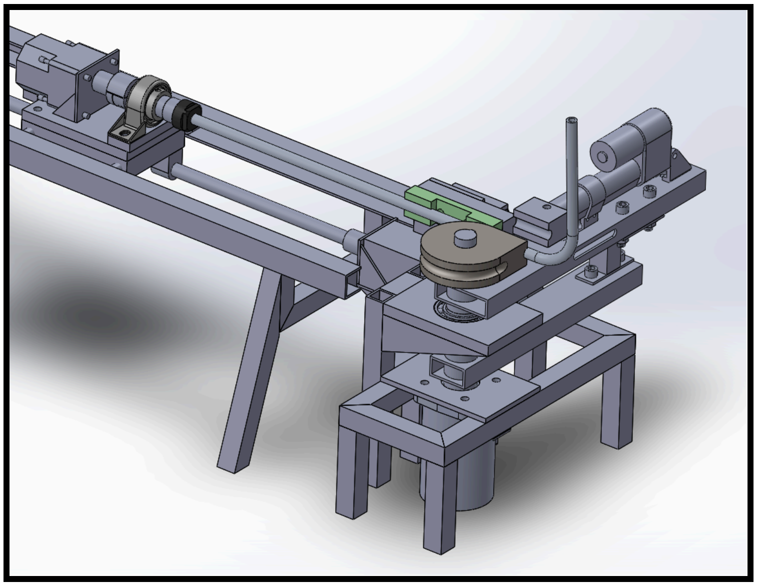

# 3-D CNC Tube Bender

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**Abstract**

The criteria required to build a 3D CNC Tube Bender involved designing an easy to use machine for a small manufacturing setting, as well as the calculations required to make selections on components and assemblies. Easy operation of the machine will involve making the tube bender that offers CNC operation so designs are repeatable.

The CNC tube bender is semi-automatic; it uses an automated bending die, clamping die, carriage, and tube rotator. Each motion happens incrementally to produce a bent tube. The tube rotator will require manual tightening around the workpiece (tube), it is the only manual operation on the machine.

The power to bend a tube will be supplied by the motor connected to the bending die. Bending the tube will cause the back end of the tube to want to spring back which is why in the design there is a static follower clamp. The follower clamp resists spring back and has a nylon insert to reduce friction as the tube slides through. A pressure die clamps the tube against the bending die and rotates on a table for different length bends. The pressure die is CNC controlled and un-clamps after every bending operation to allow for more tube to feed through for additional bends.

During ME 191, different steel tubes will be tested at different bend angles to induce plastic deformation. Springback is a main testing concern with the CNC tube bender. If the tube bender is intended to bend steel hollow tubes there will be a minimum diameter bend which can be produced. During testing different parameters will be tested to reduce springback and to set a range of possible bend angles.

**TABLE OF CONTENTS, INCLUDING LIST OF FIGURES AND TABLES**



**NOMENCLATURE**

-STRESSES IN BEAMS

-PARTS IDENTIFICATION

-"DRIVE SHAFT"

-"BEND DIE"

-"CLAMP DIE"

-"FOLLOWER DIE"

-"TUBE ROTATOR"

# 1 PROBLEM STATEMENT & INTRODUCTION/BACKGROUND

## Problem Statement

The objective is to build a CNC rotary draw tube bending machine that can bend steel tubing from 0 to 180 degrees in 3-dimensions and will be fully automated by an arduino microcontroller. CNC is chosen over a manual bending machine to eliminate the human error associated with non automated processes. The driving force of this design is to have a consistent tube feed rate and to output consistent bends within a tolerance of +/- 1.5 degrees.

## Introduction & Technical Background

## Thomas W. C. Carlson

**Standard Machine Design Approach**

- **Beam loading**
- **Motion, resultant forces**
- **Assumptions**

Market options for the deformation of metal tubing to produce bends of certain radii and degree include press bending, rotary draw bending, and jig bending. In press bending, the die around which the tube is to be bent is pressed against the tube to force it to deflect around the surface of the die [Citation D1]. This process is simple and quick to repeat, but invites low precision and concerns about crushing the tube. Roll bending is performed through the use of 3 or more rolling dies, each of which deforms the tube slightly. Summing the deflections produced by each roller produces the output bend of the tube. This is generally suited toward longer gradual bends, as the space taken to produce each angular deflection is much larger than can be produced by a single bend die [Citation D1]. Rotary draw bending, the method of choice for this paper, offers greater precision than press and roller bending in tight radius situations and greatly expands the configuration options [Citation D2]. Specific advantages are conferred through the linear nature of loading new length of tube into the bending region of the machine, allowing for multiple bends in a single tube as well as a simplified method of removing the workpiece from the machine after bending completes.

Rotary draw bending machines have a standard design which can be varied in a few ways. The four basic components of any rotary draw bending machine are the bend die, the clamp die, the pressure die, and the wiper die, shown below in Figure C1. The bend die supports the tube, guiding it around the bend radius of the die to graduate the bend. The clamp die is attached to a bending arm which rotates the clamp around the bend die, pulling the tube through the bend and causing plastic deformation to occur. The pressure die ensures that the tube stock remains linear, effectively forcing the material into the bending region. There are two options for pressure dies: dynamic and static. The dynamic pressure die aims to assist the bend and clamp dies in achieving a clean deformation (no kinks or wrinkles) by moving forward with the tube and applying a “boost” pressure. A static die remains in place, simply guiding the material into the clamp and bend die as the operation proceeds. These three components are absolutely necessary in any rotary draw application--the wiper die remains optional. The benefit of the wiper die is felt in conjunction with the use of a ball mandrel, becoming required when the ratio of tube diameter to wall thickness exceeds a certain figure at a certain bend radius [Citation M2]. The machine designed in this paper will take care to avoid this situation.

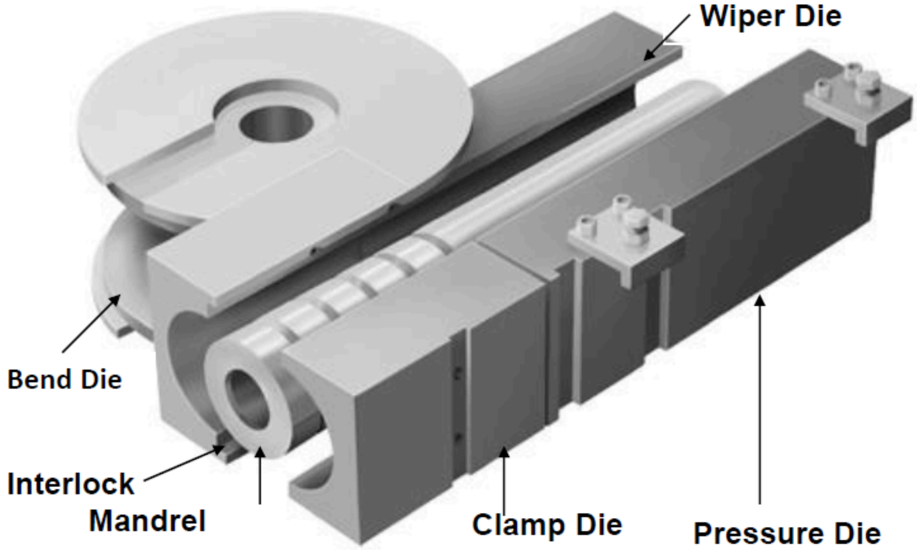



Figure C1 - Rotary draw bending assembly [Citation M4]

Design of the die dimensions is of key importance to the bending process--their algebraic dimensions can be seen in Figure C3. Of note is the fact that a bending die is selected through the desired centerline radius of the bend and the outer diameter of the tube and is not a factor in designing the machine. It must be purchased. The clamp die must conform to the same dimensions, and its length will be exactly equal to the clamping surface of the bend die, shown in the figure as **d**. For a static pressure die, the value of **c** will be set to zero, creating a cantilevered beam about point B as shown in Figure C4. The selection of the length **b** of the pressure die is the most variable design parameter. It's effect can be felt through variation of parameters that produce different bending moment graphs via the use of singularity equations and the closed form solutions for certain values as shown in Figure C5. It is important to note that the desired shear and bending moments across the section defined by length **a** should be zero so as to avoid damaging the carrier--this structure should only be lightly supported against bending moments and able to move forward.

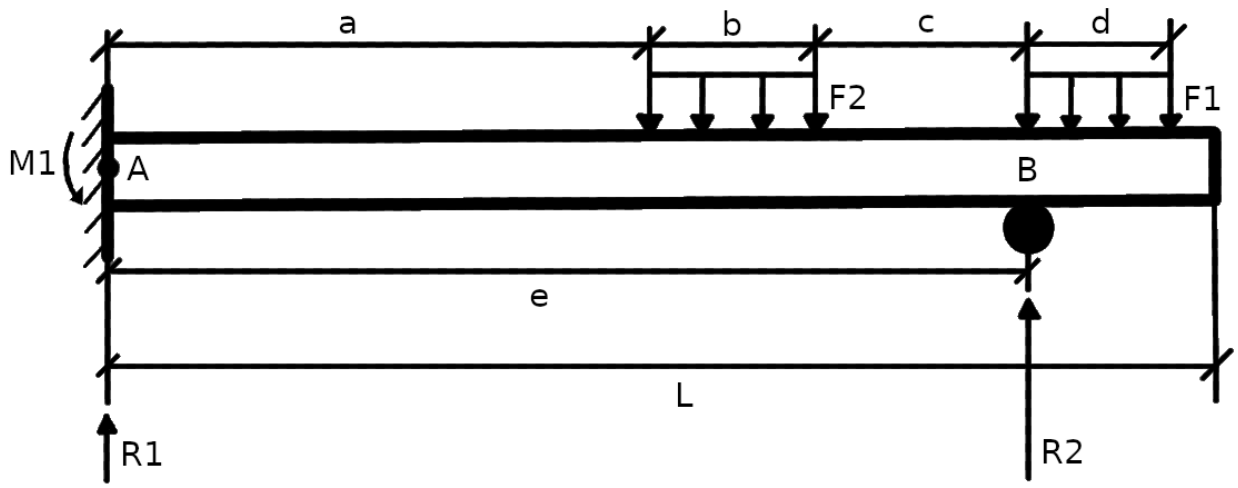



Figure C3 - Free body diagram of the tube modeled as a beam with fixed and roller support

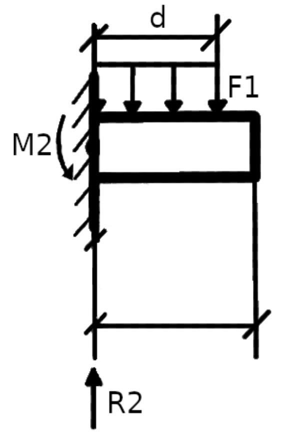



Figure C4 - Free body diagram of the cantilever created by the pressure die and bend die

$$R_1 = F_1 d - \frac{F_1 d^2}{2a + b - 2e}$$

$$F_2 = \frac{F_1 de}{(a + \frac{b}{2} - e)(b)} - \frac{F_1 d(e + \frac{d}{2})}{b(a + \frac{b}{2})} - \frac{F_1 de(e + \frac{d}{2})}{(a + \frac{b}{2} - e)(b)(a + \frac{b}{2})}$$

$$V(x) = -F_2\langle x - a\rangle^1 + F_2\langle x - (a + b)\rangle^1 + R_1\langle x - e\rangle^0 - F_1\langle x - e\rangle^1 + F_1\langle x - (e + d)\rangle^1$$

$$M(x) = -\frac{F_2}{2}\langle x - a\rangle^2 + \frac{F_2}{2}\langle x - (a + b)\rangle^2 + R_1\langle x - e\rangle^1 - \frac{F_1}{2}\langle x - e\rangle^2 + \frac{F_2}{2}\langle x - (e + d)\rangle^2$$

Figure C5 - Values and singularity equations used to produce bending moment diagrams

The rear end assembly of a tube bending machine must be capable of carrying the weight of the tube as well as moving forward toward the bending region at the same speed that the tube is drawn away. It consists of a carriage, responsible for bearing the weight, and a method of pressing the tube forward--either an actuator, drive shaft, or gearing.

Many of the applications for bent tubing, such as the airplane electrical conduit guide shown in Figure C2 require the functionality of applying multiple bends to a single section of tubing without significant deformation of the cross-section and surfaces of the tube. Integrating CNC provides a significant advantage over manually-controlled deformation approaches, allowing for seamless operation cycles with minimal retooling downtime. It also improves the accuracy of bends through its precision control and the elimination of human error brought into the production cycle via hand-operated and controlled tube benders. As production demands increase, CNC becomes very desirable for its ability to produce consistent output and reduce the need for human attention during operation. A particularly useful feature of CNC tube benders is the possibility of designing for multiple bend radii, further reducing time spent retooling and increasing manufacturing capabilities.

