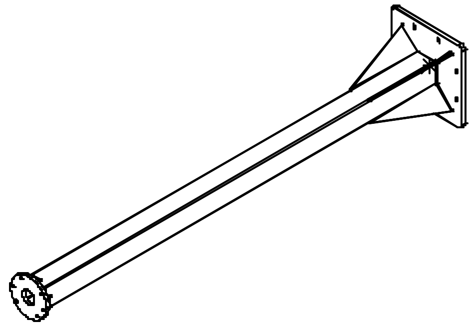
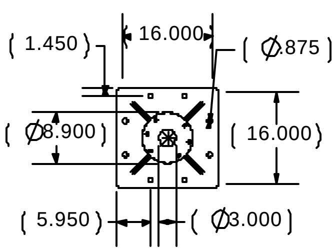


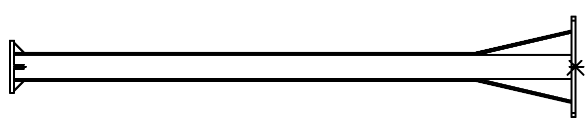
Figure A.1 *Component Drawing Reference*

1 Mast (steel)

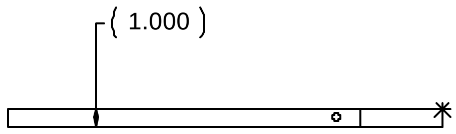

(.650)
(16.738)
(4.250)
(100.400)
(1.729)
(1.729)
(9.000)

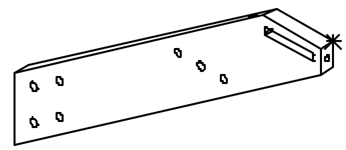


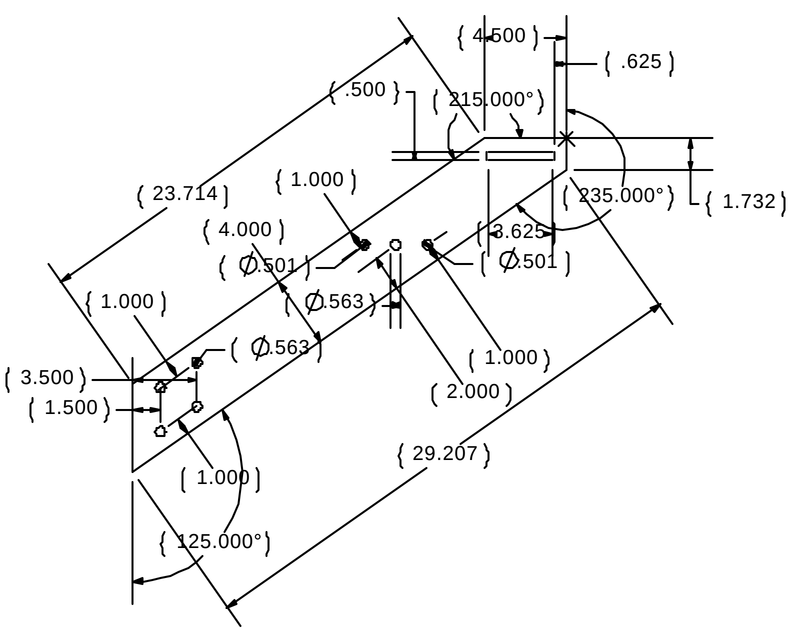

(1.450)
16.000
(Ø.875)
(Ø8.900)
(16.000)
(5.950)
(Ø3.000)

