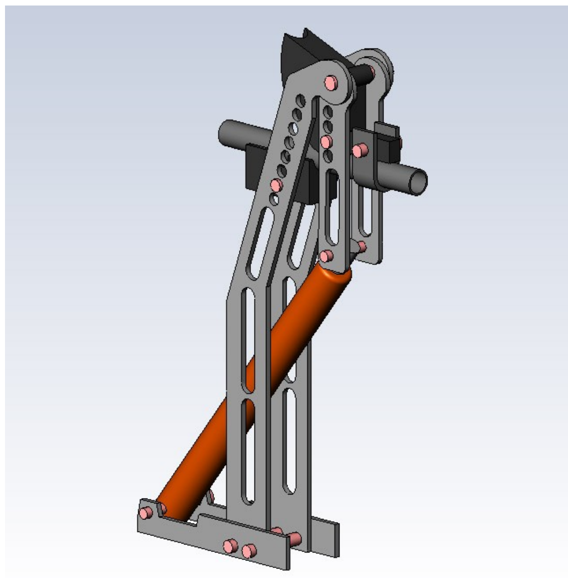
Tube Bender Analysis and Redesign



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Abstract

This project involves the analysis and redesign of a sub-industrial model tube bender. Motivated by a personal purchase, the analysis portion was performed to ensure that the tool would stand up against the forces introduced to the bender's structure during the bending operation. The analysis was performed using the ANSYS Workbench Finite Element Analysis software. The forces were determined by free body diagrams at the pin connections of the tool. The analysis results were successful in showing that most of the material was in a very low stress state and, therefore, was structurally unnecessary.

The model results were validated using experimental testing on the physical model using strain gauges. Equiangular rosette strain gauges were used and the experimental validation demonstrated excellent agreement.

A few different redesign options were entertained. Using thinner plate stock and adding flanges were two of the ideas that were not successfully implemented, the former because of the catastrophic stress state the plate was in and the latter because the cost of an additional process was not warranted given the gains in stress state. Instead, a less powerful and cheaper hydraulic ram was used. Additionally, quite a bit of material could be removed making the structure lighter. Even with the material removed, most of the plate was only at half of the material's yield strength, providing an overall safety factor of 2.

The project was ultimately successful in meeting its goal of reducing the weight and cost of the tube bender. Weight was reduced by 17% (20.9 lb) and cost was reduced by 11% ($55). It was also determined the "heavy-duty" model offered by the manufacturer was also grossly overdesigned given that the lighter duty version was the model being utilized in this project. The analysis and redesign were successful in meeting the desired objectives of this project.

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Introduction and Motivation

Tube benders are used in several industries in the fabrication of different tubular structures. A variety of such machines exist for both industrial and personal needs. These tools can be rather expensive with even the lighter duty personal models costing in the upwards of $500, excluding the cost of the similarly pricey die sets (ranging from about $200 to over $400, as based on size). These tools can also be quite heavy with the heavier duty versions obviously weighing significantly more. These properties make this tool rather unmarketable to the individual user and represent a rather significant investment on even adequately funded machine shops.

This analysis and redesign project was motivated by the purchase of a tube-bending machine by one of the researchers. Because of the significant cost of the machine, some insurance is desired that guarantees the tool will work properly without replacement for an extended lifetime. Also, the machine is quite difficult to move with a weight of about 120 lbs. The redesign of this machine is driven by the desire to remove weight, therefore making the tool more portable, as well as maintaining or reducing the present cost. The impact of this project theoretically affects any machine shop that would carry such a tool as well as individuals that would consider purchasing the tool for personal use.

Background Information

The model used in this project is a Pro3 Bender as can be purchased on the Probender website, www.probenders.com. It is meant to offer an affordable solution to private users and owners of small metal fabrications and machine shops. It is composed mostly of $^{3}/_{8}$” thick steel plates that are CNC laser-cut into the desired shape and configuration. An 8 ton hydraulic ram is also included with the tube bender that has both manual and compressed-air operating functionality. However, as aforementioned, because of its weight the unit lacks the desired portability that becomes a prominent issue in such small shops where space is an issue. Additionally, the manufacturer provides a “heavy-duty” model of tube bender that has the same functionality but utilizes ½” thick plate instead of the $^{3}/_{8}$” lighter duty version. This model will also be analyzed to determine the usefulness of the additional plate as opposed to its additional cost (approximately $200). Though there is not a particularly large market for this tool, the manufacturer estimates sales of about 30 units per day. At almost 7000 units per year, any savings on either the part of the consumer or the manufacturer can be significant.

Methods, Procedures, and Results

Computer Modeling

The analyses and redesign that comprise the bulk of this project first required an accurate and reliable computer model. In order to establish this model, numerous dimensions of the physical model were either measured or provided through the tool's manufacturer. These dimensions were then translated to a geometric model of each of the components using the SolidWorks computer aided design software. For matters of comparison, a photograph of the physical model and a screenshot of the computer-based model are offered in Figures 1, 2, and 3 below.