Figure C2 - Image of Boeing 737 wheel well, taken at Sacramento Aerospace Museum

**Limitation on Tube Deformation**

**Crumpling, Kinking, Buckling**

**Wall thinning**

**Centerline Radius**

**Springback**

The performance of a tube bender rests heavily on the accuracy of the bend and on the minimization of unnecessary damage to the workpiece and the machine itself. Damage to the workpiece can heavily impact the accuracy of a bend relative to the commanded operation by deforming the tube while damage to the machine reduces the degree of repeatability that can be achieved without constant and expensive maintenance. The process of rotary draw tube bending is vulnerable to the failure modes represented in Figure N1.

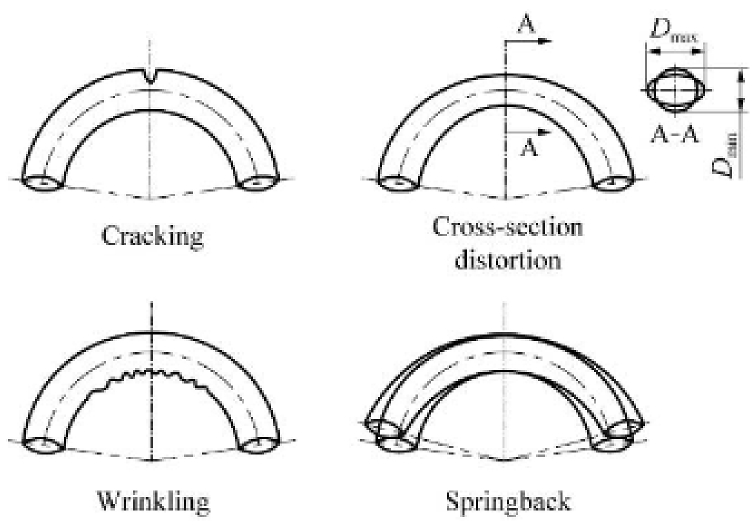



Figure N1 - Bending manufacturing defects [Citation M1]

In general, the smaller the Centerline Radius (CLR) of the bend, the more stressful the operation and the more likely uncontrolled deformation of the tube will occur. [Citation M2] For small CLR the typical solution to the problems of cross-section distortion, cracks, and wrinkles is the implementation of a mandrel to support the frame of the tube during the bending action and ensure that the pressure is translated through both tube walls equally as in Figure N2, ensuring that the bend die guides the shape without shear stresses causing the tube shape to collapse [Citation M2].

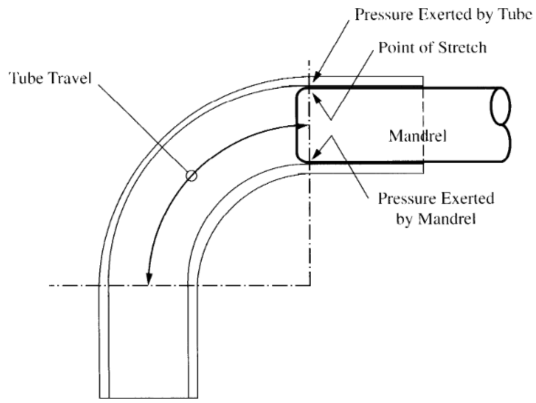


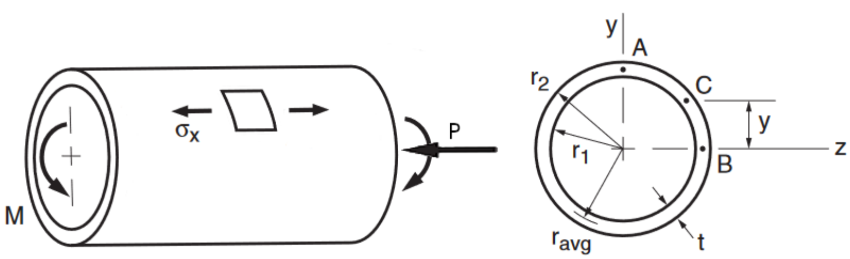

Figure N2 - Mandrel supporting the bend [Citation M2]

Mandrel bending increases tooling costs and introduces new engineering challenges regarding placement and retraction. To develop a system which avoids the implementation of a mandrel, the minimum CLR must be regulated. The following chart describes the minimum suggested CLR for bending without a mandrel: [Citation M2]

| | Wall Thickness (inches) | | | | | |
|---|---|---|---|---|---|---|
| Tube Diameter (inches) | .035 | .049 | .065 | .083 | .093 | .120 |
| 3/16 | 5/16 | 1/4 | 3/16 | | | |
| 1/4 | 1/2 | 3/8 | 5/16 | | | |
| 5/16 | 7/8 | 3/4 | 5/8 | | | |
| 3/8 | 1-1/2 | 1-1/4 | 1-1/8 | 1 | | |
| 1/2 | 2-1/4 | 2 | 1-3/4 | 1-1/2 | | |
| 3/4 | 4 | 3 | 2-1/2 | 2 | | |
| 1 | 8 | 6 | 4 | 3 | 2 | 2 |
| 1-1/2 | | | 12 | 10 | 8 | 6 |
| 2 | | | | 24 | 20 | 16 |
| 2-1/2 | | | | | 24 | 20 |
| 3 | | | | | | 25 |

Table T1 - Minimum Bending Radius [Citation M2]

Excessive wall thinning as a result of elongation of the material in rotary draw may produce cracking or a workpiece which is too weak against pressurization for the desired application. It is possible to evaluate the degree of thinning by performing a stress-strain analysis on the workpiece along the thinning section of the tube as in Figure N3. The analysis assumes that Hooke's Law applies, and that the tube is constrained. The worst case for thinning can be found at the point furthest from the surface of the bending die, indicated by point A. [Citation M5] The forces indicated in Figure N4 are those produced by the clamping die and the bending die, while P from figure N3 is the force of the carriage pushing the tube forward into the bend. Force P will help to correct the problem of wall thinning by preloading the tube to lessen the effect of the axial stress induced by the bending action [Citation M2]. The amount of wall thinning can be calculated as the out of plane strain $\boldsymbol{\varepsilon}_y$ of the element located at A over the initial wall thickness. Determination of the necessary force P, "boost", to prevent a certain threshold of thinning is to be done by iteration. It should be noted that for larger wall thicknesses, the effect of the out of plane strain may be small enough that P = 0 [Citation M2].



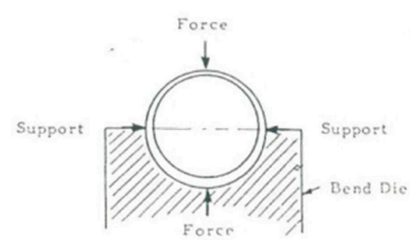




Figure N3 - Model of the tube Element [Citation M5, modified]      Figure N4 - Orientation diagram [Citation M4]

If P is not necessary or present, a model for direct calculation of the inner and outer wall thicknesses is presented in [Citation M7]. Further, in some cases the reduction in wall thickness at the outer surface actually results in a greater strength when the tube is pressurized, possibly due to the strain hardening effect [Citation M7]. The relevant equations are presented in Figure N6 [Citation M7].

$$t_o = 1 - 2k + 12k(4k+2)T$$

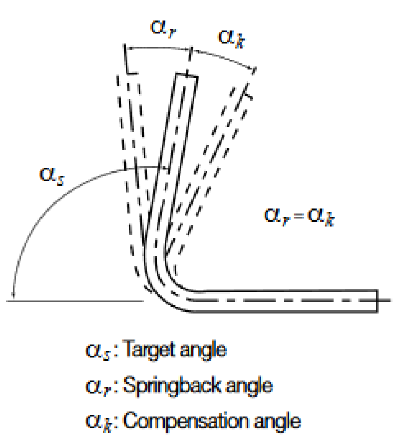
$$t_i = 1 + 2k + 38k^2T$$

$$k = \frac{R}{2r}$$

| | |
|---|---|
| k | Geometry parameter |
| T | Wall thickness |
| R | Bending radius |
| r | Average radius of the tube cross-section |

Figure N6 - Wall thicknesses without "boost" [Citation M7]

Elastic springback of the tube after the bending forces are removed is of great concern. It has been found both experimentally and theoretically through beam theory that the amount of springback which reduces the final bend angle is linearly related to the bend angle the moment after bending has ceased, as shown in figure N7 [Citation M1, M10]. It is therefore necessary to overshoot the target angle by some angle $\boldsymbol{\alpha}_k$, the compensation angle, so that the effect of stress relaxation returns the tube to the target angle as in Figure N5 [Citation M6].



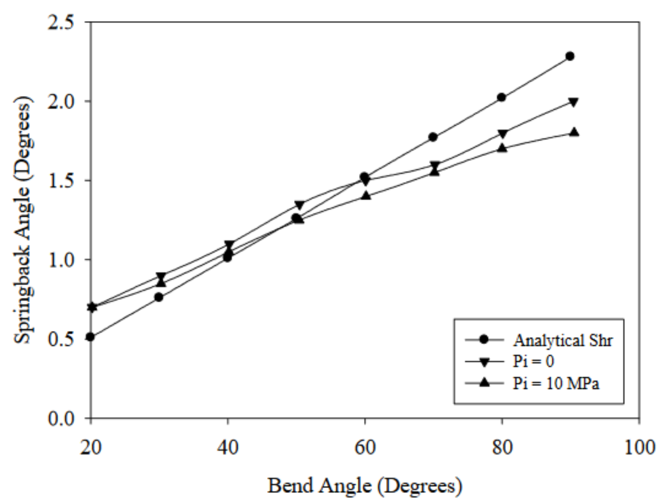




Figure N5 - Bending Angles [Citation M6] Figure N7 - Springback angle-bend angle relationship for SS304 (D=57.15mm) [Citation M10]

Key factors in the springback behavior are the rigidity of the material being bent, the thickness of the tube wall, and the radius of the bend [Citation M4]. In general, the higher the yield strength of a material, the greater the springback angle [Citation M9]. This behavior can also influence the end radius of the bend. Springback angles for different materials are experimentally determined, and for some materials may be tabulated and easily accessible. An applicable estimate of this value can be derived through finite element analysis [Citation M10]. We then identify the ratio of the bending angle to the resulting angle (the difference of the two being the springback angle) as the "Springback Factor" of the material [Citation M9]. This is the slope of the graph provided in Figure N7. The Springback Factor represents the relationship between bent angles and final bend angles. Once the springback factor is calculated for a material, degree, and radius of bend, it is possible to predict the behavior of a tube in bending after it is unloaded, resulting in a further increase in accuracy of performance the machine.

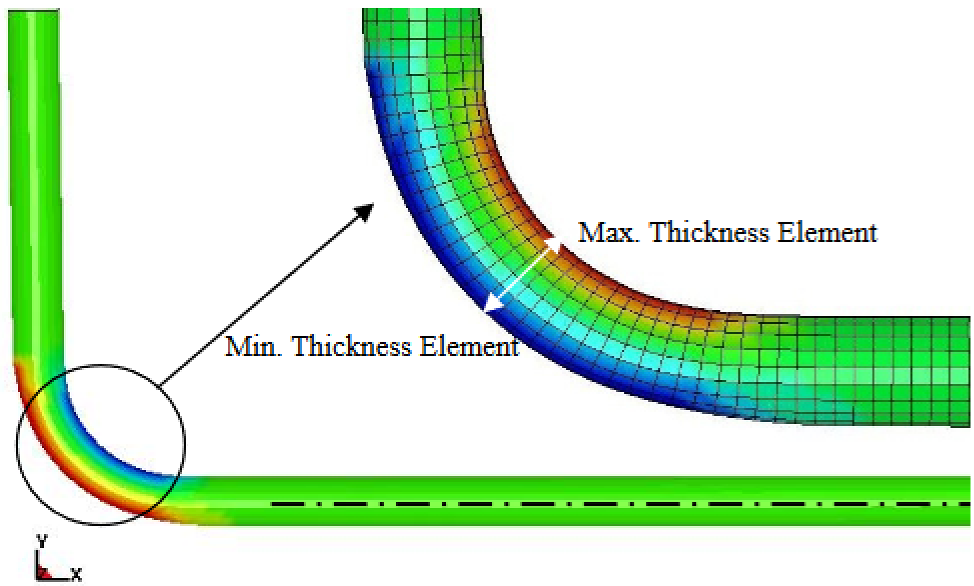



Figure N7 - Finite element analysis on the bent tube

Many of these defects are difficult to predict through pure mathematics and simple application of deformation theory. In particular, the behavior of wrinkling is difficult to model with any accuracy [Citation M1]. Best practice therefore dictates that tests be performed, and the performance of the machine be measured and recorded. In this process it is important to keep the test stock material consistent--when possible, ensure that samples are of the same material and ideally the same production batch to preserve consistency [Citation M4].