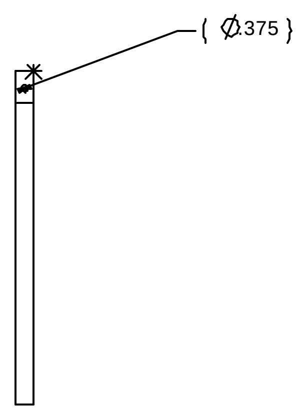


2 Diagonal Motor Mount Arm (Al)

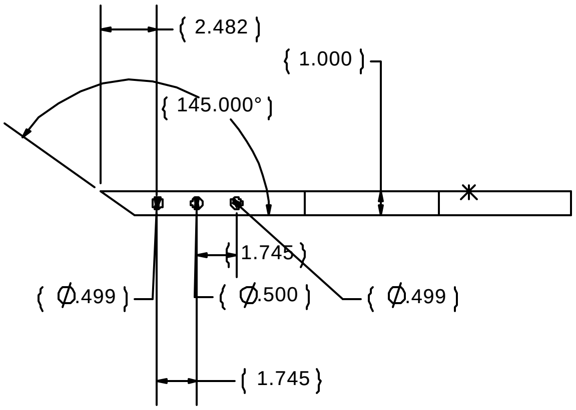


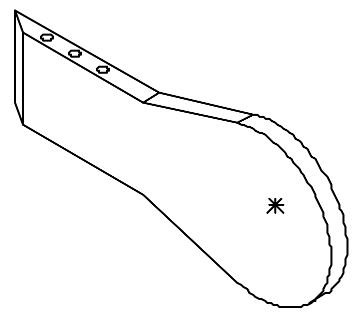


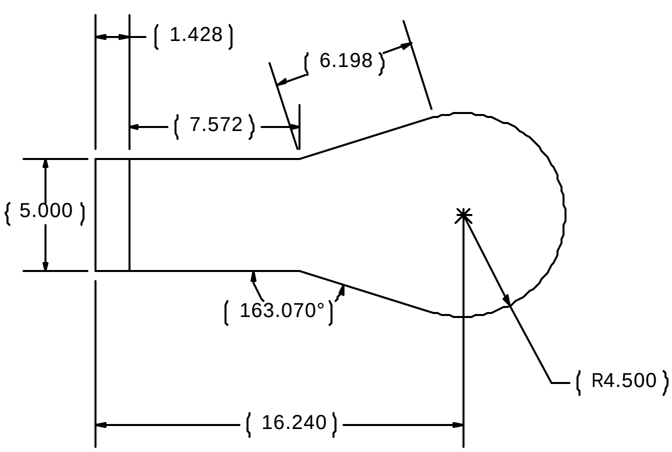


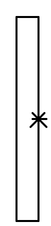


3 Horizontal Reference Motor Mount Arm (Al)

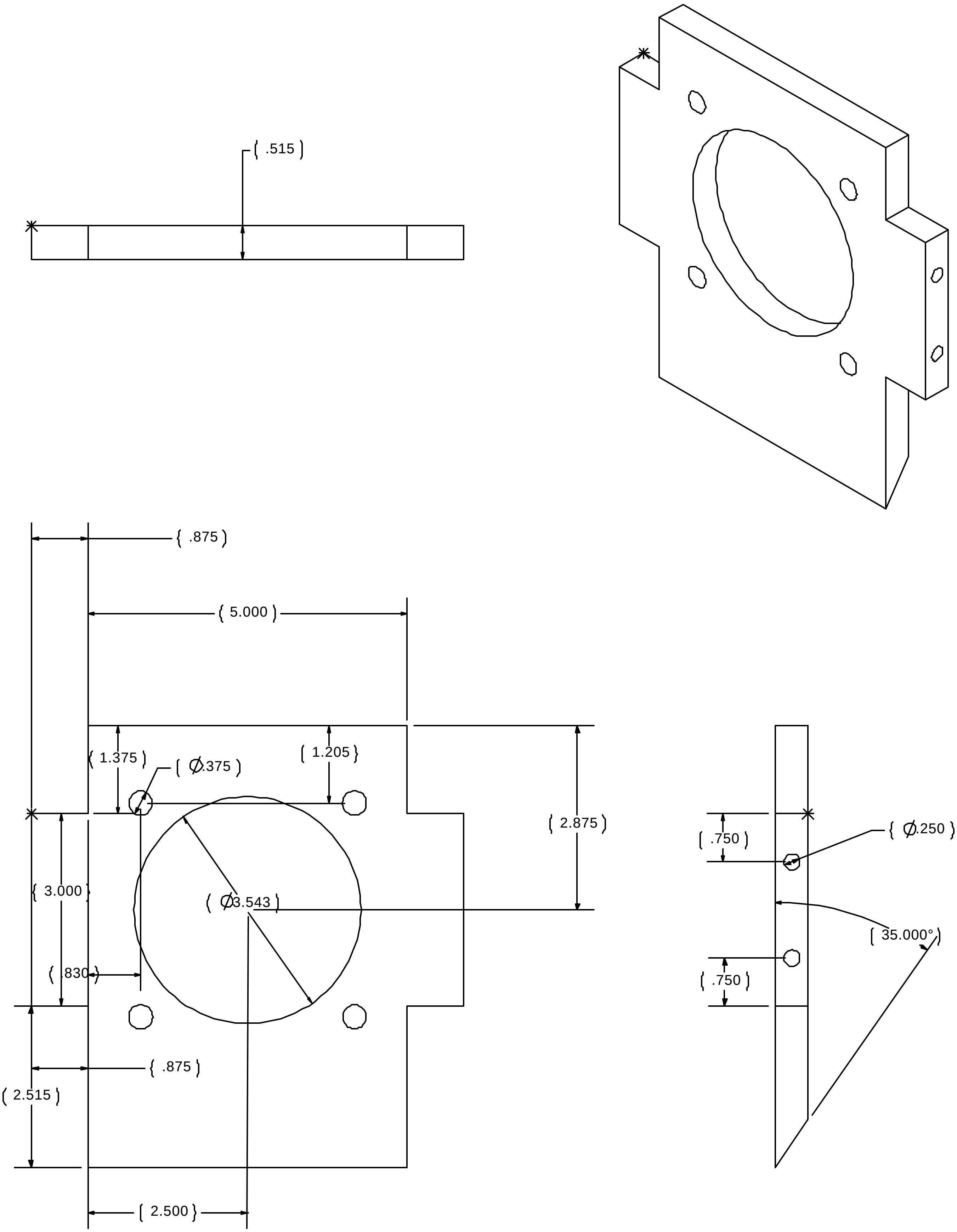




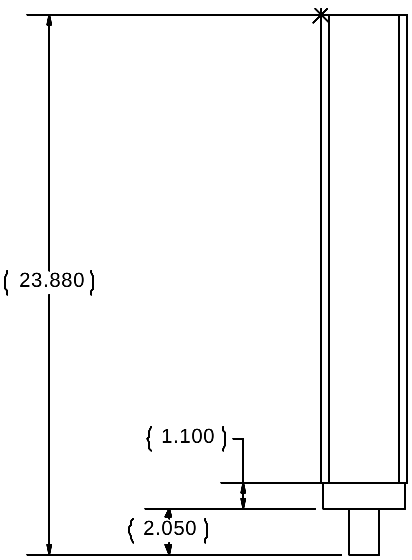


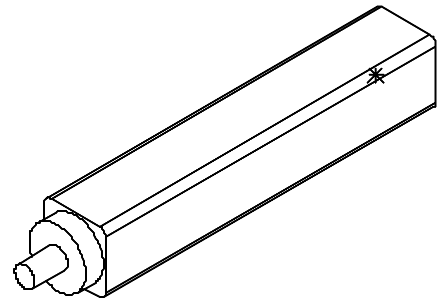


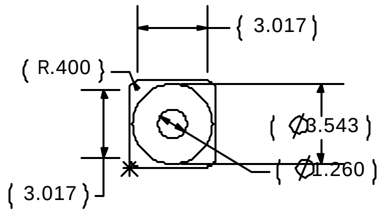
4 Motor Mount Plate (Al)