Figure 1: ProBender Tube Bender Physical Assembly

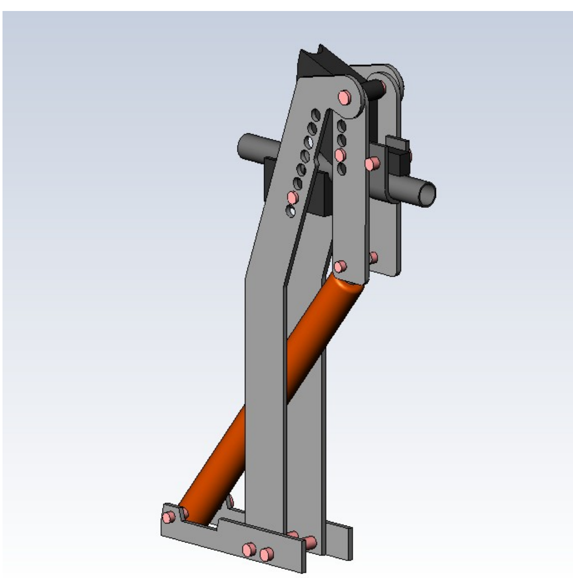

Figure 2: CAD Assembly Model, Loaded Position

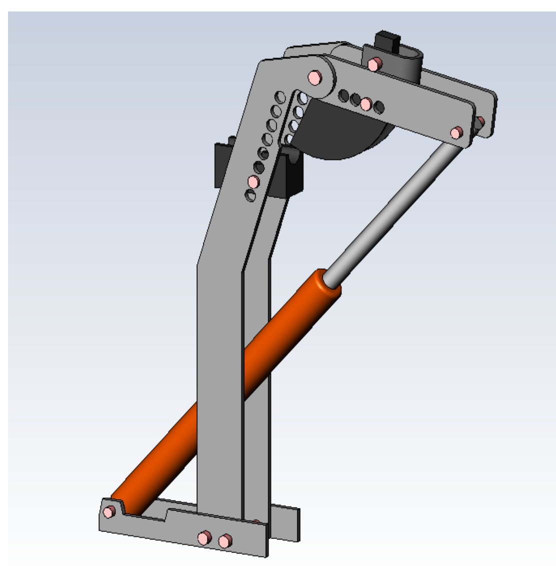

Figure 3: CAD Assembly Model, Extended Position

Finite Element Analysis

These models, after verifying the dimensions against the physical model, were then transferred into the ANSYS Workbench Finite Element Analysis (FEA) software environment. Though the entire assembly had been modeled, the only critical components that required FEA analysis were the main plates and the bending arms, shown below in Figures 4 and 5. Analysis was conducted at the beginning of the bending process where the effective lever arm is the shortest, producing the highest stresses in the bender.

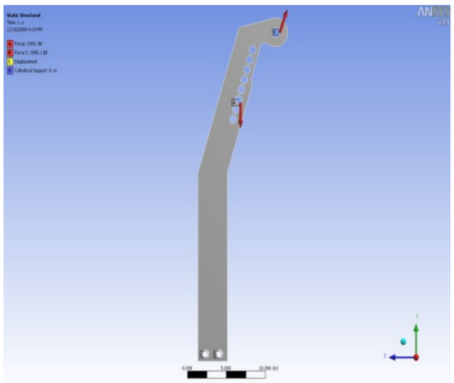

Figure 4: Main Plate Model

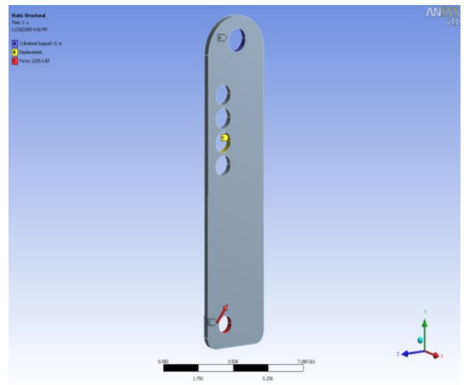

Figure 5: Bending Arm Model

In order to run the FEA on these components, some assumptions were required. First, in order to model the forces being realized on these components by the hydraulic ram, a calculation was performed using free body diagrams (seen in Figures 25 and 26 of Appendix A) to determine what magnitude of force would be required to yield the tubing which would be loaded into the tool. Though this assumption is not valid in the strictest sense, it does provide a fairly good approximation of the respective forces. The results of these calculations are recognized as the red arrows in Figures 4 and 5 above. The main plate is subjected to two forces, one with a magnitude 1943 lbs acting vertically downward and another with a magnitude of 3956 lbs acting diagonally upward, as seen in Figure 6 below. The bending arm only sees one applied force acting at a magnitude of 2266 lbs. The directions of the forces were determined through the geometry of the tool. Two kinematic constraints were applied to the main plate: a zero displacement constraint on one of the holes located at the bottom edge of the model (seen as a yellow band in Figure 7) and a cylindrical constraint on the other hole (seen as a blue band in Figure7) in order to simulate a hinge. Similar kinematic constraints were added to the bending arm model though at different locations that can be seen in Figure 5 above.