**Chase Alexander**

**Clamping Die**

**Electric linear actuators**

Tube rotary draw bending (RDB) has a lot of variations and different ways of handling tube deformation and failures. Many of the problems that arise from RDB that were previously laid out come from slippage of the tube along the die as it is being bent. This effect is the reason for the clamping die as seen in Figure C1 - Rotary draw bending assembly [Citation M4]. The primary job of the clamping die is to hold the tubing firmly to the bending die as it is rotated but if the clamping die does not have enough clamping force to hold the tube tight enough then slippage occurs. Since this clamping die is so important engineers have spent decades coming up with new and different ways of handling this issue. The two most popular methods are *direct acting* and *over the center type toggle mechanisms.* Over the center type mechanisms (OCM) works in the same way as a DE-STA-CO clamp (Figure 1) [Citation C1] It uses a mechanical lever to press a gripping force onto the tube just above its center line. Using the mechanical advantage of the lever action the clamp can lock the tube into place as it is being rotated. This is a cheap and effective approach to the clamping die and has been used for many years. However even though some OCM use hydraulics or pneumatics most OCM require that their pressure be adjusted manually by the operator; usually through the use of sort of set screw. This being said the direct acting clamping die is much more common for CNC controlled systems. Direct acting clamping (DAC) is now the most common form of clamping due to advances in microprocessor control systems. In this approach some sort of directly acting piston or cylinder is acting directly on the clamping die which presses the tube firmly against the bending die. The main advantage of DAC is that you can easily use feedback systems to control the pressure of the die. The three main ways of controlling the pressure are electric, hydraulic and pneumatic. See Figure 2 [CITATION C2]. Hydraulics have historically been the most common form of applied clamping force but in recent years electric actuators have been used as a greener substitution; since they do not leak harmful fluids.


*Figure 1 (DE-STA-CO clamp)* *Figure 2 (Linear Motion Device Comparison)*

Linear electric actuators usually work through the use of a servo motor attached to a drive screw and are controlled in one of two ways; *potentiometer feedback* or *encoder feedback*. Potentiometer feedback linear actuators use a flexible wire with a proportional resistance to the distance traveled by the actuator piston. The change in resistance is read by a microprocessor as it extends or retracts and

translates to the operator and CNC program the distance traveled. This distance is then converted in the programming and is translated into clamping pressure. The potentiometer approach is simple and uses only three terminals. See figure 3 for a circuit diagram

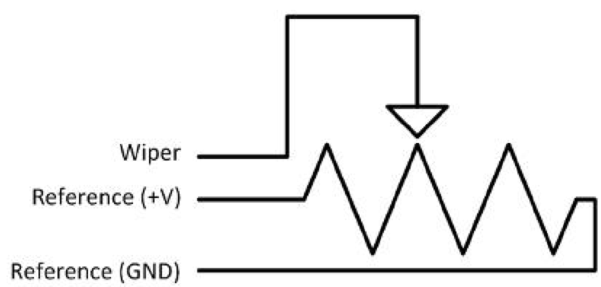



*Figure 3 (Potentiometer Linear Actuator Circuit)*

The other type of actuator is the encoder approach. This linear actuator makes use of the *Hall Effect* that works by using a sensor that detects a magnetic field. Magnets are attached to the drive shaft and as it makes rotations the sensors make note of the position and converts it to relative distance. If two Hall sensors are used then you can also account for turning direction. See figure 4.

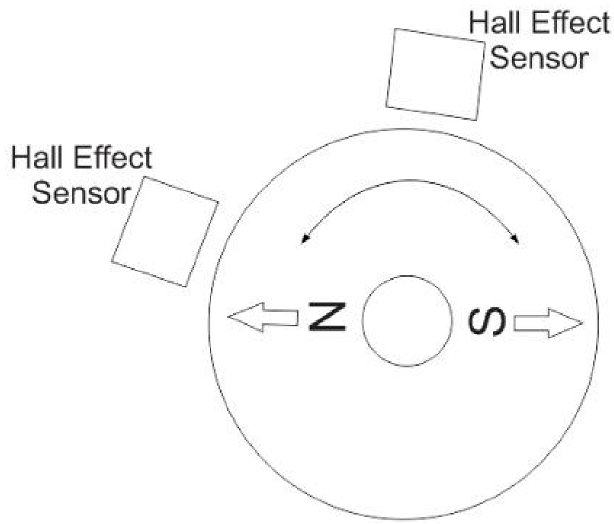
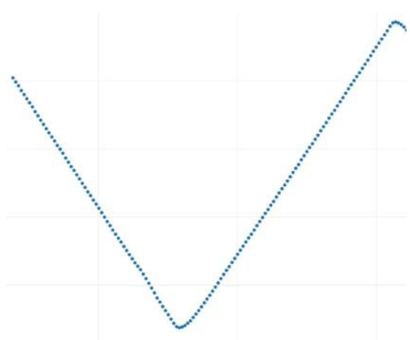
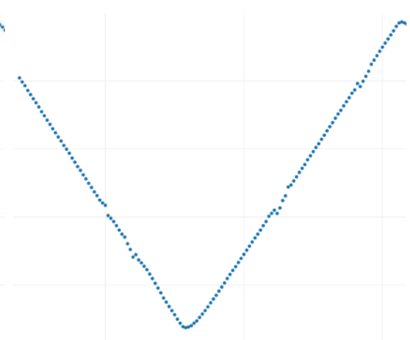



*Figure 4 (Halls Effect (Halls Effect Sensor Encoder Linear Actuator)*

There are of course pros and cons to using both approaches. A con of the potentiometer set up is that because the wiper makes physical contacts it has a limited life cycle before it starts to lose precision and eventually functionality. The Hall sensor actuator does not have any moving parts that make a connection and therefore will not lose life. You can see in Figure 5 that the potentiometer will start out in a predictable linear pattern but after 10,000 cycles it begins to lose precision [Figure 6] and after 30,000 cycles it begins to lose functionality altogether [Figure 7]. Encoders will not only live for a much longer lifetime they are also much more precise. On the other hand encoders have the con of being much more complicated to program and much more expensive making the potentiometer set up a more economical option for non-commercial use.





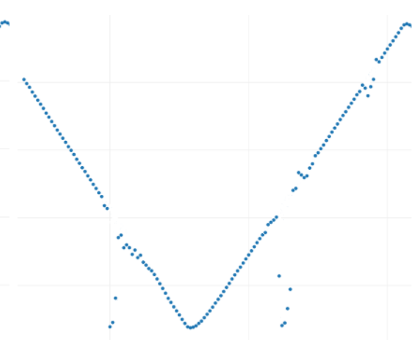


*Figure 5 (Intial Use - predictible )* *Figure 6 ( 10,000 cycles - loss of precision )* *Figure 7 ( 30,000 cycles - non-functioning )*

## Kenzie Campbell

**Coupler slip**

**Torque required for shaft**

Measuring torque applied to a shaft and the speed or distance a shaft has rotated is critical information for manufacturing applications. An example of a study on torque and speed of a a shaft uses strain gauges and zebra tape as a simplified testing method. Assuming a uniform cylindrical shape the moment of inertia would be lower than if the rotating shaft had a non-symmetric shape. A non-intrusive torque measuring system in the paper uses a non-contacting method of zebra striped tape [CITATION E1]. Optical sensors were mounted on non-rotating points to measure the rotation of the shaft.

The torque on the shaft was measured against the change in

The method of torque driving the rotation assembly for tube bending is reliant on a 5-phase geared stepper motor, and supporting the impact from the bending die action.

A 5-phase motor has similarities to a 2-phase motors, however the 5-phase will give

The rotational assembly rotates the ½ inch tube for a variation of bends. The rotator assembly is composed of a different diameter coupler attached to a stepper motor on one end and the collet shank on the other end. The assembly requires analysis of the torque on the shaft from the stepper motor and the coupler.

**CNC Application**

The increased implementation of CNC industrial machines has increased the consistency of making 3D metal parts. The tube bender has industrial applications therefore having it computer controlled makes it a reliable machine without relying on constant human interaction. The microcontrollers can be controlled with G-Code or with CAD systems. G-Code gives a machine all of its instructions, on the other hand a CAD system needs to find its operation path as well as velocities.

Kinematic modeling of translational motion produced by angular rotation of a motor can also produce inverse kinematics. Inverse kinematics start at the desired location and finding the required translational motion. Examples of inverse kinematics are common are important calculations in robotics. For robotic arms, inverse kinematics means by choosing a desired final arm orientation. Calculations can be made to find the required change in angles to achieve the desired arm orientation.

In relation to CNC machining there is an equation that relates a motors angular displacement to linear displacement [CITATION E2].

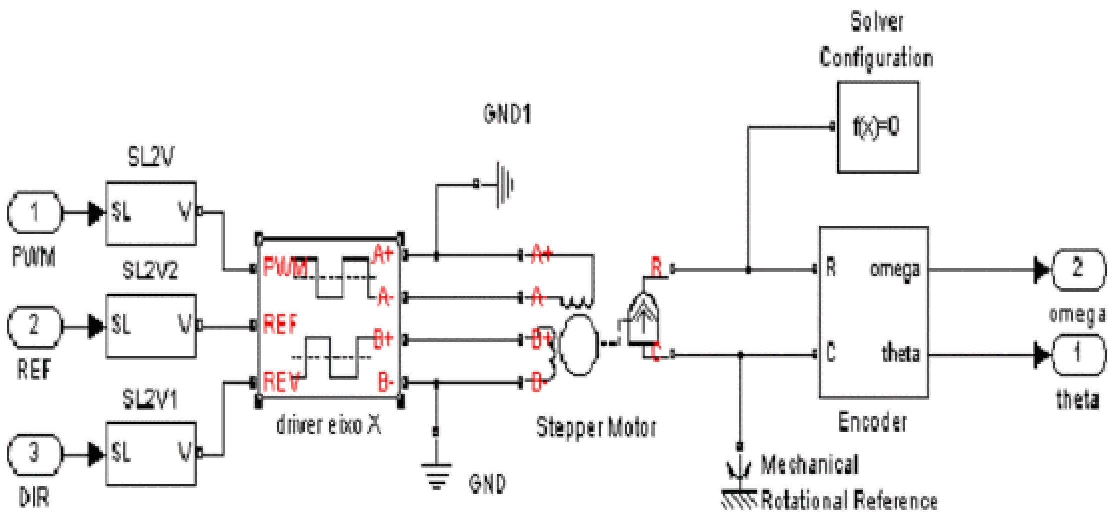
$$\vec{\mathrm{X}} = \frac{\alpha}{2\pi} \cdot \vec{\theta}, \tag{1}$$

Dynamic modeling of CNC machinery takes into account all the difficulties of moving parts such as friction and torque. The modeling relates to the control systems , requiring linearization techniques. An example of a dynamics equation for a nth degree robot is the following.

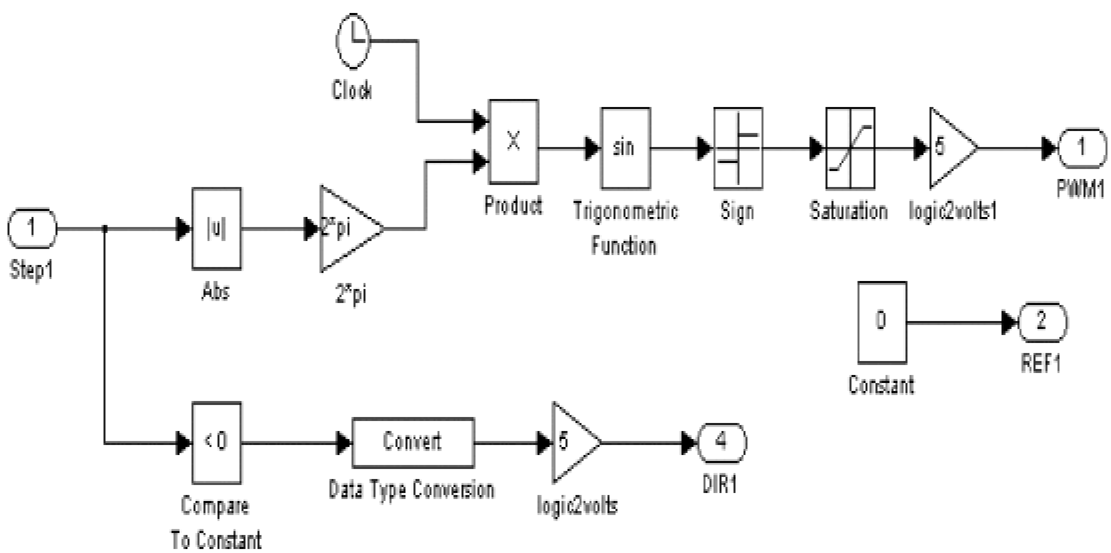
$$\tau = J\ddot{q} + B\ddot{q} + \tau_r, \tag{2}$$

Where J is the inertia, B is the viscous friction ,τis the torque and q are the coordinates.

