

Wood Cutter Design

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Objective

The goal of this project is to assess the design of a woodcutter through the creation of free body diagrams for each component and evaluating the effects that applied forces will have on the device. By computing the stress and factor of safety of each element, we will identify the most vulnerable point of the device and utilize this information to enhance the design of the woodcutter. This project holds great importance for our design profession. The ability to determine how a device will react to stress during use and pinpoint the weak areas of the device allows for optimization of the created device and ensures it will not fail during operation. The woodcutter utilizes a lever arm with a sharp edge to apply force to a piece of wood and split it. This device enables easy woodcutting and has potential applications for other materials, by increasing the force applied on the material using the lever arm. However, due to the small size of the cutter, it is limited to smaller applications.

Selection of Design Load and Justification

In selecting the loads for our proposed design, the team designed the woodcutter to be able to withstand the maximum push-force that a human could output. While we realize that the average user will not be applying their maximum effort to the woodcutter, we decided best practice is to design our woodcutter to with maximum possible forces and stresses in mind. In addition, to account for the maximum bending stress, we decided to place our load at the furthest point along the cutter. This would provide us with an accurate representation of the maximum possible moment that could be applied to this member of our structure. The team identified the maximum push force for a human in the standing position to be 107 lb. [2] and subsequently, the woodcutter is designed for a maximum applied load of 100 lb. The team then quantified the average user applied forces to be 50 lb. and 75 lb. and has supplied our findings for all three tested load values below in this report.

Free Body Diagrams

The wood cutter was measured using dial calipers and measuring tape, all the measurements taken were in inches. Applied forces were drawn on the free body diagram of each member of the wood cutter to be analyzed for improvements using the assumptions that were made. The free body diagram of each member helped us visualize the effects caused by the force applied on the wood cutter. After determining the effects for applied force, laws of static equilibrium were used to quantify the maximum stress and strain on each of the four members. It should be noted that the forces were counted as applied at the center line of each body. The depicted forces are drawn at the actual points on the body at which they are applied. This assumption is noted later, as it affects the calculations more than the depictions. Below is depicted the overall free body diagram of the entire subject, with the applied force and the two reactionary ground forces.

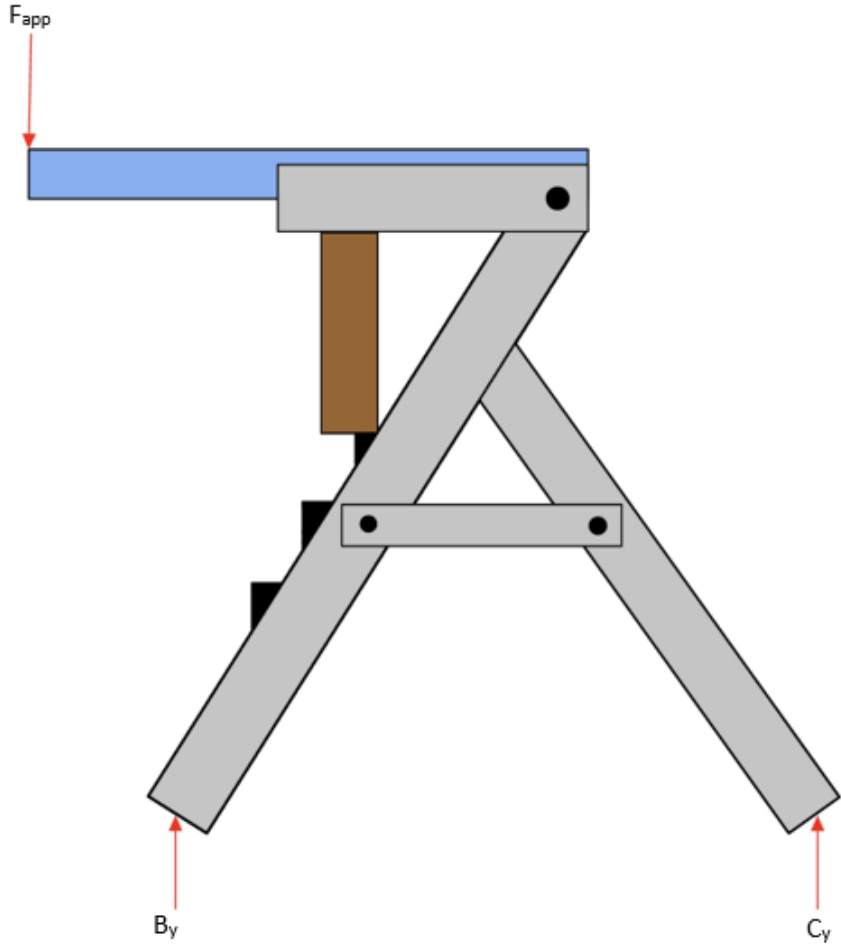


Figure 1: Overall Free Body Diagram

The first part to analyze is The Cutter. This is the part of the tool that has the blade and performs the actual cutting. Below is the free body diagram including the forces and measurements. The axis creates the base that all measurements are taken from, the left most size of the part. The forces that are present in the cutter are force applied by the user, reaction force by the wood that is vertically positioned, reaction force A_y by the pin connection.