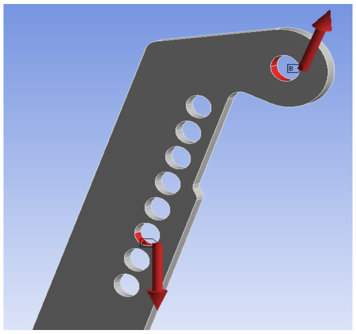

Figure 6: Forces acting on the Main Plate

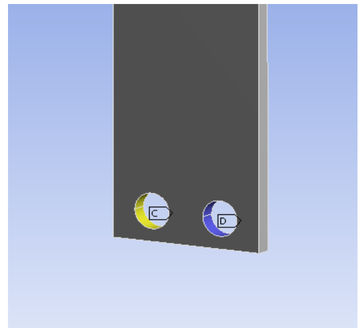

Figure 7: Kinematic Constraints on the Main Plate

Several iterations of the FEA were performed at increasingly refined mesh sizes. The stress levels converged appropriately in all areas of both plates with the exception of highly localized areas in corners. These were assumed to be artifactual results of the analysis because of the lack of convergence. Therefore, these highly localized areas, though considered during the redesign phase, were ignored for the purposes of the design testing. The results of the FEA can be seen in Figures 8 and 9. These results were attained at a mesh size of 1/8”. The highest stresses, excepting the localized regions of stress concentration, were approximately 25 ksi. This is about half of the yield strength of the material (assumed to be 1020 steel) which has a yield strength of approximately 50 ksi. However, as can be seen in the figures below there are large areas of very low stress, a clear indicator of component overdesign. These are appropriate areas to remove material in order to distribute the stress more evenly across the component.

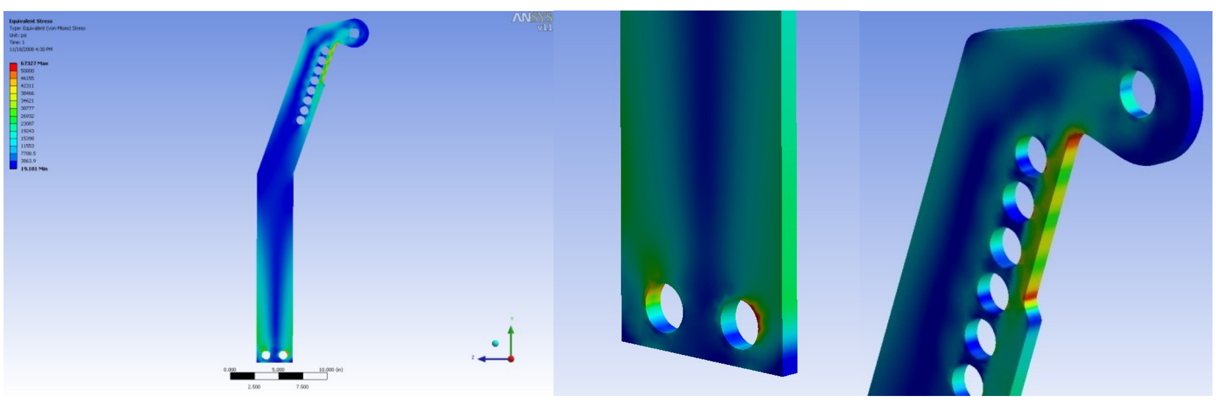

Figure 8: FEA results on the Main Plate

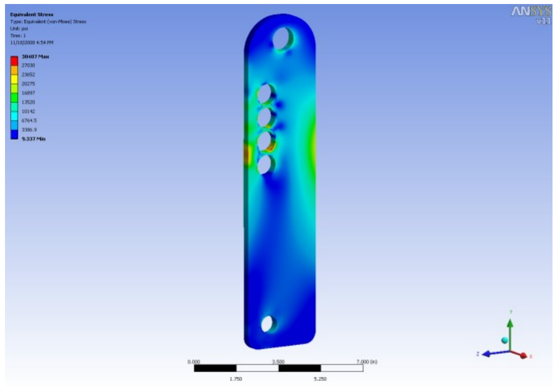

Figure 9: FEA results of the Bending Arm

Probabilistic Analysis

Based on the force analysis seen in Appendix B, the forces exerted by the hydraulic ram during the bending process are a function of tube dimensions. When bending a nominally sized tube of 1.75 inch outer diameter and 0.120 inch wall thickness, the ram is required to exert 4500 lbf to carry out the bending operation. However, tube dimensions will not always be exactly the same for a given tube size due to inherent small flaws in tube manufacturing. For the tube being bent in this specific application, manufacturers supply tolerances on two independent variables; the first being the outer diameter which has a tolerance of +/- 0.006 inches and the second being the wall thickness which can be +/- 10% (0.012 inches)

Tolerance limits are usually held to 3 sigma, meaning 99.73% of any randomly picked tubes will be sized within the tolerance limits. In this application, there are two independent tolerances, meaning that the probability of a tube being greater than both maximum tolerance limits is approximately 0.073%, or approximately 7 out of 10,000.

Applying both maximum tolerances to the force analysis yields a new exerted force by the ram of 5050 lbf. This force is later used to conduct FEA analysis on redesign options to ensure that new bender will safely handle 99.997% of bends that it will undergo.