A stepper motor moves by incremental step impulses when a voltage is applied to it. An encoder on a stepper motors shaft converts the step impulses to speed and angular position. Blocks in simulink can produce a model of the stepper motor as well as the PWM values which cause rotational motion [CITATION E1].



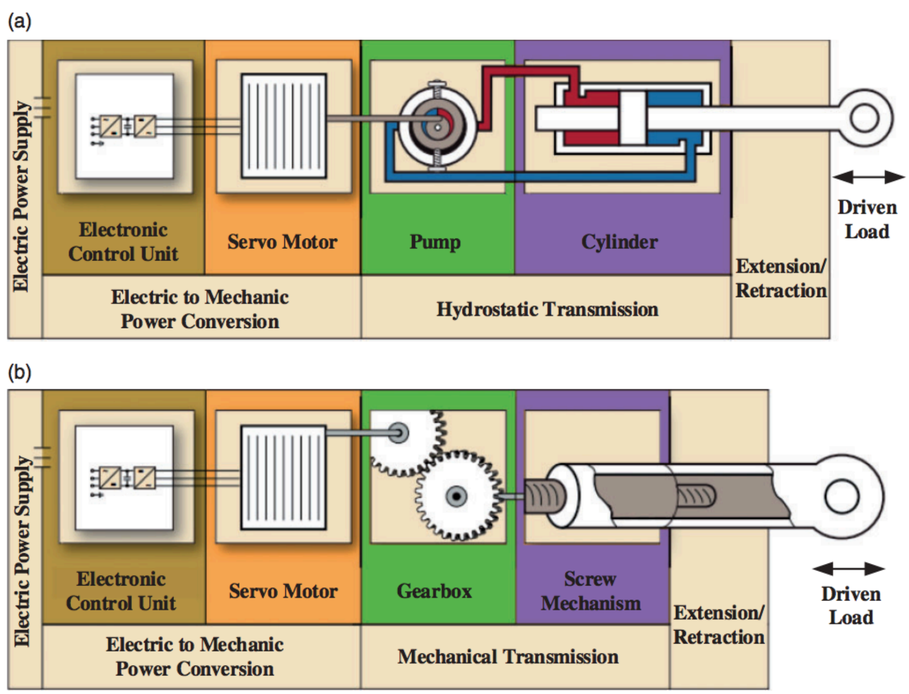

Stepper motor diagram




PWM Generator

Linear motion and rotational motion for the CNC tube bender are controlled by a central control unit. The tube bending process for this senior project will be automated with the a central controller , in order to make operations consistent and repeatable. The linear actuator component of CNC tube bender supplies the clamping force to reduce spring back. In a manufacturing setting a smart linear actuator should have the ability to change clamping settings between different materials. In the case of this senior project the correct clamping force for the ½” diamater tubes will be calibrated. Testing and calibrating the clamping actuator as well the other motors will provide data on what is sufficient force for the workpiece with out at the same time crushing it.

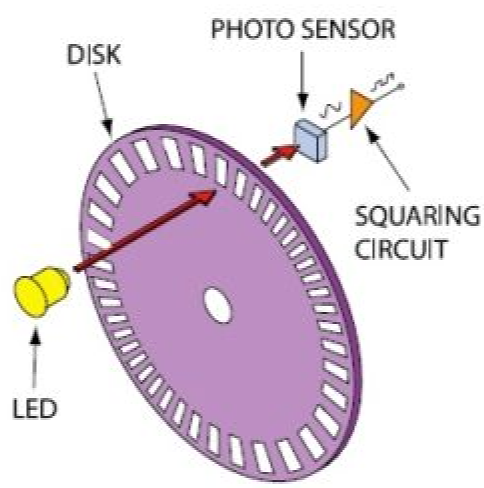
Originally the clamping actuator was designed as a hydraulic actuator. Hydraulic actuators supply a great force however the system is high maintenance and has higher operating costs than an electromechanical actuator [CITATION E3]. Electromechanical actuators are increasingly regarded as more reliable systems than hydraulics. This is due to the response time for an electromechanical actuator relies on mechanical movement from electrical supply rather than power supplied by a moving fluid.




Linear and rotary motions will be controlled by an electronic control unit, sending commands for distance to travel as well as speed. The signals sent to the motors are in the form of pulse width modulation (PWM). PWM are short pulses of regulated voltage to provide consistent power to the motors.  The pulses switch between high and low states at a fast enough rate so that the voltage appears to be constant. The speed is controlled by code commands , as well as gearing. Gearing transforms the motors high speed and low torque to low speed and high torque.

Linear drives such as the ball screw drive used to move the rotator carriage , will also be controlled by the central control unit. It is required to move to the desired bend length.

Position and velocity of the workpiece and the motors will be sent back to the ECU for a closed loop system. A closed loop system allows feedback from the tube bending system to go back to the ECU. Position of the bending die as it rotates will go back to the ECU to inform it how much further it needs to rotate for the specified bend. To measure the rotation of the drive shaft for the bending die an encoder can be mounted around the diameter of the shaft. The sensor for the encoder will be mounted in a stationary position. The information of shaft rotation will be sent to the ECU do determine length of operations. Rotary and magnetic encoders are the common approach for measuring position of a moving object.
A rotary optical encoder is a common option which consists of an LED light , a light sensor , a coded disk and a signal processor. The signal uses dark versus transparent slots to create light pulses to send to the processor. [CITATION E5]




Optical Encoder

The slots create a repeating wave form from the incident light on the photo sensor. The wave form can come in the form of a:

- Sine wave
- Square wave
- Or equally spaced pulses

Causes of error for the encoder disk would arise from light extra transmission through slots that are two wide. The encoders input to the controller will be connected to the mechanical movement of the shaft.

The line count for a desired resolution for rotary encoders.

Rotary: 360/ desired resolution in degrees = counts per revolution

Example: 360/.01 degrees = 36,000 counts per revolution

Equations to find maximum RPM, line count or maximum frequency when the two other values are known.

Rotary:

(RPM/60) *line count = maximum frequency response

(Frequency response * 60)/line count = maximum RPM

(Frequency response * 60)/RPM = maximum line count

[CITATION E6]

- Encoder for measuring how much tube is fed through for a bend attach it to the shaft connecting to the bending die motor and something else
- Encoder for tube rotation
-

**Feed-Rate**

Matching the linear motion and the angular motion of the tube is necessary to prevent any unnecessary damage to the tube while it is being bent. To do this we first have to convert the angular speed of the motor driving the bending die to a linear speed that would be needed, the equations needed include:

$$\omega = \frac{\theta}{t} \qquad v = \frac{s}{t} = \frac{r\theta}{t} = \frac{\theta}{t} . r \qquad v = \omega r$$

Where:
w (angular speed) - in radians/sec which can be converted from rpm
θ (angle/angle of bend) - in radians
t (time) - seconds
s (length) - the length at which the bend is occuring
r (radius) - radius of bending die

Once we have determined the linear speed required, we can convert the linear speed back to rpm to determine the speed the stepper motor powering the drive screw needs to spin to achieve the necessary linear speed. To do this we will need to use the equations:

$$P\ (lead) = \frac{mm}{rev} * \frac{rev}{s} = \frac{mm}{s} \qquad \frac{m}{s} * \frac{1}{P * \frac{1\ m}{1000\ mm}} = \frac{rev}{s}$$

P (lead) - in mm/rev, distance traveled per one revolution

Once we know the rpm that the motor (driving the bending die) is spinning at we can determine at what speed we need the stepper motor (that drives the drive screw) need to spin at. [CITATION E7]

# 2 FUNCTION AND CONSTRAINTS

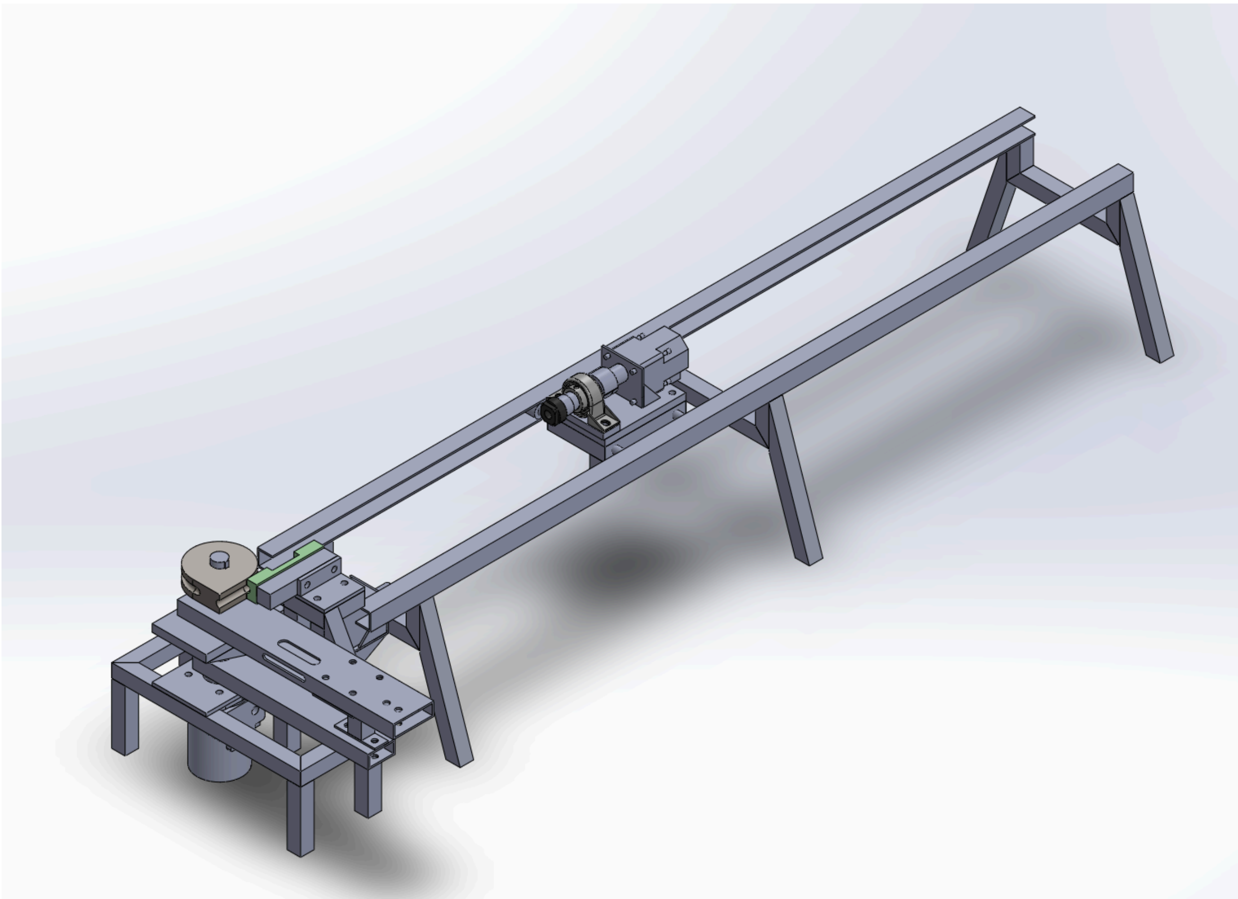
The function of the CNC Tube Bender is to bend a ½” diameter tube into a three-dimensional shape. A common application for an industrial tube bender would be electrical conduits for protecting the wiring of a structure. Electricians can use the tube bender to program their desired bend locations and angles and achieve consistent bends.

| Design Criteria | Details |
|---|---|
| Automation | Automated bending die rotation, automated clamping die, automated carriage feeding the tube into the bending die, the tube rotator.<br>Not automated are the tube rotators clamping, and the follower die. |
| Power | Type of tubing required to be bend it ½” steel, industrial applications. |
| Precision | Industrial application doesn’t require extreme precision |
| Complexity | Can bend tubes in different directions because of tube rotator |