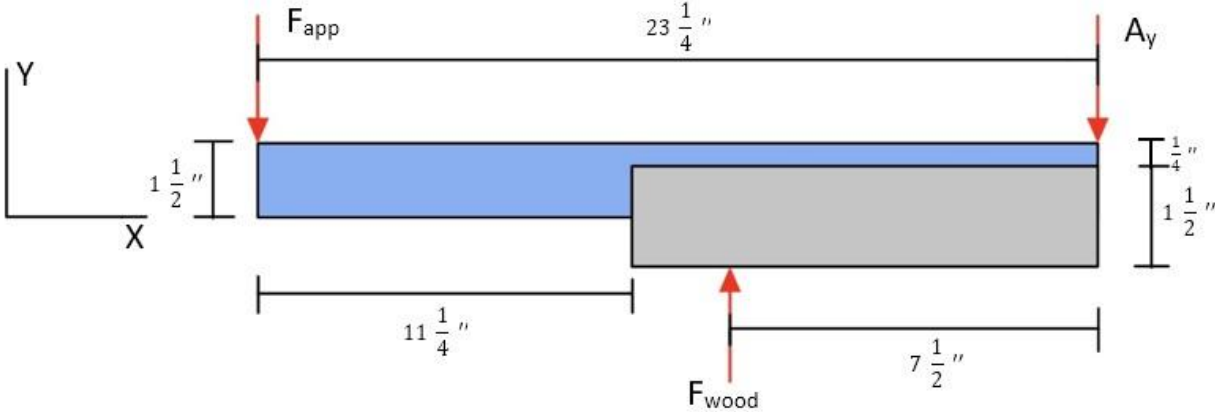


Figure 2: The Cutter Free Body Diagram

The next part to analyze is the Left Leg. This part of the tool is where the wood is held against while it is being chopped with the cutter part. Below is the free body diagram of the forces and the measurements for the part. The axis is oriented below, beginning at the left most part of the left leg. The forces present in the left leg cause by the wood, reaction forces in x and y from ground, moment by the force applied by the user, force acting inwards by the 2-point connecting member.