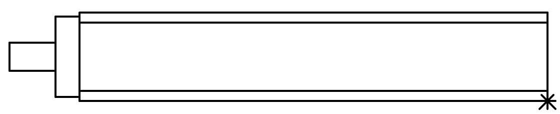


5 Servo Motor and Gearhead

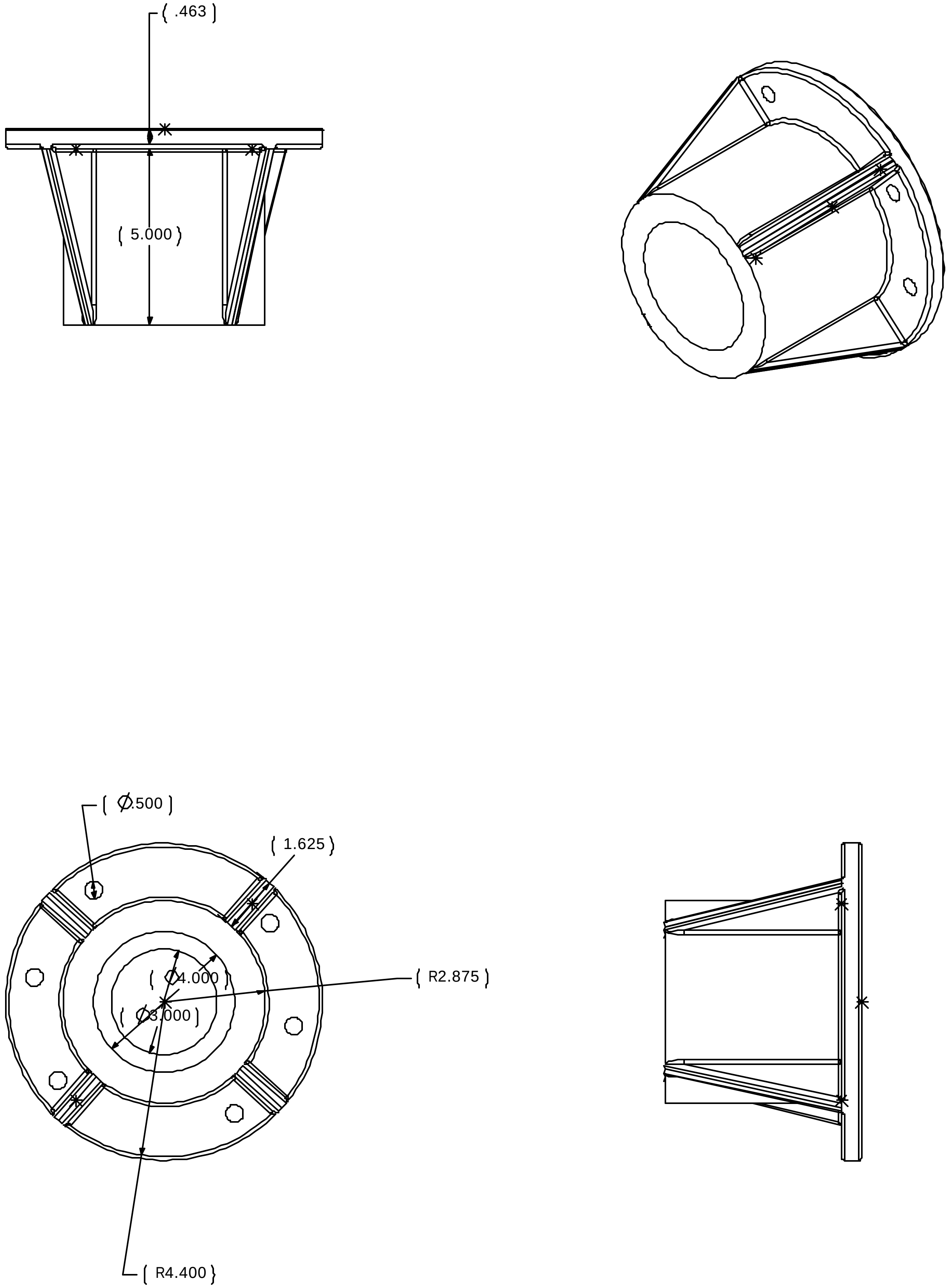




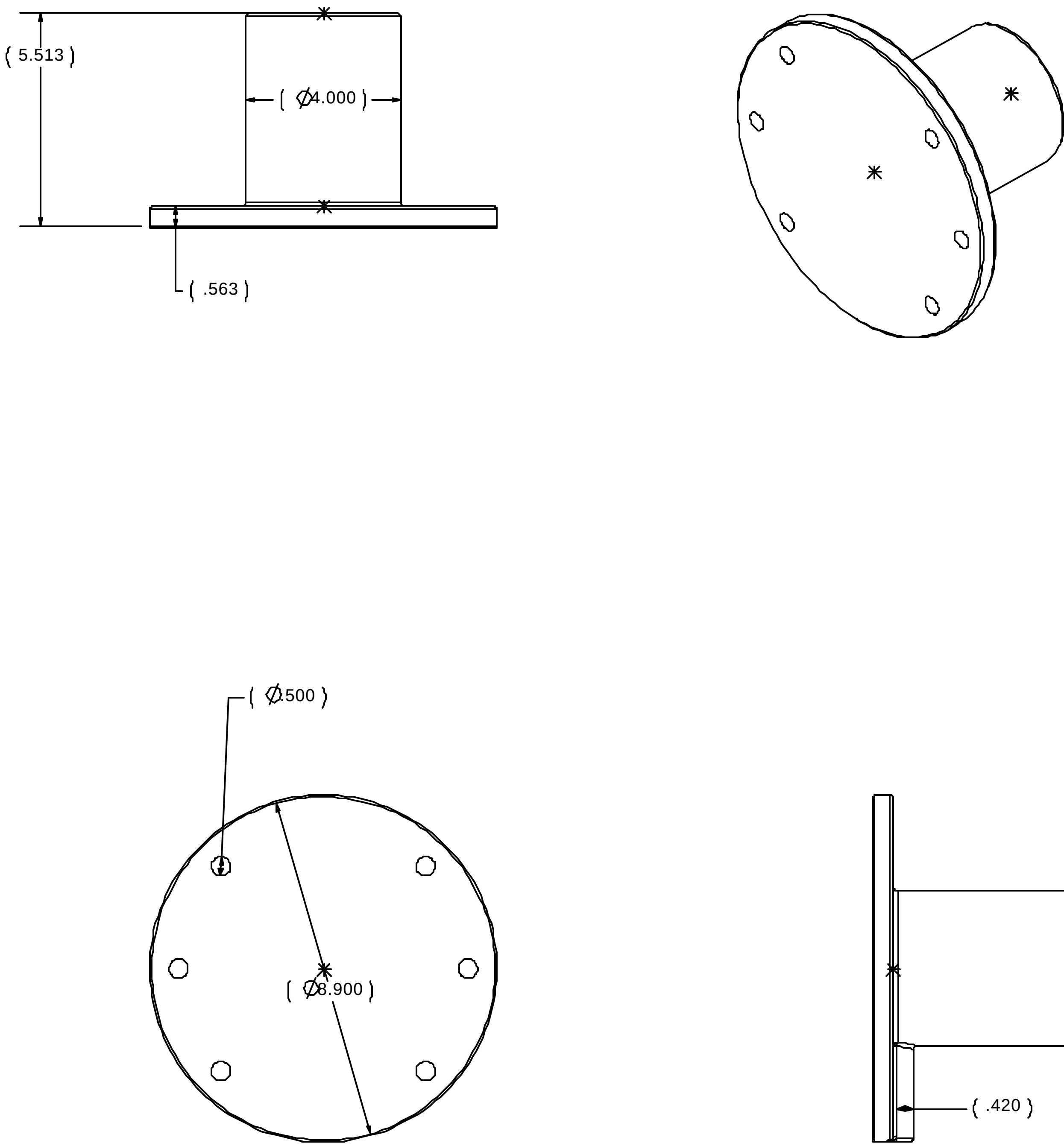