Bending Analysis

In order to further understand the bending process, an analysis of the stresses in the tube undergoing bending was performed. Due to the complexity of the analysis, there is no simple mathematical formula to find the minimum thickness of a tube to bend without buckling. For this process a model of the die, the follow bar, and the tube were created in ANSYS Workbench. The bottom of the follow bar was set as a fixed support. The surface of the tube and the front face of the die were modeled to not have any separation. The top corner of the die was modeled as a hinge, with a moment applied around it. This moment varied with tube thickness, and was calculated using the analysis in Appendix B.

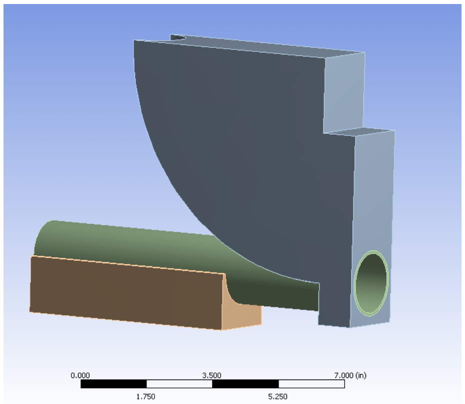

Figure 10: Model of tube, follow bar, and die

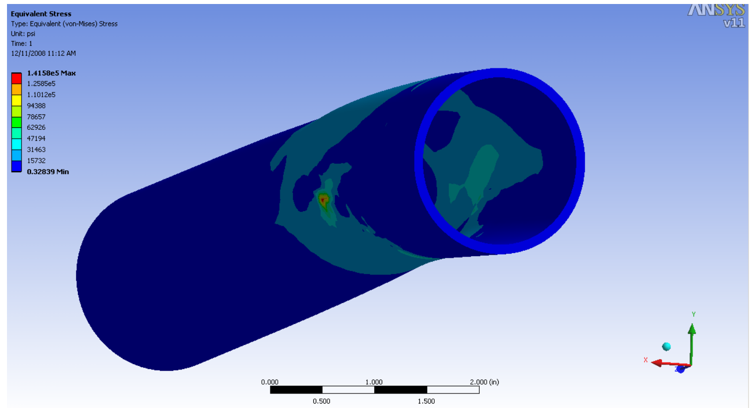

Figure 11: Stress results of the analysis using a .09" thick tube. Colors brighter than green represent a stress greater than the UTS

The original tube bender manufacture advises against bending 1.75" diameter tubes with a thickness of less than .09". The stresses in bending the thinnest tube were analyzed. Although the stresses on the tube are quite high, the ultimate tensile strength does not pass through the entire surface of the tube as seen in Figure 11. This shows that the stresses go beyond the yield strength as hoped for the moment calculated. If the stresses went beyond the ultimate tensile strength, then that would probably indicate that buckling or cracking in the tube would occur. In an analysis with a thinner tube (.075"), it can be seen in Figure 12 that a stress greater than the ultimate tensile strength is present through the thickness of the tube. An extreme deformation is also shown in Figure 13, to show what could happen to a tube that is less than the recommended thickness. This analysis suggests that the minimum thickness provided by the manufacture is a valid recommendation.

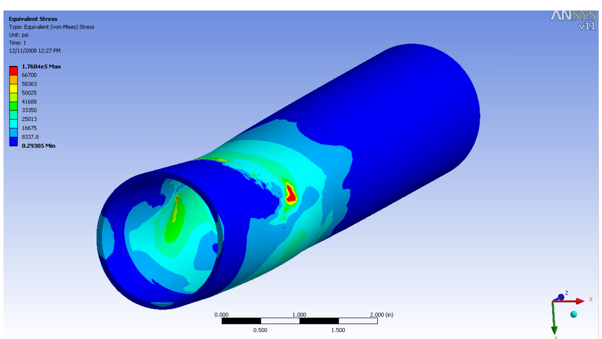

Figure 12: Stress results of the analysis using a .075" tube. The red in this picture represent a stress above the UTS

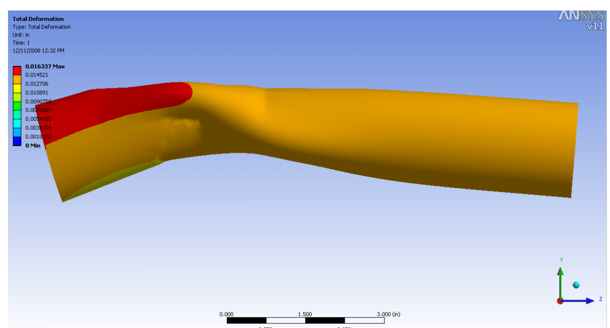

Figure 13: Estimated buckling behavior in a tube

Model Validation

In order to validate the computer model and analysis results, equiangular rosette strain gauges were used to measure the plane strain on the physical model during use. The strain gauges were placed in the areas of highest strain as seen by the computer model. In Figures 14 and 15 below, the highest strains were realized in vectors parallel to the edges of the components.