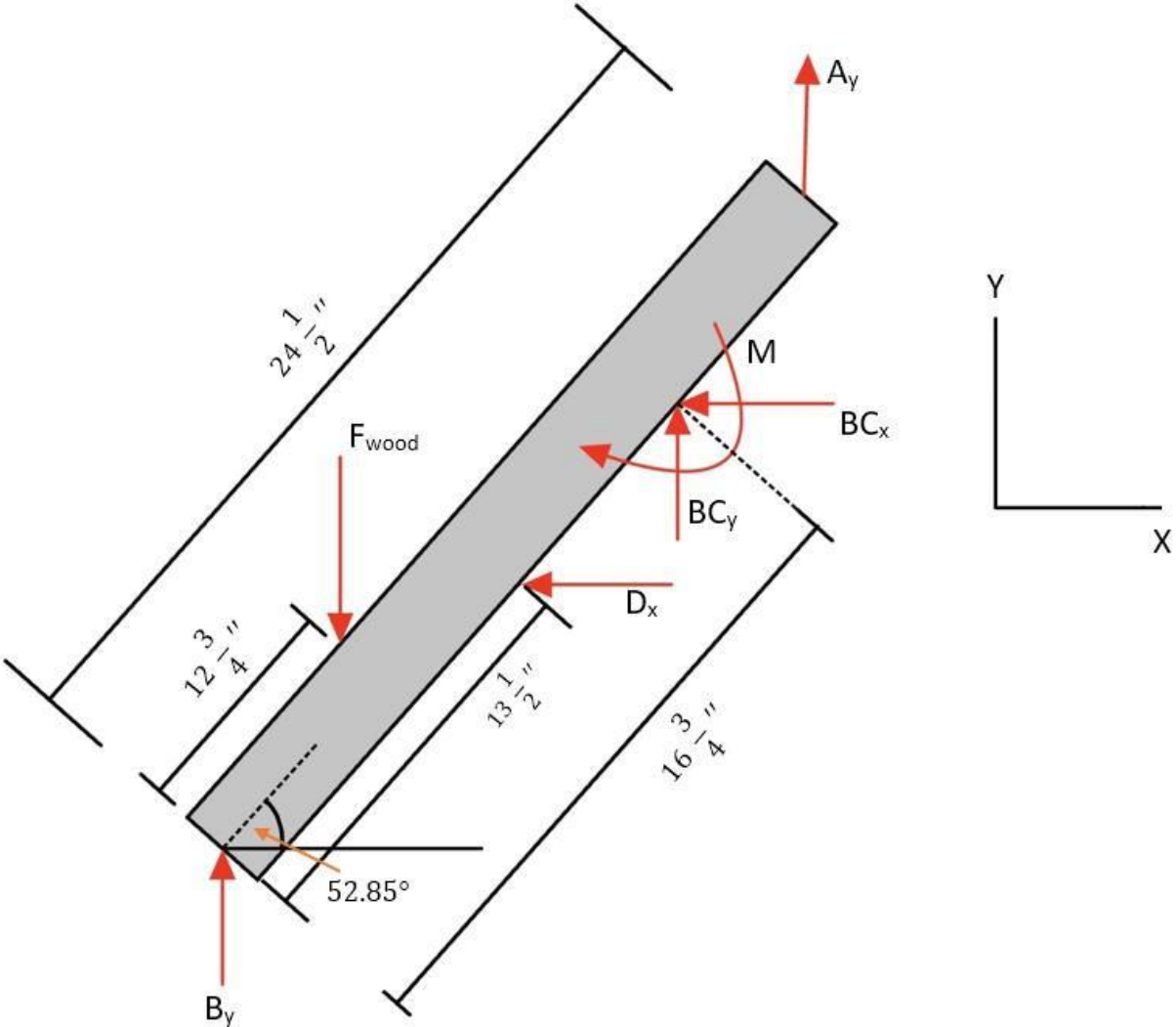


Figure 3: Left Leg Free Body Diagram

Now below is the free body diagram of the right leg of the assembly. This leg is opposite the main left leg and supports the left leg. The axis is oriented as shown, and also begins at the left most part of the leg, the top corner. This is important to know when looking at the shear and bending diagrams. The free body diagram for the right leg includes that reaction force in x and y direction by the ground and reaction force by point B, moment caused by the user force applied.