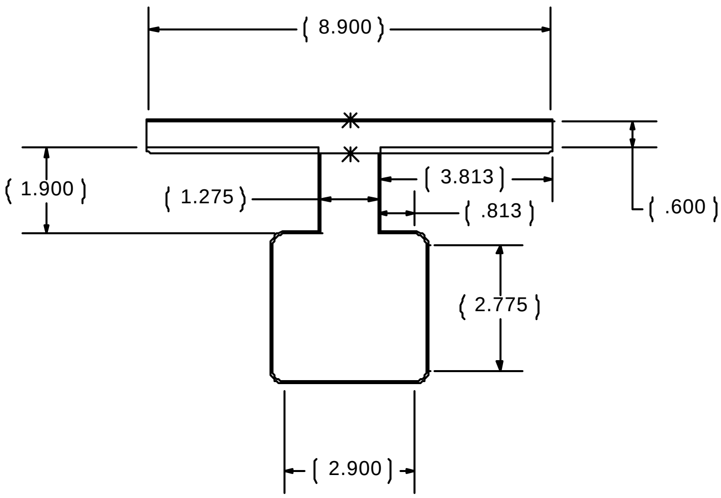
6 Lower Pivot (steel)

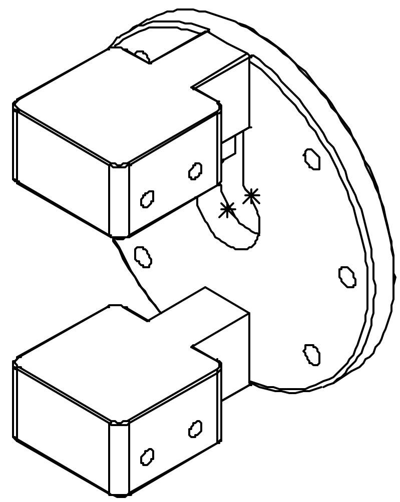


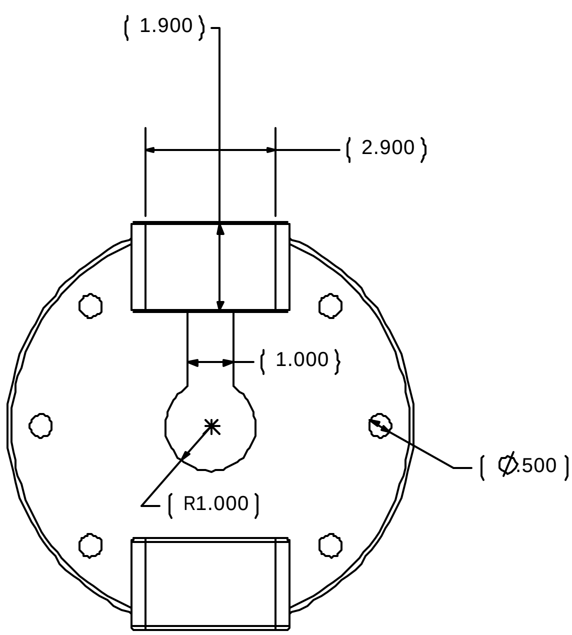
7 Upper Pivot (steel)