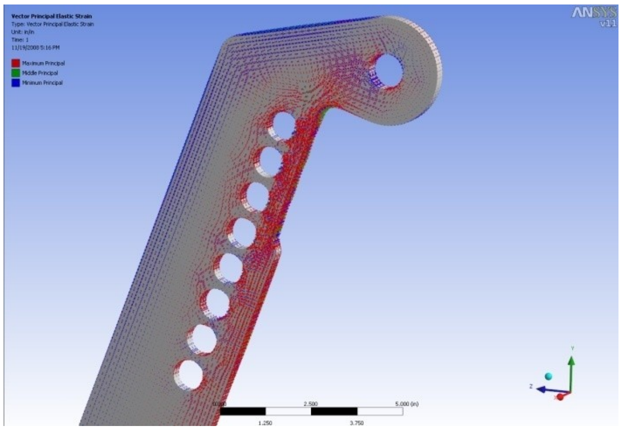

Figure 14: Principal Strain Vectors of the Main Plate

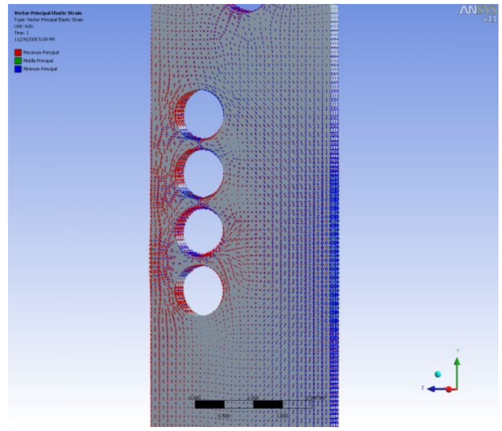

Figure 15: Principal Strain Vectors of the Bending Arm

The rosette stain gauges, seen in Figure 16, were carefully glued onto the model at the selected sites and soldered to electrical connectors as shown in Figure 17. Values of strain outputted by the gauges were recorded throughout a bending process which spanned the stroke of the ram, bending the tube to 90 degrees.

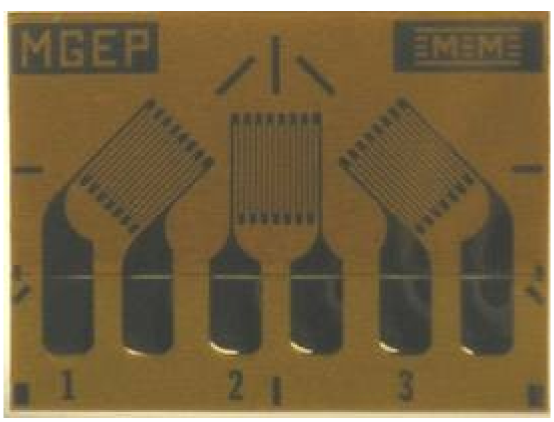

Figure 16: Equiangular Strain Gauge

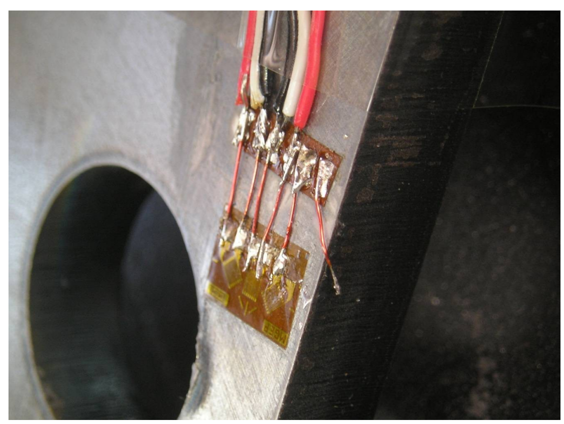

Figure 17: Strain Gauge glued on Main Plate

The results of the strain gauge analysis can be seen in Table 1 below. It should be noted that the third strain gauge for the main plate trial broke off during soldering, resulting in no data for this strain gauge. Therefore, agreement between the computer model and physical model should be most accurately measured by the agreement between the vector strain parallel to the edge of the plate and the strain gauge parallel to the plate (Strain Gauge 2 in Table 1). As is readily noticed, the strain analysis showed excellent agreement with the computer generated results.

Table 1: Strain Gauge Analysis Results

	Main Plate	**Bending Arm**
Strain Gauge 1	0.000430	-0.000192
Strain Gauge 2	0.001296	-0.000631
Strain Gauge 3	N/A	-0.000341
FEA Strain in direction of Strain Gauge 2	0.001113	-0.000624

Redesign Options and Results

Several redesign ideas were entertained in order to make the tube bender both more marketable and more efficient for the cost. It should be noted that due to configuration constraints, the availability of redesign options was limited. The prescribed holes built into the components as they are seen above allow the tube bender to accommodate a variety of die sets, allowing for an assortment of tube sizes, and several different bending radii. Therefore, the only redesign options considered were those that could be implemented without decreasing the functionality of the tool as it stands. Each of these redesign options are considered at length below. All FEA tests were conducted using the upper force limits found from the probabilistic analysis.