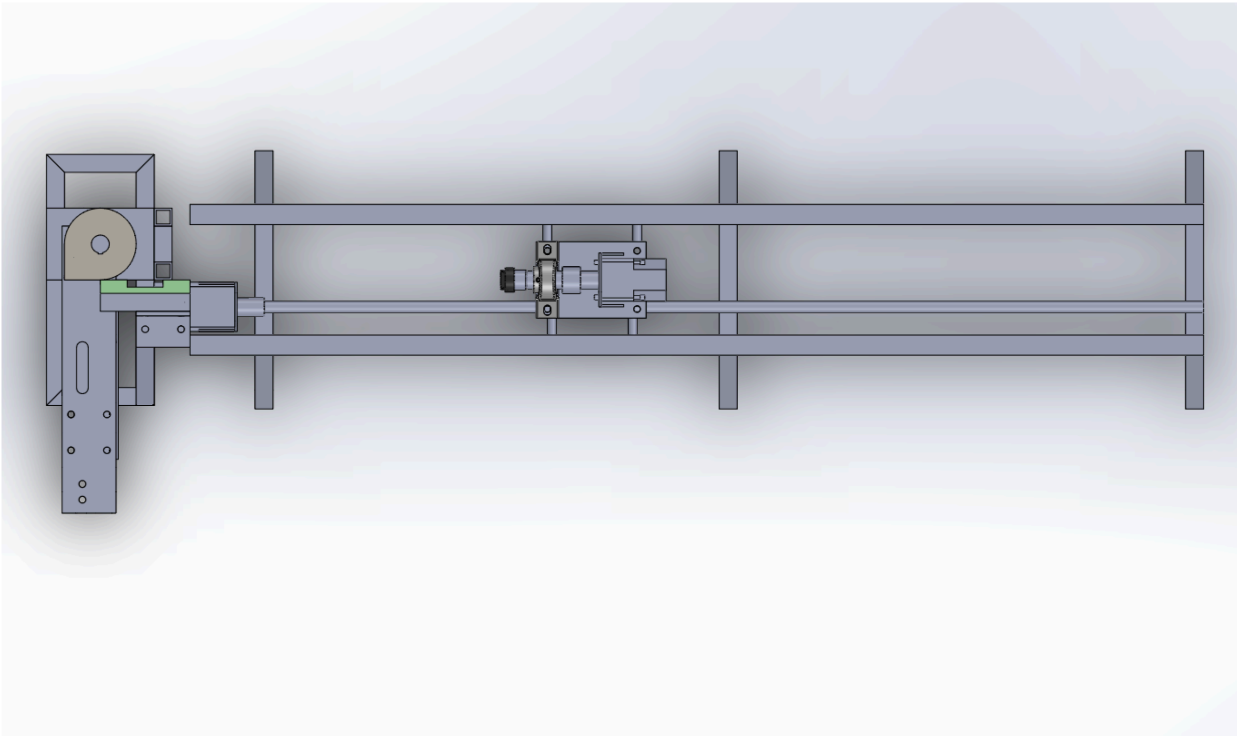
The position control system for the tube bender involves multiple moving parts moving sequentially.  All machine designs are based off of the cartesian coordinate system. The cnc tube bender will have a 4-coordinate axis system. Two rotational

# FINAL DESIGN

## Isometric Tube Bender



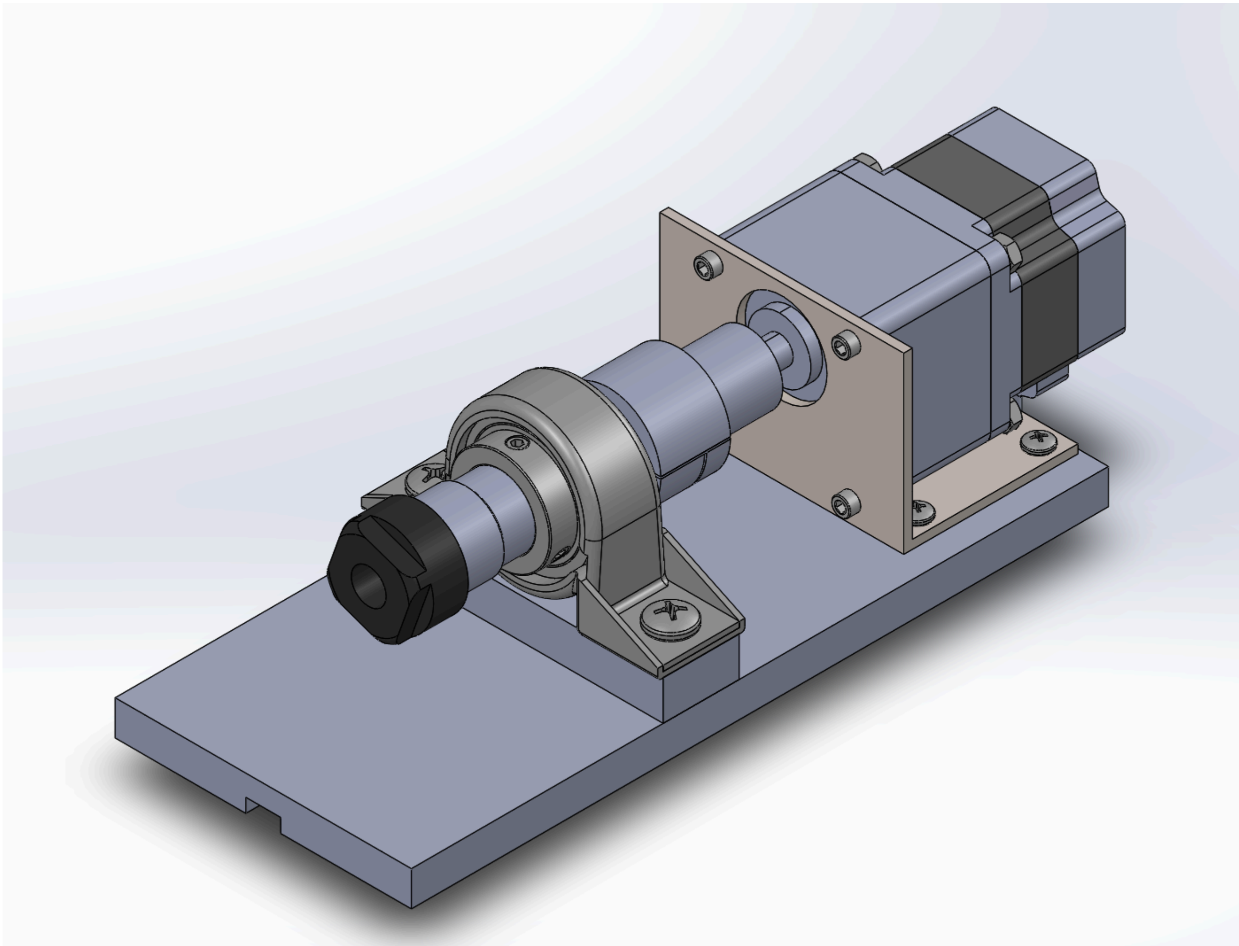
## Top View



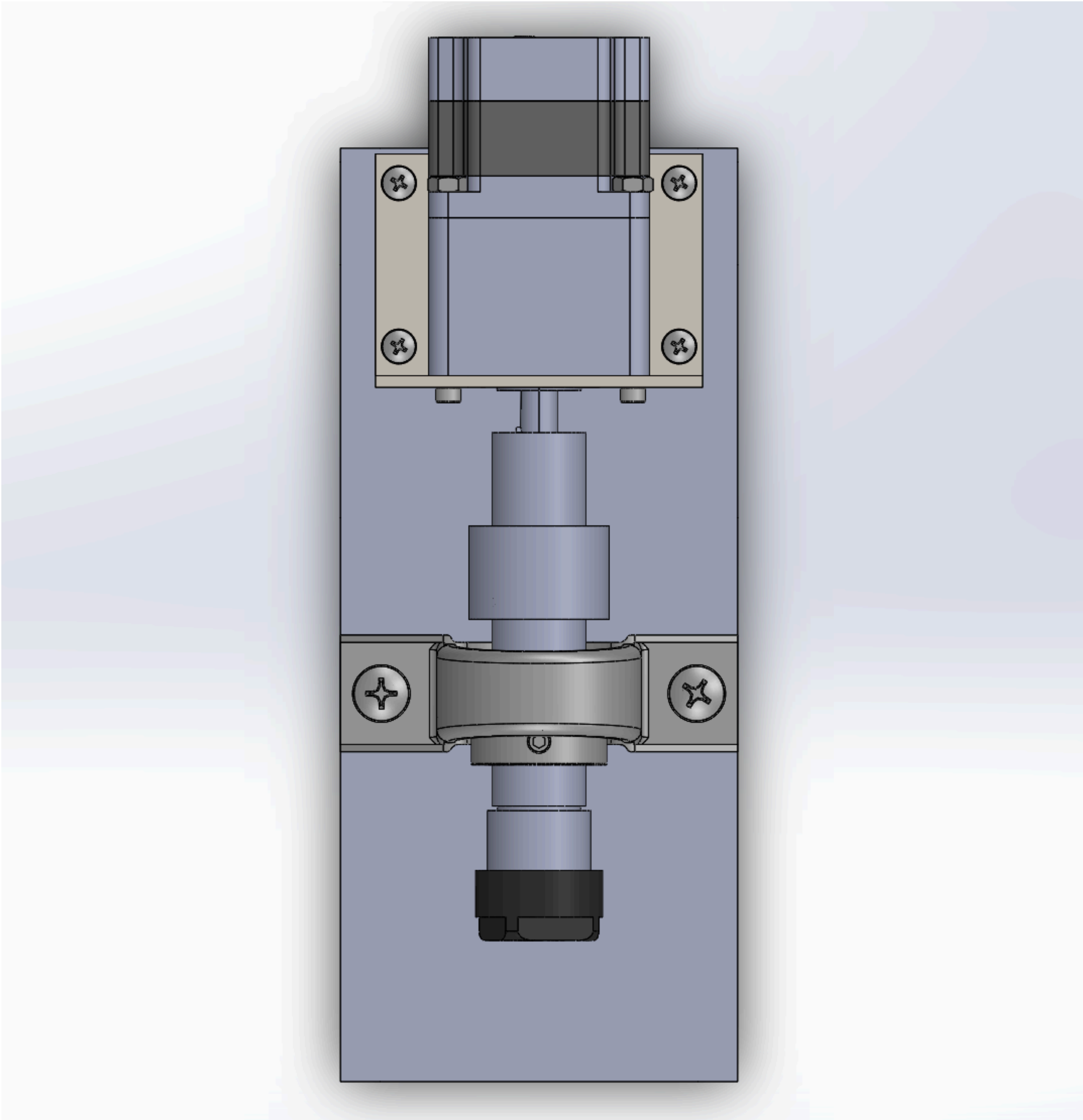
**Tube Rotator**

The rotator assembly is powered by a 5 Phase Geared Stepper Motor mounted on a bracket to the carriage. The carriage is carried forward to feed the tube into the bending die. The stepper motor is attached to a two diameter coupler. The opposing end of the coupler is attached to the collet shank. The collet shank contains a collet which tightens down onto the workpiece (½” diameter tube).Supporting the collet shank and its ability to rotate is the ball bearing.