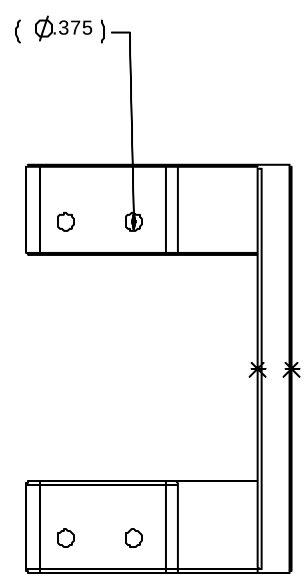


8 Boom Mount (steel)

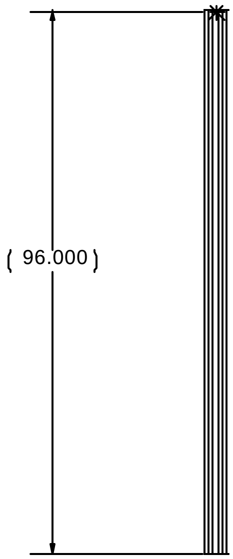


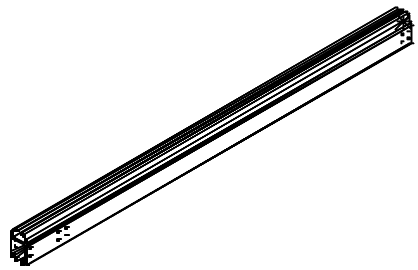


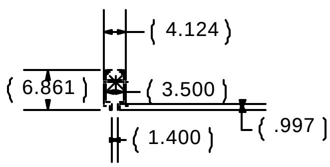




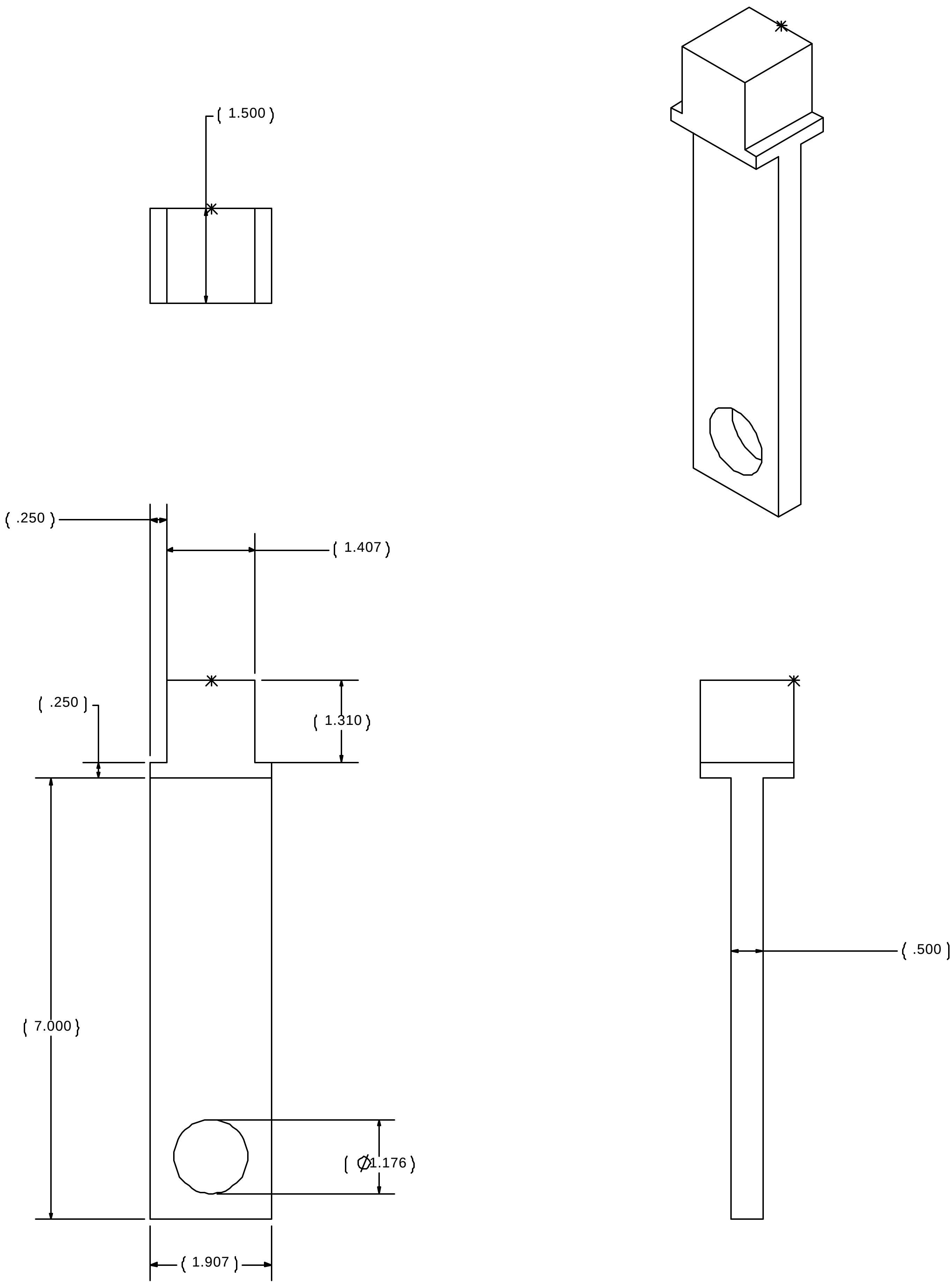
9 Boom (Al)



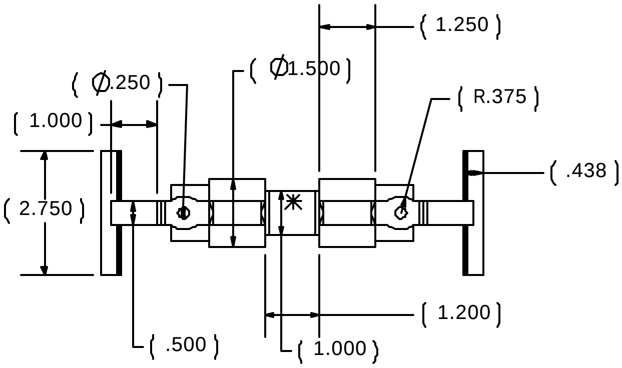


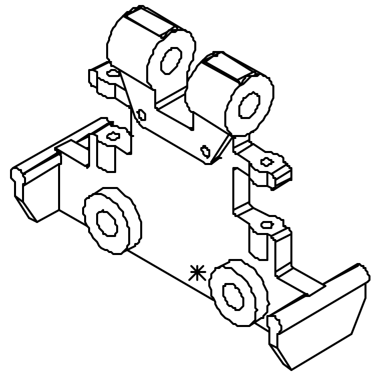


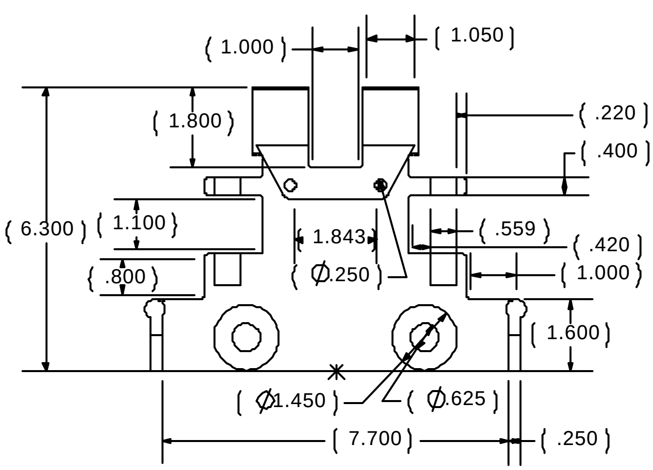
10 Ultrasonic Sensor Bracket (Al)