Using Thinner Plates

The first design option considered was to use thinner plate from which to cut the components. The plate on the current physical model was 3/8" thick. Additionally, the manufacturer also provides a "heavy-duty" version of the same bender using ½" thick plate for an additional $200 in cost. The initial analyses seem to indicate that the heavy-duty model is even more overdesigned than the current 3/8" model. The analysis was iterated on the components with a thickness of 1/4", the next standard plate size down from the current model. However, this redesign choice was deemed unacceptable because of the high stresses across the entire component as seen in Figure 18. It should be noted that though the bending arm *could* have been cut from thinner plate, this would have added cost to the consumer because the manufacturer would then have to cut from two separate plate stocks. Therefore using thinner plate was dismissed as a redesign option.

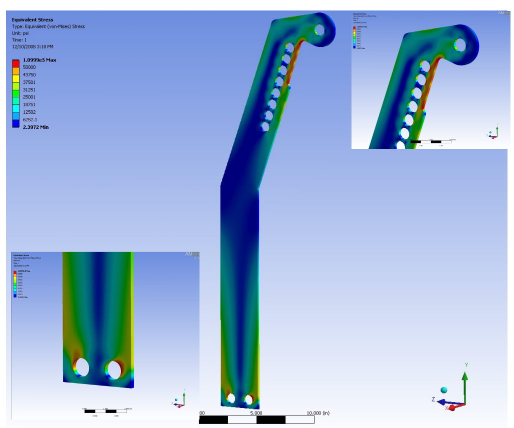

Figure 18: Thin Plate Analysis Results

Adding a Flange

Another redesign option that was entertained was adding a flange to the outer edge in of the main plate order to help absorb some of the stress of the thin plate. Though the flange did help distribute the stress in this component, lowering the stress concentrations, it did not perform this function sufficiently enough to warrant the cost of the additional fabrication process of bending. Furthermore, this design option could not be implemented on the bending arm because of the interference it would have caused with moving parts. Additionally, this option would add weight which would oppose one of the project's goals. Therefore, this redesign option was also dismissed.

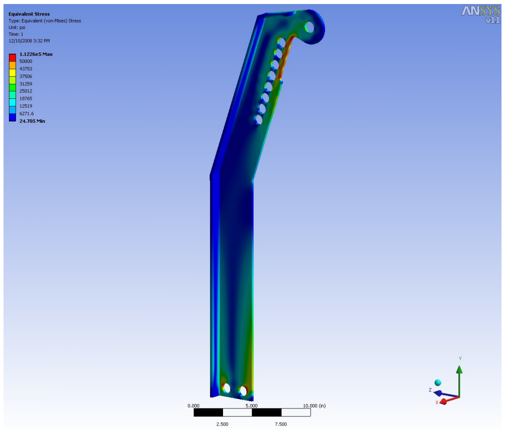

Figure 19: Flange Option Analysis Results

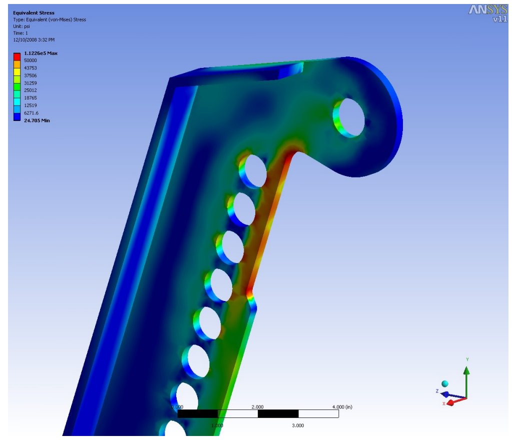

Figure 20: Critical Area of Flange Option Analysis Results

Removing Material

The third redesign option that was considered was removing material from the low stress areas in each of the components. Since the bending arm and main plate are both CNC laser-cut from plate-metal stock, removing additional material would not add significant cost to the manufacturing process. Also, removing material has the added benefit of reducing the weight, a favorable addition given the goals of the project. Manual iteration was used to determine the optimal width and length of the ovals of removed material in each of the components. Additionally, further manual iteration was used to determine the maximal width that could be taken off the spines of each component, making the components slimmer. Finally, some material was added at the high stress notch in the upper portion of the main plate. This helped diffuse the highly localized stresses in this area. With all the material removed, the highest stresses realized across most of the surface were only 25 ksi, providing a general safety factor of 2 for the component. The finalized components with the material removed can be seen in Figures 21 and 22 below.