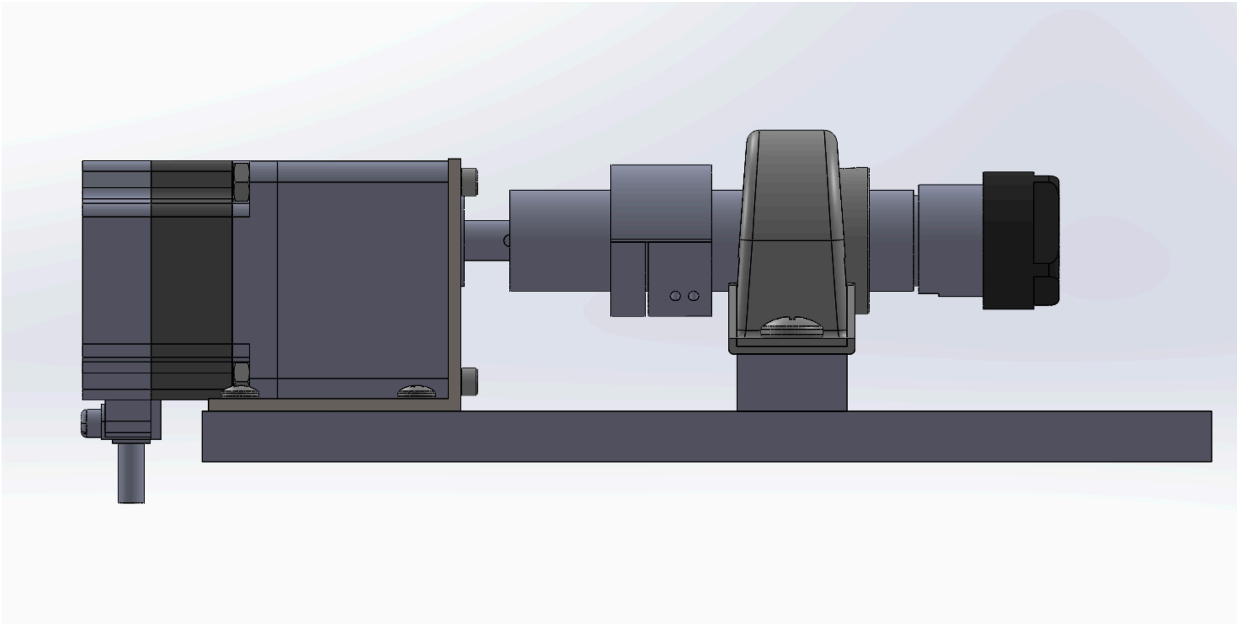
**Isometric View**



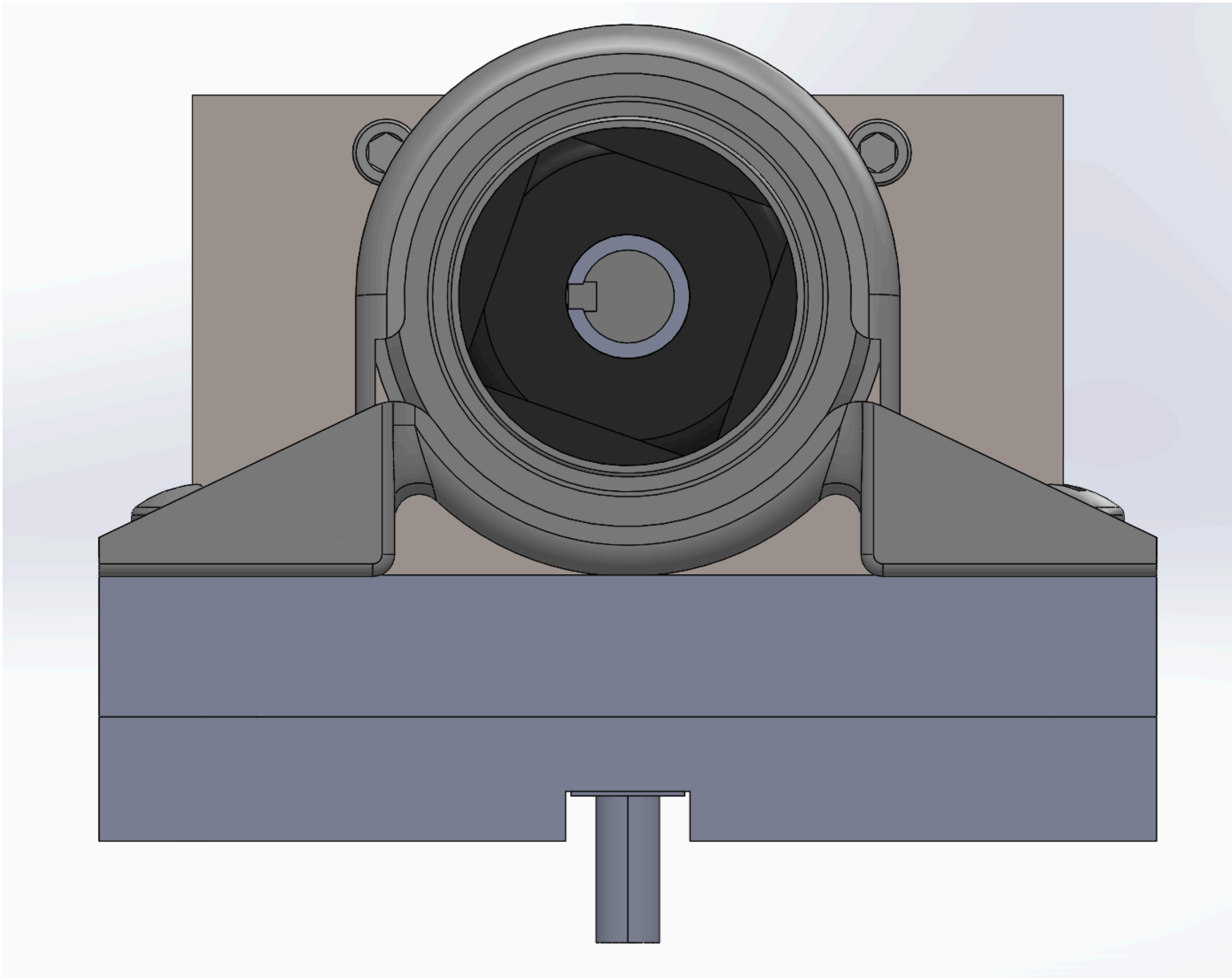
**Top View**



**Side View**



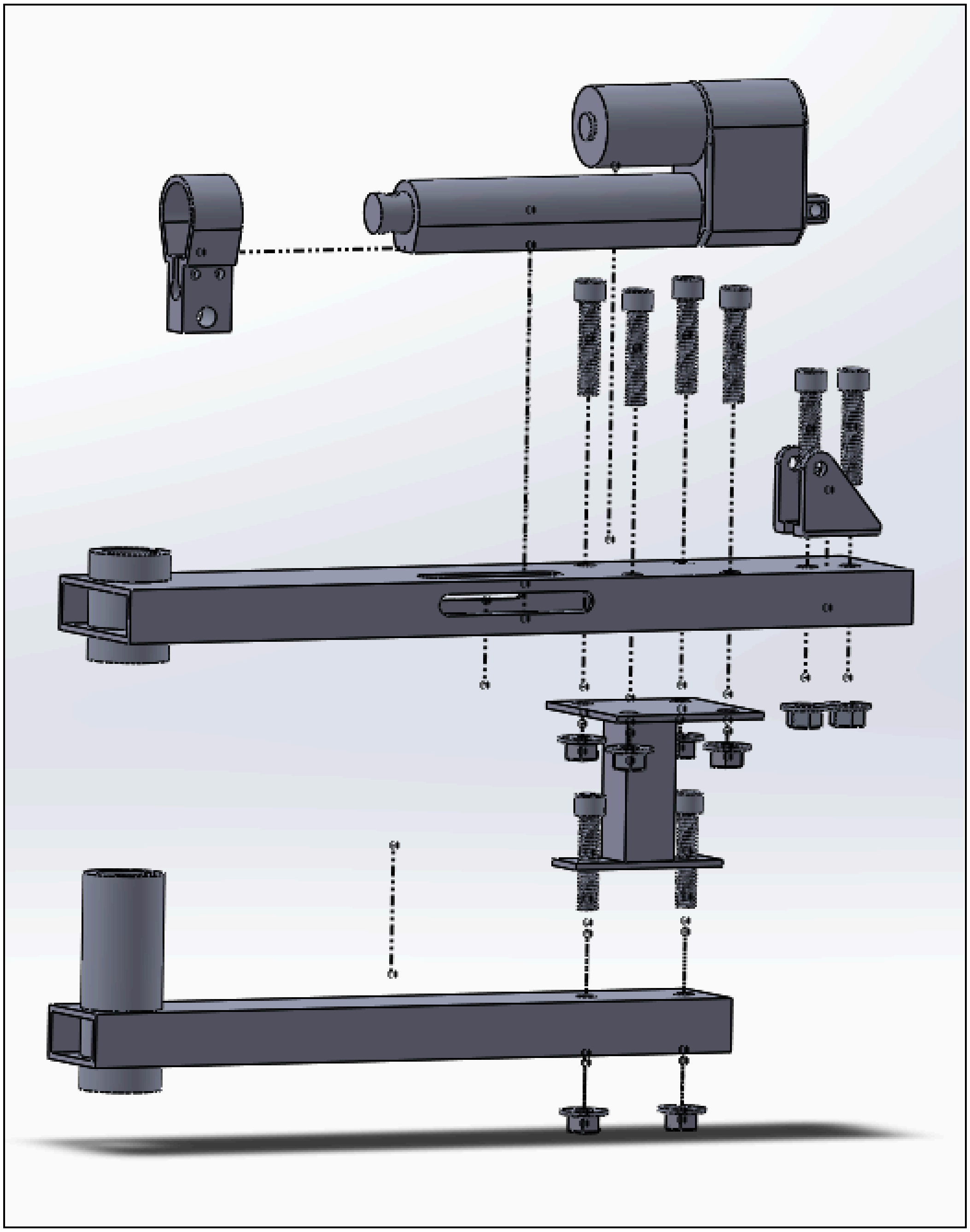
**Front View**



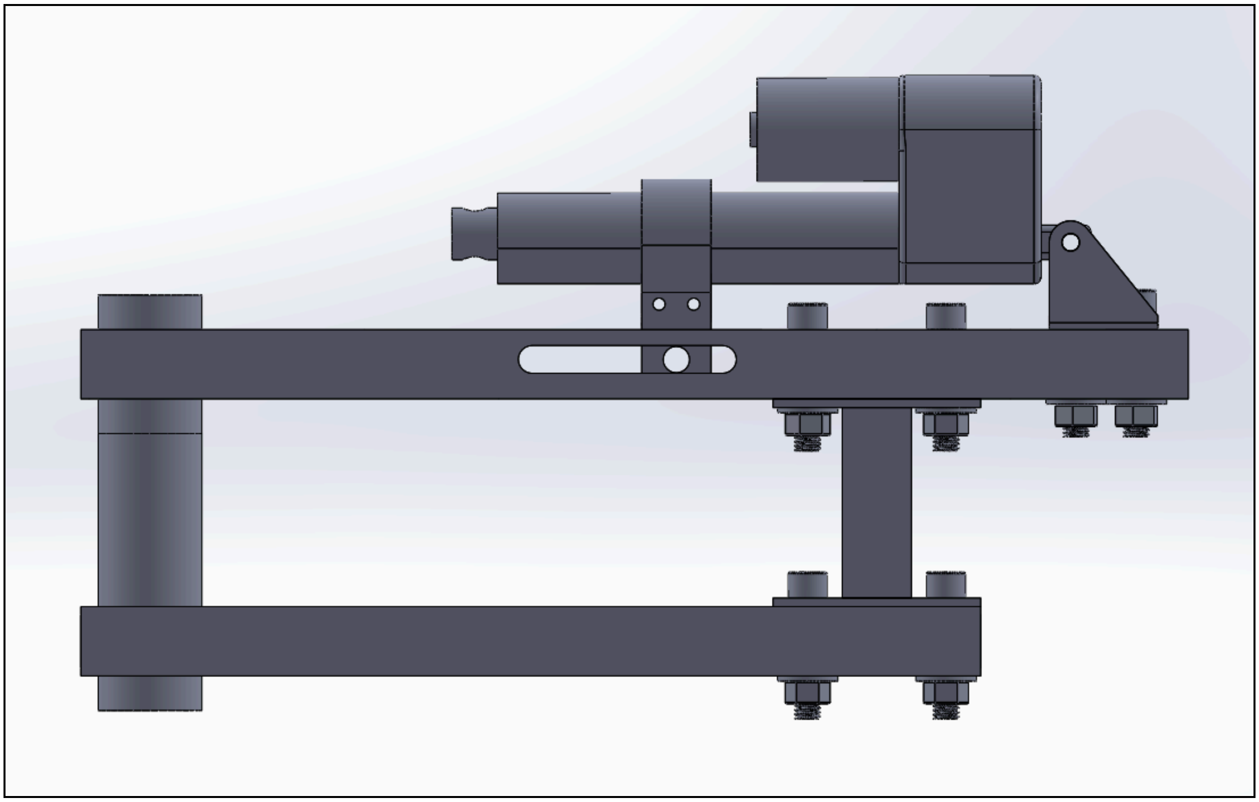
**Actuator Support**

An electric linear actuator capable of 200 pounds of pressure and 2 inch stroke, will be used to clamp and hold the tube specimen in place during the bend. The support arms utilize precision milled square tube stock, and machined mounts to fit on the shaft from the hydraulic motor. 1x1 inch square tubing will be used to support the two components of the support.

## Exploded view



**Front View**



# 4 ANALYSIS

Thomas W.C. Carlson

Front end analysis

The analysis procedure for ensuring the machine is capable of deforming the beam is done in the following steps:

1. Calculation of the force P required to deform the tube
2. Calculation of the reaction force R1 and the pressure die force F2 necessary to hold the tube in place
3. Calculation of the motor torque necessary to produce force F1 at the shaft
4. Comparison of required values and selected motors, actuators

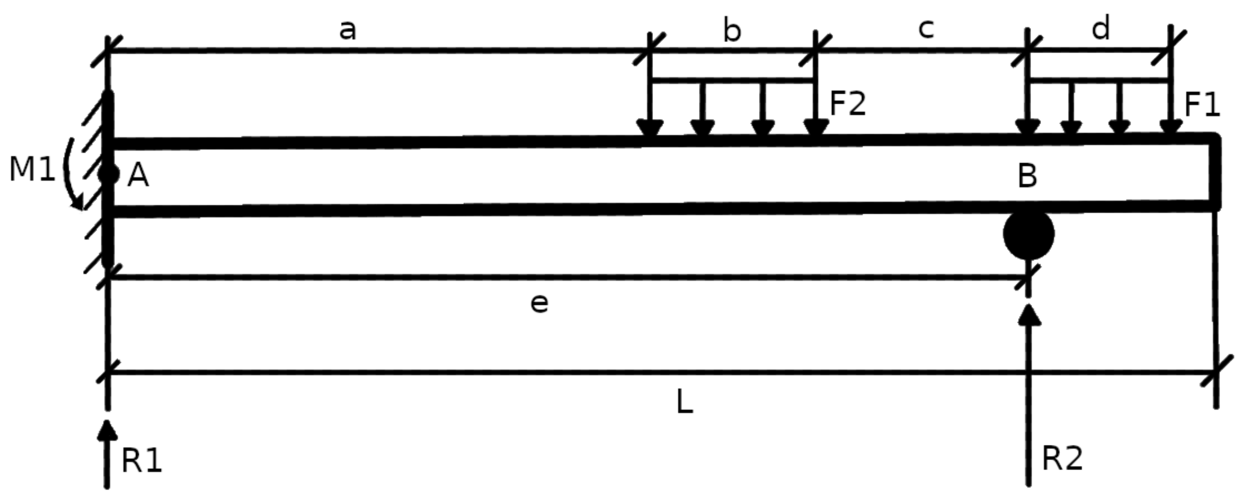



Figure A1 - Tube represented as a beam

## 4.1 Calculation of required force P

Begin by assuming that **c = 0**, such that the pressure die is adjacent to the clamp die and lines up with the bend die, which acts as a roller support. Assume also that we desire **R1 = 0** and **M1 = 0** to avoid stressing the carrier assembly. By the assumption of **c = 0**, we create a cantilever beam fixed at point **B**.