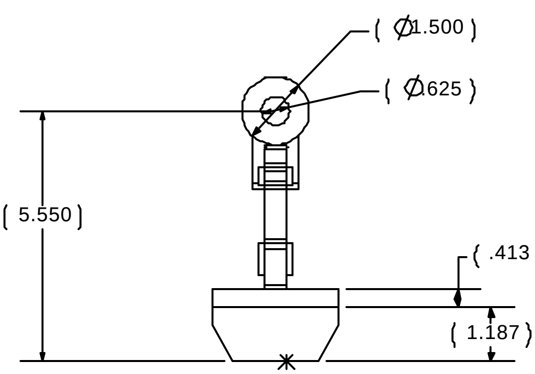


11 Trolley Body (Al)

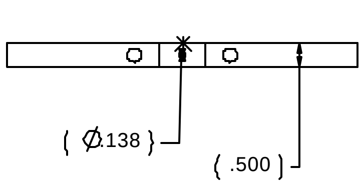


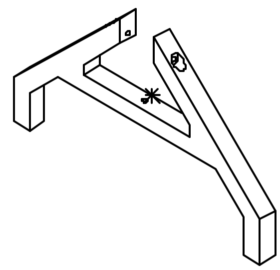


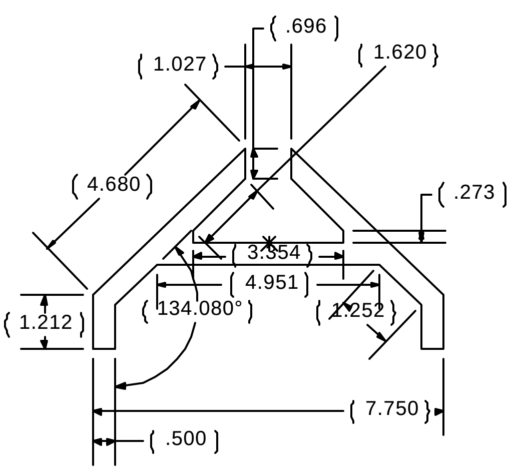


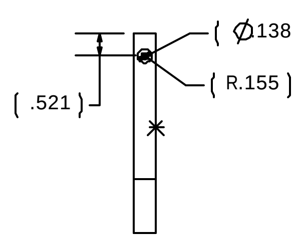


12 Cable Angle Sensor Arm (Al)

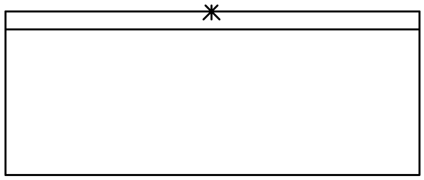


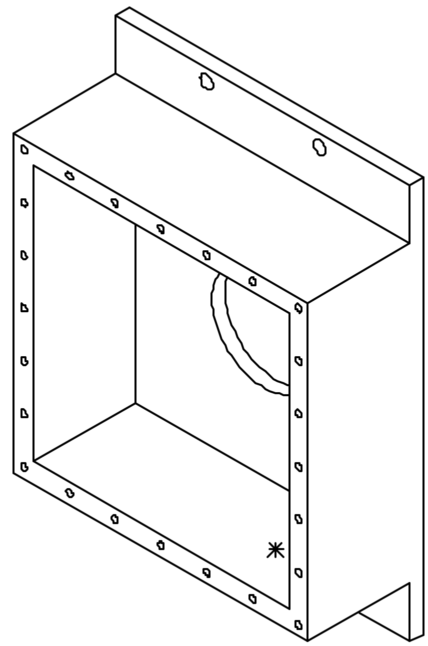


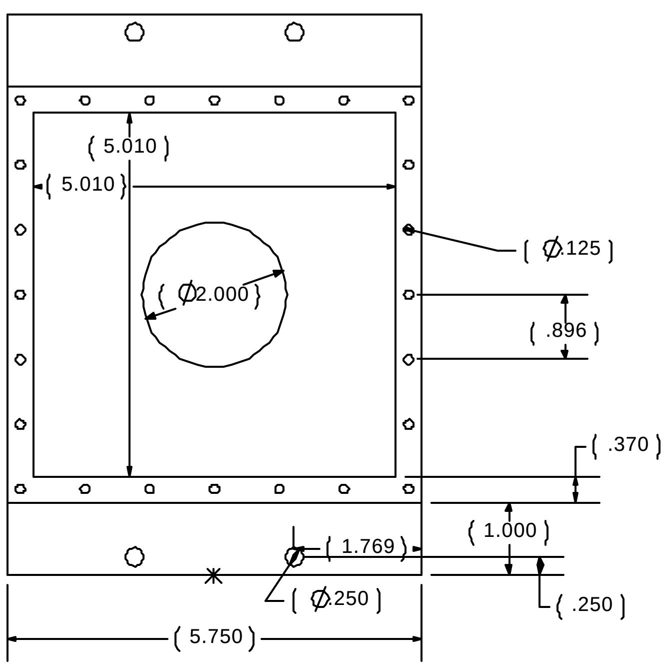


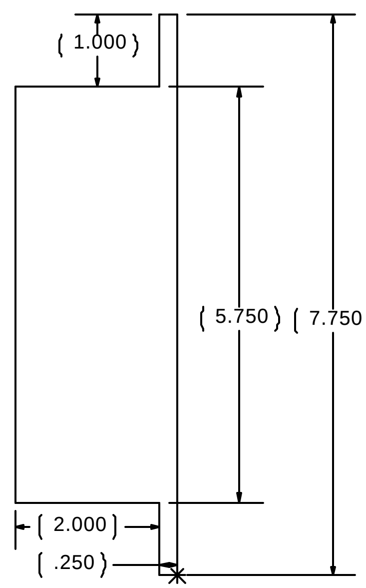


13 Cable Angle Sensor Enclosure (Al)

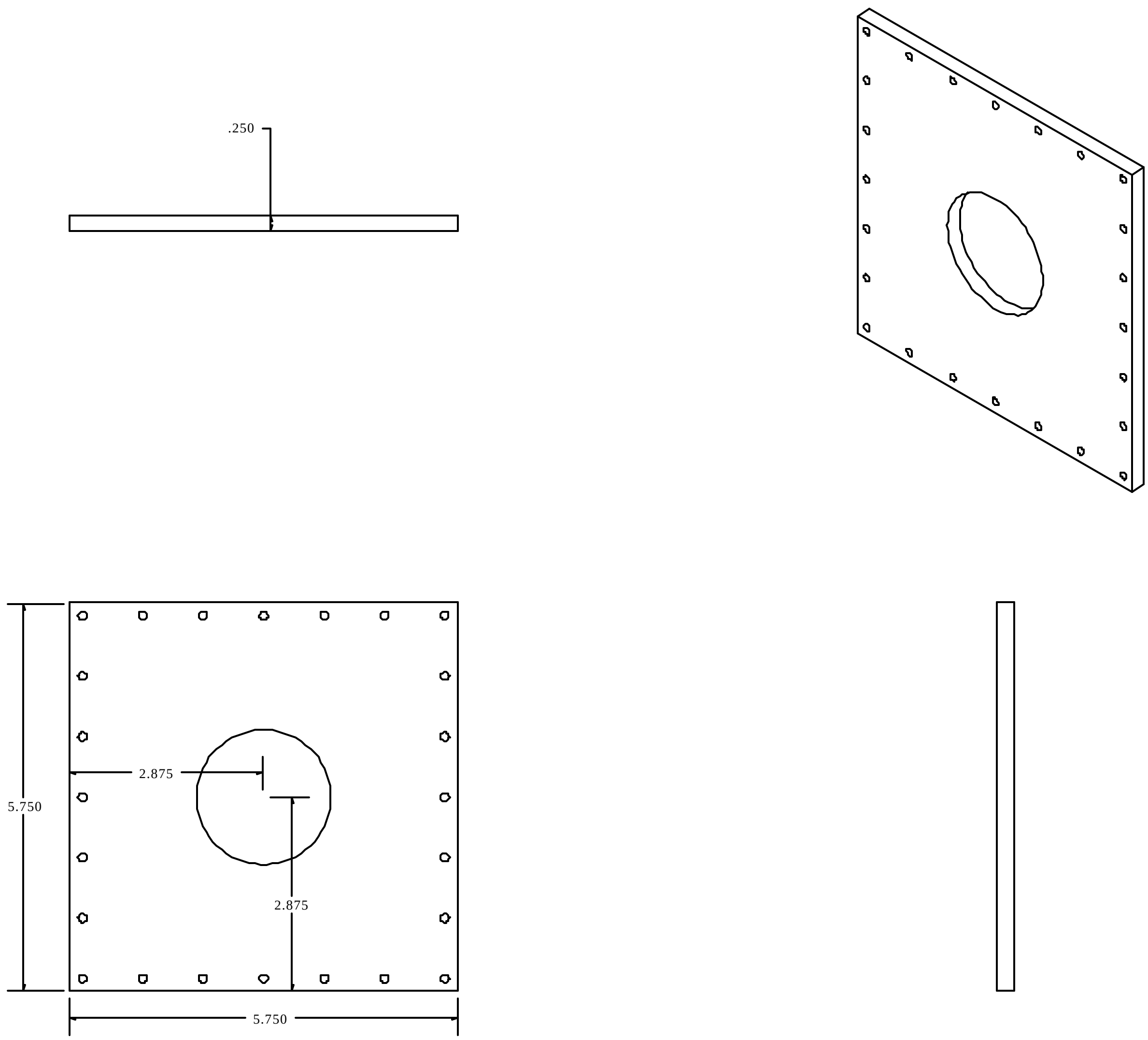




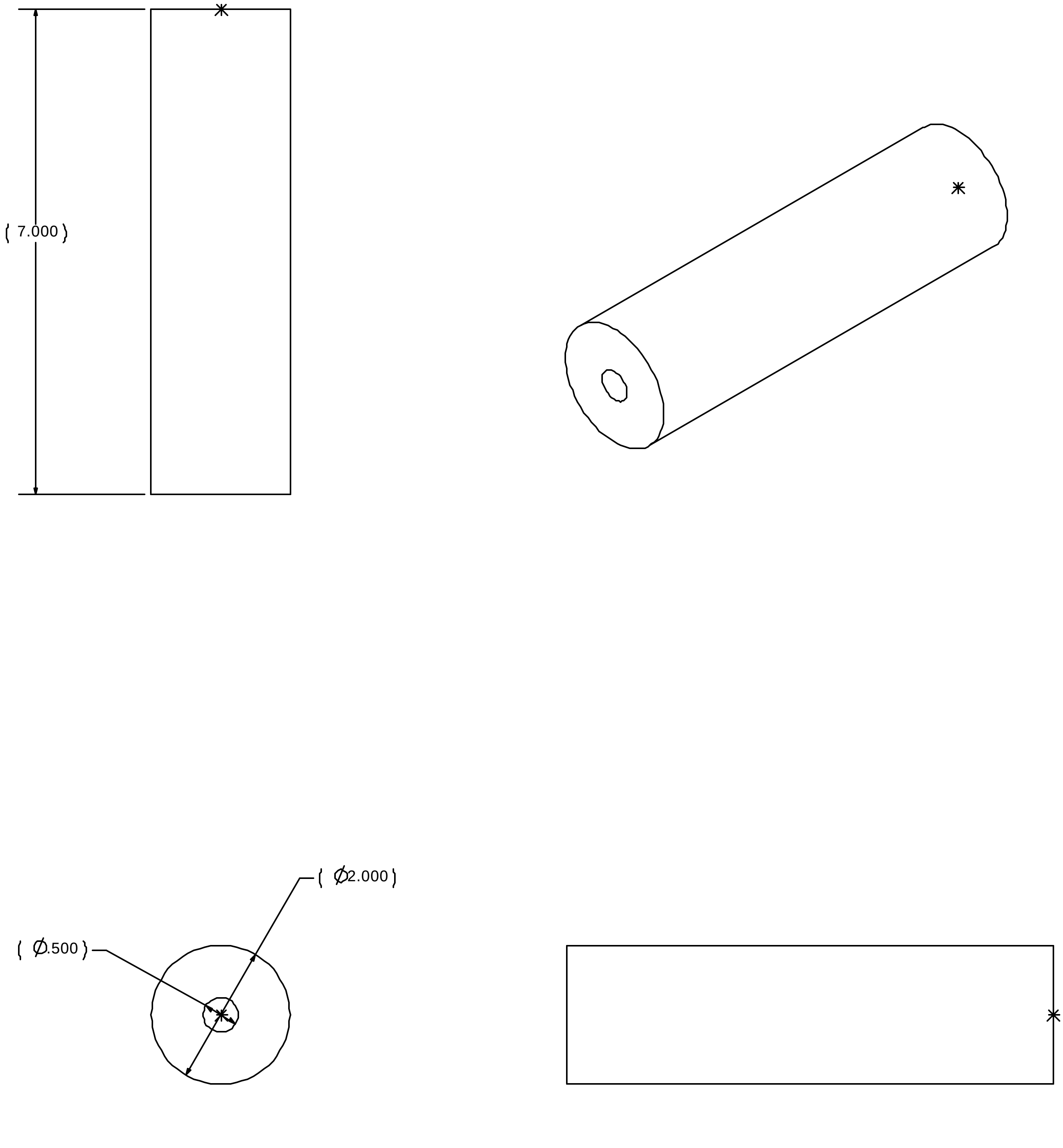




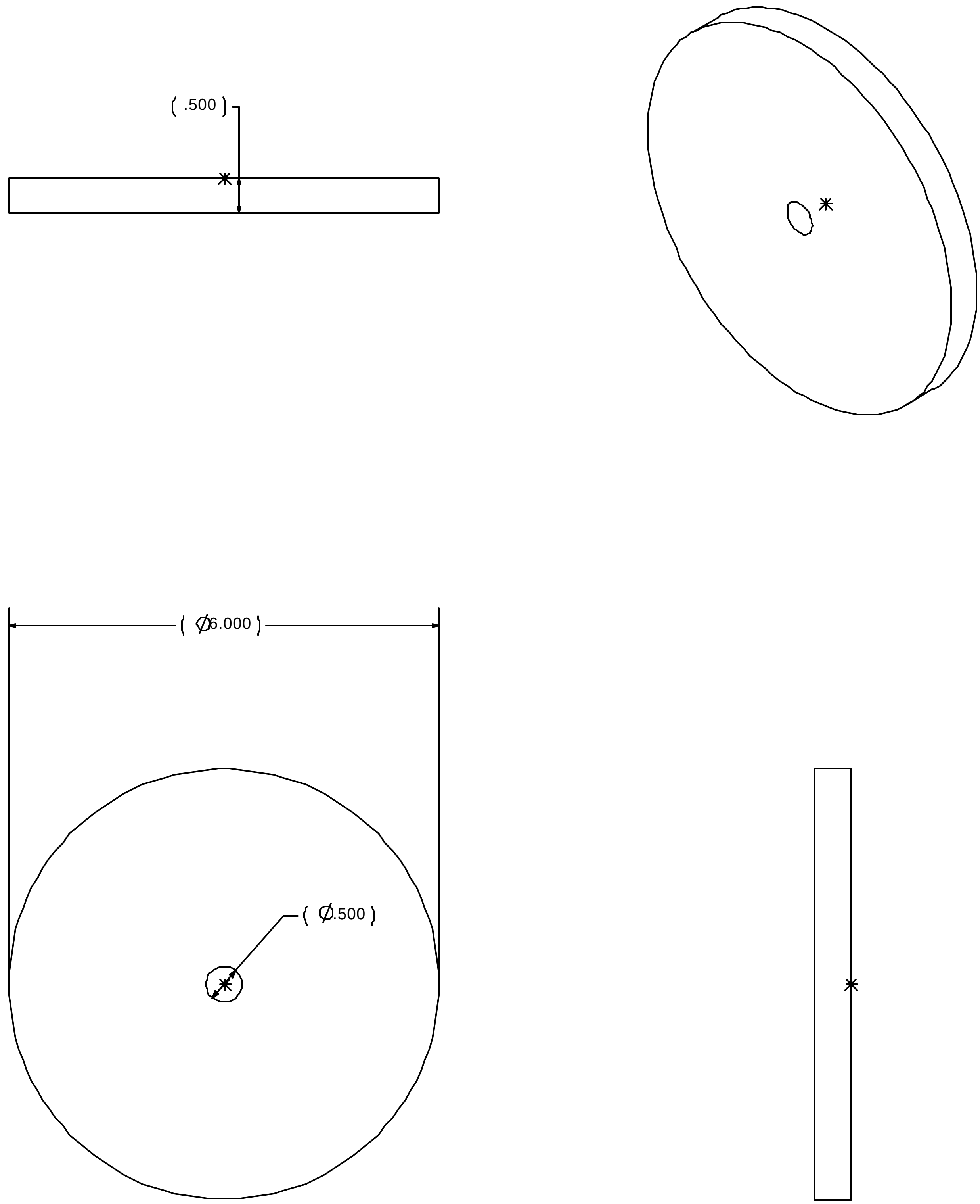
14 Cable Angle Sensor Cover (Al)



15 Weight Column (Al)



16 Weight Plate (Al)



Appendix B

Component Information

<table>
<tr><th>Component</th><th>Manufacturer and Model Number</th><th>More Information</th><th>Price</th></tr>