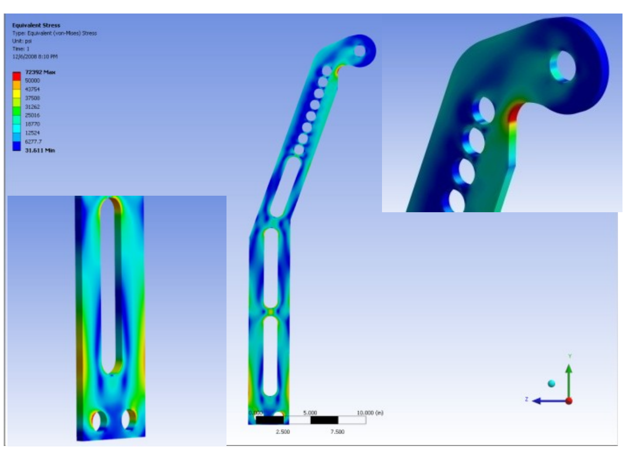

Figure 21: Final Main Plate Redesign

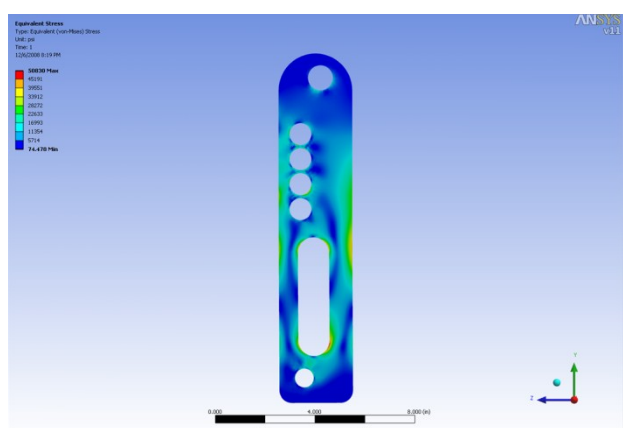

Figure 22: Final Bending Arm Redesign

Size of the Hydraulic Ram

The probabilistic analysis discussed earlier demonstrated that an overwhelming percent of the bends (99.992%) performed would only require, at most, 5000 lbf. The hydraulic ram that is included as a standard for this particular tube bender is capable of 8 tons (16000 lbf). A very quick comparison yields that the ram is overdesigned for this particular application. Therefore, a less powerful, but comparably sized, ram was found that could be used in this tube bender. The current 8 ton ram has a stroke of 17 in., can be run either manually or by air compressor, costs approximately $100, and weighs about 37 lbs. The new ram has a stroke of 20 in., costs about $45, and weighs about 27.5 lbs. As it is currently sold, the new ram can only be operated manually but should the sales of such a unit increase, a compressed-air operation can be added for little cost.

Discussion

For ease of comparison, the original and redesigned assemblies are demonstrated in Figures 23 and 24 below.

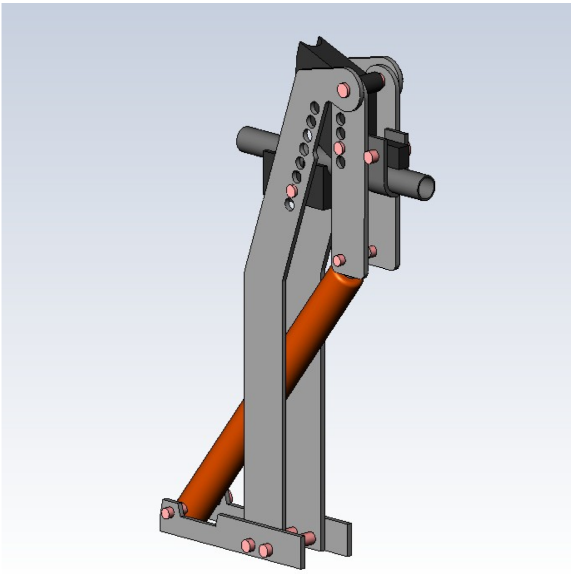

Figure 23: Original Tube Bender Assembly

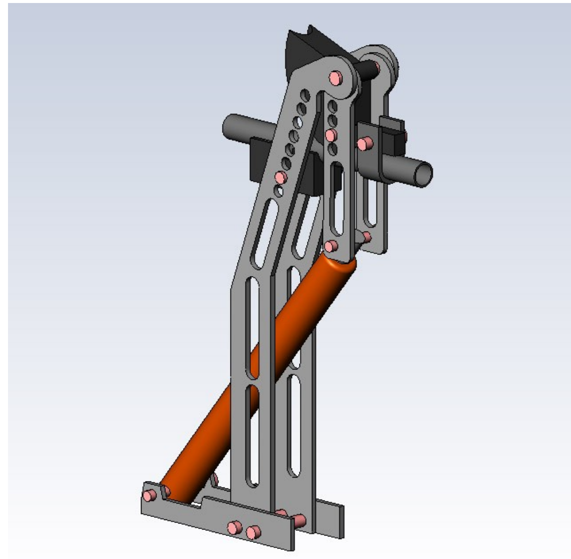

Figure 24: Redesigned Tube Bender Assembly

The redesigned tube bender weights about 20.9 lbs (17%) lighter than the original design. The redesign is also about $55 (11%) cheaper. Therefore, the results of this project meet its initial goals. The redesigned tube bender maintains all of the functionality of the original, including the sizes of tube which it can accommodate. This makes the tool quite a bit more marketable to personal users or small machine shops that would be interested in purchasing it.

Additionally, the material being removed adds little work to the manufacturer (because it is CNC laser-cut) and no significant cost. Furthermore, the removed material may help the manufacturer save money in scrap steel costs. Based on the weight removed, it was estimated that the additional material removed will result in an additional scrap steel savings of about $0.75 per unit (based on scrap steel prices of approximately 150 $/ton). From discussions with the manufacturer, it is estimated that about 30 units/day (7000 units/year) are sold for a total annual savings of about $5400 from the additional scrap steel costs alone.