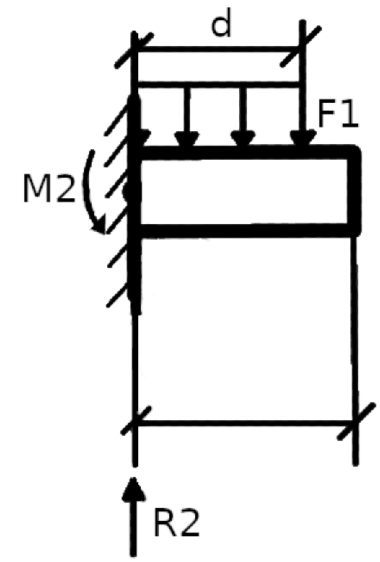



Figure A2 - The cantilevered section of the beam

Using beam bending theory equations and the second area of inertia of the cross-section of the tube, we can determine the required moment **M2** using the yield stress of the material:

$$I = \frac{\pi}{4}(R_o^4 - R_i^4) \quad (1) \qquad\qquad \frac{My}{I} = \sigma_y \Rightarrow M = \frac{\sigma_y I}{(R_o)} \quad (2)$$

Using Figure A2, we can apply static equilibrium equations from beam theory to determine the value of **F1** for the onset of deformation. Substituting the result into (2) yields

$$\frac{\sigma_y I}{(R_o)} = \frac{d}{2}(F_1 d) \Rightarrow F_1 = \frac{2\sigma_y I}{R_o d^2} \quad (3)$$

**F1** is a distributed force which can be determined by the force **P** of the actuator divided over the clamping length **d**. Substituting this relation into (3) yields

$$(\frac{P}{d}) = \frac{2\sigma_y I}{R_o d^2} \Rightarrow P = \frac{2\sigma_y I}{R_o d} \quad (4)$$

**4.2 Calculation of Reaction forces**

With the assumption of **R1, M1 = 0**, the application of force and moment balance equations yields a system of 2 equations with 2 unknowns so long as **F1** is treated as an input value calculated from (4):

$$\Sigma F = -F_2 b - F_1 d + R_2 + \cancel{R_1} = 0 \quad (5)$$

$$R_1 - F_2 b = F_1 d \quad (6)$$

$$\Sigma M_A = -(a + \frac{b}{2})(F_2 b) - (e + \frac{d}{2})(F_1 d) + eR_2 + \cancel{M_1} \quad (7)$$

$$eR_2 - (a + \frac{b}{2})(F_2 b) = (e + \frac{d}{2})(F_1 d) \quad (8)$$

This system can be represented using the matrix

$$\begin{bmatrix} 1 & -b \\ e & -b(a + \frac{b}{2}) \end{bmatrix} \begin{bmatrix} R_1 \\ F_2 \end{bmatrix} = \begin{bmatrix} F_1 d \\ (F_1 d)(e + \frac{d}{2}) \end{bmatrix}$$

Performing gaussian elimination yields the algebraic expressions for values of **R1** and **F2**

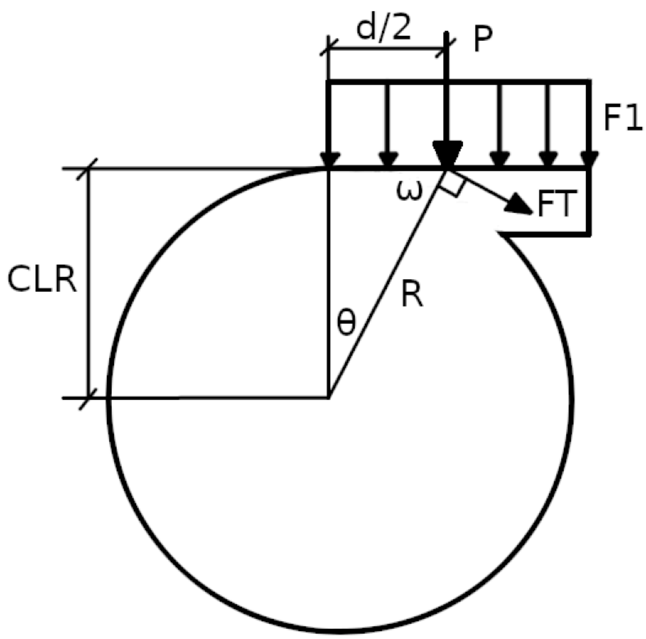
$$R_1 = F_1 d - \frac{F_1 d^2}{2a + b - 2e} \quad (9)$$

$$F_2 = \frac{F_1 de}{(a + \frac{b}{2} - e)(b)} - \frac{F_1 d(e + \frac{d}{2})}{b(a + \frac{b}{2})} - \frac{F_1 de(e + \frac{d}{2})}{(a + \frac{b}{2} - e)(b)(a + \frac{b}{2})} \quad (10)$$

These expressions are large and cumbersome, but provide the ability to script calculations using MATLAB, allowing for an array of inputs and optimization by iteration. Of particular interest are variations of **F1**, **b**, and **d**.

### 4.3 Calculation of Required Torque

The distributed force **F1** can be reduced to a single force **P**, whose value was determined in 4.1. It's location is a distance **d/2** from point **B**. This x-axis distance therefore means that only a fraction of the torque **T** produced by the rotation of the bending shaft is effective in bending the tube. The geometry is illustrated in Figure A3.




$$\theta = \arctan\left(\frac{\frac{d}{2}}{CLR}\right) \quad (11)$$

$$\omega = \arctan\left(\frac{CLR}{\frac{d}{2}}\right) \quad (12)$$

$$P = F_T \sin\theta = F_T \cos\omega \quad (13)$$

$$R = \sqrt{CLR^2 + (\frac{d}{2})^2} \quad (14)$$

$$F_T = \frac{P}{\sin\theta} = \frac{P}{\cos\omega} \quad (15)$$

Figure A3 - Bending torque arm geometry

Equations (11)-(15) can be used to find the motor torque necessary to deform the beam:

$$T = F_T R = \frac{P}{\sin\theta}\sqrt{CLR^2 + (\frac{d}{2})^2} \quad (16)$$

**4.4 Comparison of Calculations and Selected Parts**

For a 0.5” OD tube made of AISI 304 Stainless Steel with 0.46” ID 3’ in length to be bent around a CLR of 2”, the following values were selected for calculation:

| | |
|---|---|
| L - Length of tube | 36 in |
| e - Tube length from collet to bend die | 34 in |
| d - Clamp die length | 2 in |
| c - Distance between clamp and pressure die | 0 in |
| b - Pressure die length | 6 in |
| a - Distance between collet and pressure die | 28 in |
| Ro - Outer radius of the tube | 0.25 in |
| Ri - Inner radius of the tube | 0.23 in |

| | |
|---|---|
| **$\sigma_y$** - Yield strength of tube | 31200 psi |
| CLR - Centerline radius of bend die | 2 in |

Following the procedure from 4.1, using equations (1) and (4),

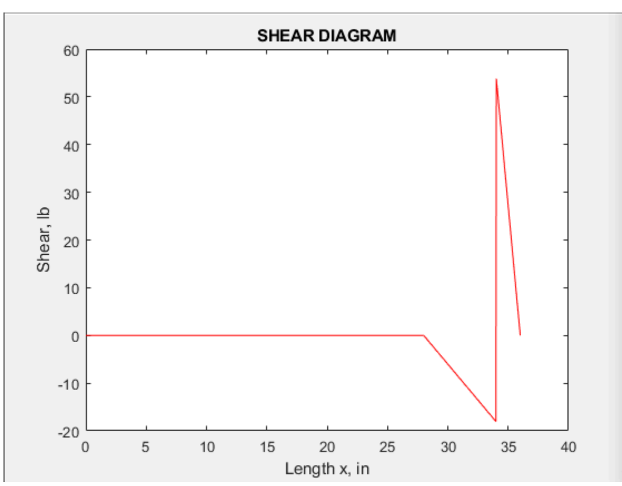
| | |
|---|---|
| I - Moment of inertia of the tube | 8.7010e-4 in^4 |
| P - Clamp die force | 54.3 lb |

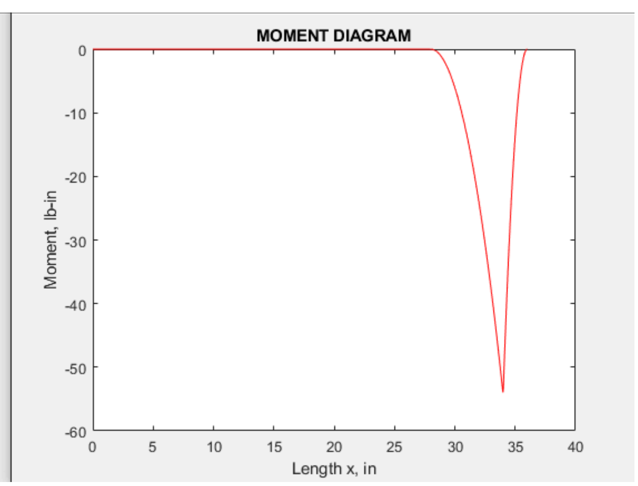
P is the minimum force that the actuator must be able to deliver in order to ensure that the clamp die stays pressed against the bend die. Without this force, the clamp die will not deform the beam when it is rotated by the bending torque.

Continuing with the procedure from 4.2, using equations (9) and (10),

| | |
|---|---|
| R2 - Bend die reaction force | 72.4 lb |
| F2 - Pressure die distributed force | 3.02 lb/in |
| P2 - Pressure die force | 18.1 lb |

We can see from this that the force required of the pressure die is not large, and it is best to value its extension precision to reduce any effect of friction and avoid damaging the tube during operation. Insertion of these values into singularity equations provides shear and bending moment diagrams:






Continuing with the procedure from 4.3, using equation (16)

| | |
|---|---|
| T - Motor torque required | 271.5 lb-in |
| T * 4 | 1086 lb-in |

This is only the torque required to deform the tube--it does not account for the inertia of the assembly or frictional forces. Applying the safety factor of four compensates for frictional losses of efficiency. The MATLAB script used to calculate these values is provided in APPENDIX.

## Tube Rotation Analysis

### 1. Moment of Inertia

The stepper motors desired torque is dependent on the tubes moment of inertia. The stepper motor will require a force high enough to move the load while overcoming the loads resistance. An object's moment of inertia is dependent on the size and distribution of mass around its geometric center. The overall tube rotation assembly should have a small moment of inertia; the components consist of the collet , collet shank , coupler, all holding on to the tube getting bent. The tube can be analyzed as rotating about its cylindrical axis (z-axis).

$I_z = 1/2M(R_1^2 - R_2^2)$

$M = 2lb\ mass\ of\ the\ tube$

$R_1 = 0.0127\ m$

$R_2 = 0.00952\ m$

$I_z = 3.318\ x\ 10^{-7} N * m$

### 2. Torque for Tube Rotation

$\tau = I_z \alpha$

Finding α with Kinematics Equation

$\theta = \theta_o + \omega_o t + \frac{1}{2}\alpha t^2$

Assumptions:

Rotate the tube 120 degrees in less than 10s

No initial angle ($\theta_o$)

No initial $\omega_o$

$\omega_f = 100rpm = 10.47\, rad/s = 600 deg/s$

$\alpha = 0.0418\, rad/s$

$\tau = 4.97\, x\, 10^{-8}\, N * m$

$Power\ (P) = \tau\omega = 6.98\, x\, 10^{-10} Hp$

**Shaft Coupler Design**

Useful transmission designs have no change in diameter over the length of a part. Key elements secure shaft connections and help transmit torque.