<tr><td>jib crane</td><td>Zimmerman 900J</td><td>www.irhoist.com</td><td>n/a</td></tr>
<tr><td>servo motor & gearhead</td><td>Kollmorgen
ME2-207-B-A2-B3P3D</td><td rowspan="4">www.kollmorgen.com</td><td>4185</td></tr>
<tr><td>drive</td><td>Kollmorgen
CE06200-2G207B</td><td>1140</td></tr>
<tr><td>motor cable set</td><td>Kollmorgen
CS-SS-CAA27</td><td>454</td></tr>
<tr><td>regeneration resistor</td><td>Kollmorgen
ERH-26</td><td>149</td></tr>
<tr><td>electrical enclosure</td><td>Hoffman A-16N12BLP</td><td>www.hoffmanonline.com</td><td>92</td></tr>
<tr><td>circuit breaker</td><td>Siemens 40 amp</td><td>www.newark.com</td><td>50</td></tr>
<tr><td>high frequency grounding pieces</td><td>Phoenix Contact
SK14, AB/SS, NLS-CU 3/10</td><td>www.phoenixcon.com</td><td>60</td></tr>
<tr><td>data acquisition</td><td>Dataq DI-150RS</td><td>www.dataq.com</td><td>100</td></tr>
<tr><td>rollerchain</td><td>Diamond 60HS</td><td>www.diamondchain.com</td><td>81</td></tr>
<tr><td>small & large sprocket</td><td>Martin 15B15, 60A60</td><td>817-473-1526</td><td>120</td></tr>
<tr><td>ultrasonic sensor</td><td>Honeywell
946-A4V-2D-2C0-175E</td><td>www.honeywell.com</td><td>546</td></tr>
<tr><td>numerous bolts, hooks, etc.</td><td>many</td><td>www.mcmaster.com
www.grainger.com
www.digikey.com</td><td>n/a</td></tr>
</table>

Appendix C

Conversion Factors

density	$\dfrac{515.4\dfrac{kg}{m^3}}{\dfrac{slug}{ft^3}} = 1$
force	$\dfrac{4.448\,N}{lb} = 1$
length	$\dfrac{0.3048m}{ft} = 1$
mass	$\dfrac{14.59kg}{slug} = 1$
mass moment of inertia	$\dfrac{1.355kg \cdot m^2}{slug \cdot ft^2} = 1$
power	$\dfrac{745.7W}{hp} = \dfrac{550\dfrac{ft \cdot lb}{s}}{hp} = 1$
torque	$\dfrac{1.356\,N \cdot m}{lb \cdot ft} = 1$