The analyses also demonstrate that the ½" thick "heavy-duty" model currently sold by the manufacturer is an entirely unnecessary investment. Priced at $200 more, the additional plate provides neither added functionality nor necessary protection against failure. This model is, in all aspects, thoroughly overdesigned.

Table 2 summarizes the comparable differences between the original and redesigned models.

Table 2: Summary and Comparison of Designs

	Original Design		**Redesign**	
Component	**Weight**	**Cost**	**Weight**	**Cost**
Ram	37 lb	$100.00	27.5 lb	$45.00
Main Plate	13.8 lb	N/A	10.1 lb	$0.75 saved per unit in scrap metal costs
Bending Arm	4.3 lb	N/A	3.3 lb	
Total for assembly	117.2 lb	$500.00	98.3 lb	$445.00

Conclusion

The purpose of this project was to analyze and redesign the structure of a currently modeled tube bender with the ultimate goal of offering redesign suggestions that would make the tool more efficient and marketable by making it lighter and less costly. Given the objective of this project, the goals were satisfied successfully. As outlined in the discussion above, the tube bender redesign is both lighter and cheaper. Furthermore, this was able to be accomplished at no loss of functionality. This demonstrates the importance of proper engineering technique and analysis when designing a costly and, as seen here, potentially overdesigned product.

Appendix A: Free Body Diagrams

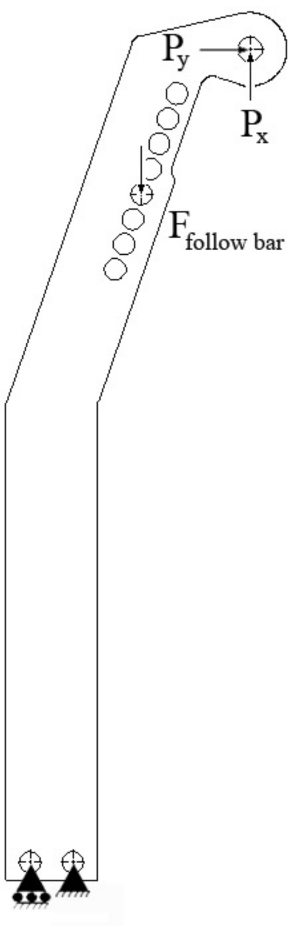

Figure 25: Free Body Diagram of the Main Plate

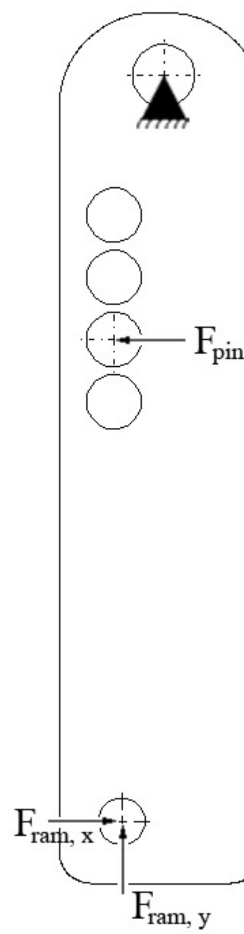

Figure 26: Free Body Diagram of the Bending Arm

Appendix B: EES Code for Force Determination

Tube Specs

$$\sigma_y = 50990 \quad \text{[psi]}$$

$$D_{out} = 1.75 \quad \text{[in]}$$

$$D_{in} = D_{out} \; y \; 2 \cdot 0.12 \cdot 1 \quad \text{[in]}$$

Geometry

$$a = 3 \quad \text{[in]}$$

$$b = 1 \quad \text{[in]}$$

$$L = 4 \quad \text{[in]}$$

Tube Eq's

$$I = \frac{\pi}{64} \cdot \left(D_{out}^{4} \; y \; D_{in}^{4} \right)$$

$$c = \frac{D_{out}}{2}$$

$$M_{bend} = \frac{\sigma_y \cdot I}{c}$$

$$F_{bend} = \frac{M_{bend} \cdot L}{a \cdot b}$$

$$F_{strap} = \frac{F_{bend} \cdot a}{L}$$

$$F_{follow} = \frac{F_{bend} \cdot b}{L}$$

Die Analysis

$$d = 3 \quad \text{[in]}$$

$$e = 4 \quad [in]$$

$$F_{pin} = F_{strap} \cdot \frac{d}{e}$$

$$P_{die,x} = y\ F_{pin}$$

$$P_{die,y} = y\ F_{bend} + F_{strap}$$

Bending Arm Analysis

$$f = 4 \quad [in]$$

$$g = 8 \quad [in]$$

$$F_{ram,x} = \frac{F_{pin} \cdot f}{f + g}$$

$$F_{ram} = \frac{F_{ram,x}}{\mathbf{sin}\ (40)}$$

$$F_{ram,y} = F_{ram} \cdot \mathbf{cos}\ (40)$$

$$P_{bending,x} = F_{pin}\ y\ F_{ram,x}$$

$$P_{bending,y} = y\ F_{ram,y}$$

Main Plate Analysis

$$P_{main,x} = y\ P_{die,x}\ y\ P_{bending,x}$$

$$P_{main,y} = y\ P_{die,y}\ y\ P_{bending,y}$$

$$P_{main,follow} = F_{follow}$$