Power transmitted through the shaft relates torque and angular velocity

$P = T\omega$ (9.1a)

Multiaxial stresses occur on a shaft experiencing bending and torque

Max Shear Torsional Stress

$$\tau_{max} = \frac{TDo}{2J}$$

$$J = \frac{\pi}{32}(Do^4 - Di^4)$$ (Polar Second Moment of Inertia)

Coupler sizing should be chosen so stress does not exceed 10% of the ultimate tensile strength of the material.

**Manufacturing Plan**

The 0.5 in bending die and clamp will be produced via CNC precision milling processes. The grooves for the die and clamp will be machined using a slot milling procedure.

All the shaft mounts and the main shaft will be machined using a lathe and broached to accommodate the needed keyway.

The actuator supports will be cut to length from square tubing using a band saw. Precision milling will be used to manufacture the mounting points for brackets.

The pressure die, mounting plates, and components of the actuator support will be manufactured from plate steel of the appropriate thickness.

The structure will be made of 1x1 in. square tubing and welded to form a solid foundation.

The pressure die will be precision milled and tapped to secure it to the structure, as well as secure the low friction plastic pressure die to the steel base of the die.

## 16 Week Timeline

**Week 1**

Manufacturing begins; Parts ordered

1x1 structural tubing cut and welded

**Week 2**

Start programming the code to run the CNC machine,

Continue Structure manufacturing,

Mounting plates manufactured and added

Actuator support components manufacturing

**Week 3**

Structure is done. Die, clamp and pressure die CNC. Actuator support assembly.

Axel and shafts manufacturing

**Week 4**

Front of Tube bender Completed.

Manufacture carrier components. Mount drive screw motor and supports.

**Week 5**

Rail system mounted. Carrier assembled.

Rotator mount components manufactured and assembled.

**Week 6**

Mount the rotator on the carrier and carrier on the rails and drive screw. Begin alignment and mount pressure die.

**Week 7**

Final alignment and last minute adjustments.

**Week 8**

Final assembly complete, all errors addressed, testing can commence

**Week 9**

Begin testing the necessary force that the clamp and follower would need to apply on the tube and the linear actuator to calibrate them to achieve the necessary results.

**Week 10**

Continue testing the rotary position sensor to test the position of motor and calculating the stepper motor rpm needed for the drive screw to achieve a linear speed that matches the rotating bending speed

**Week 11**

Continue testing the functions of the carrier (rotating the tube and feeding the tube to the bending die).

**Week 12**

Finish all testing, programming and begin the report

**Week 13**

Start report

**Week 14**

Continue working on the report

**Week 15**

Complete the report

**Week 16**

Complete the project and prepare for presentation

Vendor Quotes are required for all items listed. Save all packing slips and reciepts for reconciliation Total budget shall not exceed $1,000,000.00 All purchases must be approved Reimbursements are not possible through the fo

| Budget Edit Date | 11/19/2019 | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Item # | Category | Part # | Vendor | Vendor Part number | Vendor Info | Quote # | Unit Cost | Quantity | Cost |
| 1 | Rotator | TB5-BEA-001 | Oriental Motor | PKE564AC-TS10 | 5 Phase Stepper Motor | | 251 | 1 | 251 |
| 2 | Rotator | TB5-BEA-002 | Toolots | BA0103018 | ER20 Collet Shank | | 21.32 | 1 | 21.32 |
| 3 | Rotator | TB5-PUL-003 | Toolots | BA0304040 | End Mill Tool Holder | | 31.03 | 1 | 31.03 |
| 4 | Rotator | TB5-MOU-004 | Oriental Motor | SOL2M4 | Steppor Motor Mount | | 27 | 1 | 27 |
| 5 | Rotator | TB5-BEA-005 | McMaster-Carr | 5913K64 | Ball Bearing | | 12.69 | 1 | 12.69 |
| 6 | Rotator | TB5-STU-006 | McMaster-Carr | 9119K111 | Mount Stud | | $2.73 | 8 | $21.84 |
| 7 | Rotator | TB5-SCR-008 | McMaster-Carr | 90087A151 | Cutting Screw | | $8.31 | 4 | $33.24 |
| 8 | Rotator | TB5-BAR-009 | McMaster-Carr | 8910K821 | Low Carbon Steel Bar | | $16.66 | 1 | $16.66 |
| 9 | | | McMaster-Carr | 9069T1 | Linear Ball bearing | | $45.32 | 1 | $45.32 |
| 10 | | | | | | | | | |
| 11 | | | | | | | | | |
| 12 | | | | | | | | | |
| 13 | Pressure Die | TB3-004 | Professional Plastics | 318189 | UHMW Raw Material 8"x2.4"x2.4" | | $19.95 | 1 | $19.95 |
| 14 | Pressure Die | TB3-003 | McMaster-Carr | 6552K43 | 4140 Alloy Steel Bar 2"x2"x3" | | $49.05 | 1 | $49.05 |
| 15 | Pressure Die | TB3-001/TB3-002 | McMaster-Carr | 1388K15 | 4140 Alloy Sheet 1/16"x6"x6" | | $24.27 | 2 | $48.54 |
| 16 | Pressure Die | TB3-005 | McMaster-Carr | 98306A231 | 1045 Clevis Pin 7/16"-dia 3" long | | $12.49 | 1 | $12.49 |
| 17 | Pressure Die | TB3-006 | Grainger | S60300HC15 | 3/8"-16, Hex Head 1-1/2"L Bolts | | $3.89 | 4 | $15.56 |
| 18 | Pressure Die | TB3-007 | FastnerSuperStore | 185583 | 3/8" x 7/8" Dowel Pins | | $0.50 | 2 | $1.00 |
| | | | | | | | | | |
| 19 | Drive Screw set | TB2-DRSRW-01 | VXB Ball Bearings | | 16mm 1350mm-BALLSCREW-SET | | $156.37 | 1 | $156.37 |
| 20 | Caster wheels | TB2-CasWhl-02 | Zoro | G4808447 | 3/8 in. Bore Dia., 275 lb. | | $13.00 | 4 | $52.00 |
| 21 | Carrier | TB2-AlSHT-03 | McMaster-Carr | 9040K71 | 1/4" Thick, 12" x 12" Al Sheet | | $92.82 | 1 | $92.82 |
| 22 | Carrier | TB2-Rod-04 | McMaster-Carr | 8890K254 | .397", 3' long steel rod | | $9.77 | 1 | $9.77 |
| 23 | Stepper Motor | TB2-STPMOTOR | McMaster-Carr | 6627T56 | 600 rpm, 22 in.-oz. Torque | | $157.66 | 1 | $157.66 |
| 24 | | | | | | | | | |
| 25 | | | | | | | | | |
| 26 | | | | | | | | | |
| 27 | | | | | | | | | |
| 28 | | | | | | | | | |
| 29 | Bending Die | TB2-BDE-003 | Arc-Zone Welding | TLS-VS500-003 | | | $50.00 | 1 | $50.00 |
| 30 | Bending Motor | TB2-MOT-001 | Motion Industries | 101-2019-009 | Hydraulic Gerotor Spool Valve Motor - H-Series | | $530.34 | 1 | $530.34 |
| 31 | | | | | | | | | |
| 32 | | | | | | | | | |
| 33 | | | | | | | | | |
| 34 | | | | | | | | | |
| 35 | | | | | | | | | |
| | | | | | | | | Total | 1655.65 |

# 1 TESTING PLAN

A variety of testing will need to be done on the prototype when it is being built. As engineers we use theoretical analysis to examine different points of possible failure but it is usually a rough estimate at best. In order to learn how our machine is performing we will need to do in shop testing. A major point of concern is the stresses and strains in the tube as it is being bent. We plan to use strain gauges and strain gauge rosettes methodically placed along the tube. We can use the strain information to learn the true stresses acting on the tube as well as be able to learn more about possible crushing and wrinkling in the tube. Another point of importance that will be tested is the clamping die. We have decided to downscale from a hydraulic piston to an electric actuator because the theoretical analysis told us that the necessary stress required to clamp the tube was only about 60 lbs. We need to test to make sure this pressure is suffice enough to meet our standards. For the bending die we intend to be able to make 180 degree bends at the most but we will have to test whether the pipe would hit or obstruct the machine from continuing to work. Another thing we will have to test for is the tube springback, since we are not able to calculate how much force we would need the follower block to apply on the tube as it is being bent. We will need to do a few tests where we estimate the force needed for the follower block to apply and adjust it accordingly until we are able to make the follower block press with enough force on the tube without letting it springback and without causing any external damage on the tube as a result of that force. Feed rate of the pipe is another concern that we will be testing. We will already know the rpm of the motors that we will be using from the distributor, but we will have to match the bending speed of the bending die to the linear speed of the drive screw. To do this we will have to figure out the rpm of the motor when it is bending a pipe and translate that into a linear speed for the carrier to match the bending speed, then we will have to convert the linear speed into an rpm that the stepper motor would need to rotate for the drive screw to match the linear speed required. Another issue we will have to test is when bending a tube to a certain angle. To do this we decided that we will have to add something to the bending die, it will either be a laser of sorts or a tool that will be able to tell us the angle that the tube was bent. Then we will have to program the machine to be able to bend the tube to any angle we need, then we will test how accurate the machine is at bending a tube and if it will be consistent for multiple bends.

# 2 DISCUSSION, CONCLUSION AND RECOMMENDATIONS

After extensive planning and theoretical analysis on our current assembly model we have determined that there is sufficient evidence to safely move forward with the manufacturing of our CNC tube bender. The major focus of analysis in the beginning was on the actual tube in order to find out how much torque would be required to bend it. A theoretical analysis was done on a 3 ft piece of tubing with an outer diameter of 0.25 inches and an inner diameter of 0.23 inches. The working length from the collect to the bend die was assumed to be 34 inches and following the analysis previously stated we learned that a torque of 271.5 in-lb was needed. With an applied safety factor of 4 we decided that a hydraulic motor would be best for us. After the tube analysis was done and the motor was chosen there were still a few major points of concern with regards to the machines ability to perform safely and efficiently. The first big concern was the shaft that connected the hydraulic motor to our bending die. Our design used keys and keyways to connect the shaft to the bend die and a coupler to connect the shaft to the motor. We assumed a 1” shaft diameter and then tested it using matlab. Matlab showed us that the minimum shaft diameter needed was 0.73” which was less than our 1” assumption. We also used matlab to analyze the sheer forces on a 0.25” x 0.25” square keyway and achieved a minimum safety factor of 4 or more on each key.

After the motor and bend die sub assembly was dealt with we began to focus on the rear end of the machine which had the rotator assembly and its carrier. We decided early that the rotator would be controlled using a stepper motor and driven forward using a drive screw and another stepper motor but we weren't sure about how to connect all of this to the tube. Initially we wanted the ability to bend any length of tubing and keep it fully automated but this would have required the use of two very expensive pneumatic chucks as well as their subsystems. This alone added over $2,000 to our total cost. This forced us to downsized to a single manual chuck system that would be set by the operator during the initial setup. This limited the total length of tubing that we could bend to 6ft and added one more job for the operator. This was a hard decision but was necessary if we were to save at least $1800. This still left us with the issue of connecting the chuck to the stepper motor. We considered using belts, chains, gears and drive screws but in the end we decided it would be simpler to go with a collet instead of a chuck. The collet would add the benefit of being even cheaper than the chuck as well as being easily integrated with the stepper motor.

Another section of the rear end that was changed numerous times was the guiding system for the rotator assembly. We designed the rotator to ride on a carrier that would allow it to move forward and backward while being moved by the drive screw. After many iterations and pricing different parts we decided that it would be cheaper to create our own guide system by tapping a set of cam follower bearings into a custom carrier. The cam followers would ride inside of a steel C-channel that we will also custom make. By making these parts ourselves we could potentially save another $200 to $400 but will more than likely run into issues of friction and accuracy during the manufacturing process that will need to be addressed during testing.

Another major concern was the clamping die pressure required to hold the tubing in place as it is being bent. Through analysis it was determined that the clamping die must have a minimum of 54.3 lb of force applied to it in order to properly perform. Before analysis was complete it was assumed that hydraulics would be required but instead we found that the necessary force was easily attainable through electric actuators. We found affordable actuators that ranged from 10 to

300 lb of pressure with precision and were easily integrated into our Arduino microcontroller. This was a major last minute change to our project when we decided to switch from a hydraulic piston system to the electric alternative. We found that this saved us about 16 lbs of weight on our rotating arm which reduced rotational inertia that the bending motor and shaft have to overcome. We also saved about $40 just switching from hydraulic piston to the actuator which does not account for the extra savings achieved by not buying all of the hydraulic subsystem components which could easily exceed a few hundred dollars.

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