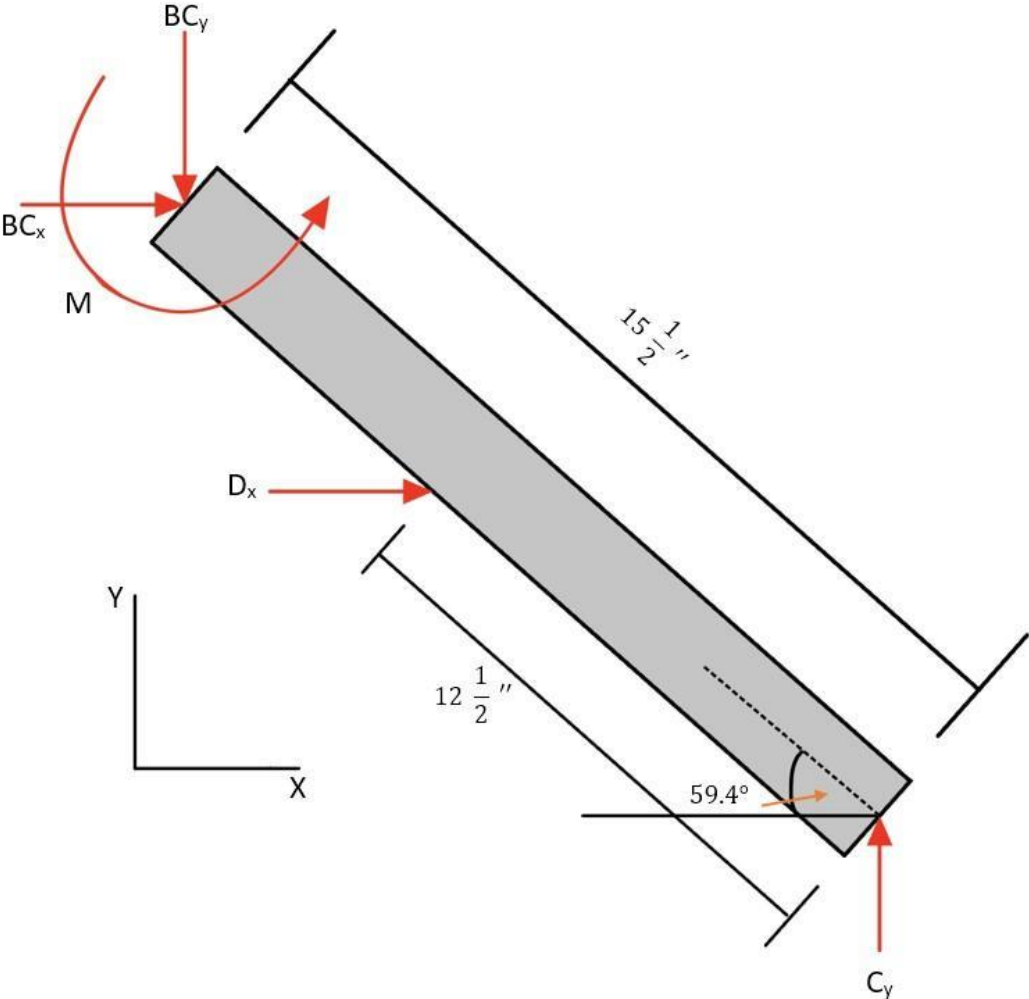


Figure 4: Right Leg Free Body Diagram

Now below is the free body diagram of the 2-point connecting member. This member has equal forces acting in compression.

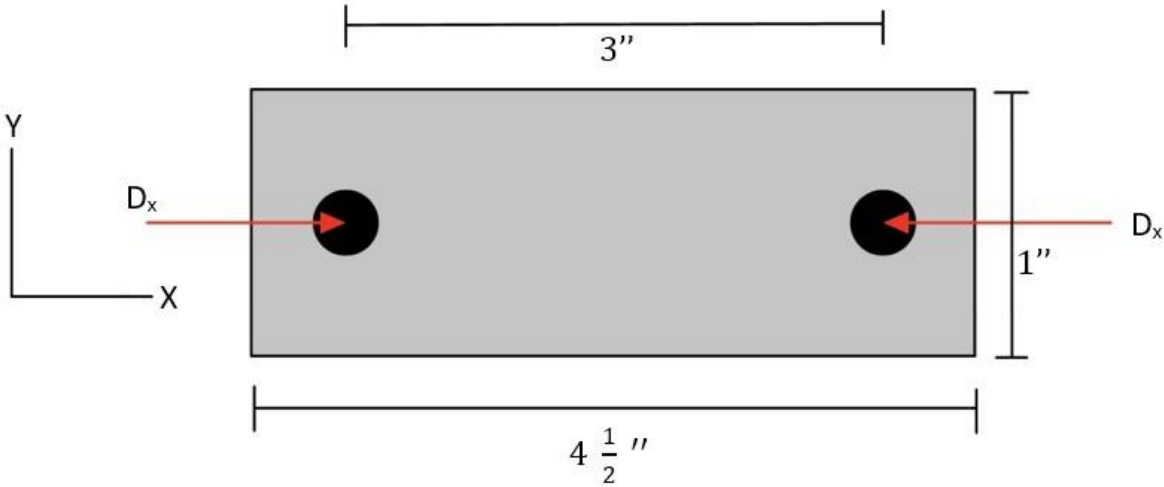


Figure 5: Center Connector Free Body Diagram

Calculation of Stress, Strain, and Factor of Safety

Simplification with Reasons

In order to perform our calculations, there are few assumptions that were considered. The wood piece was positioned perfectly vertical which made our top member (cutter arm) horizontal to the ground. The connection between the left leg and the right leg will be modeled using a moment about the pin connections from the middle connecting member. Our model is assumed to be stationary for the calculations. The weight of the wood cutter is considered to be negligible. All forces were applied to the center line of each beam. Our initial design assumed that there is a fixed connection between the left and right leg. However, after creating our equilibrium equations and the computational model, we noticed that the connection could be more accurately modeled as a pin connector. We adjusted our model of the connection to contain only forces, no reactionary moment.

Material Selection

We decided to use AISI 1018 CD Steel for our project. The modulus of elasticity is 29 [Msi] and the yield strength is 53.7 [ksi]. [1]

Sample Calculation

Now there will be a sample calculation for the right leg, as well as the equilibrium equations used to calculate the forces on each part. Since our selected material is ductile, we will be using Maximum Shear Stress Theory (MSST) to analyze the principal stresses and factors of safety.

Calculating the Equilibrium Equations:

Cutter Equilibrium Equations:

$$\sum F_x = 0 \quad [1]$$

$$\sum F_y = -F_{app} + F_{wood} - A_y = 0 \quad [2]$$

$$\sum M_A = -(F_{wood} \times 7.5) + (F_{app} \times 23.25) \quad [3]$$

Left Leg Equilibrium Equations:

$$\sum F_x = -D_x - BC_x \quad [4]$$

$$\sum F_y = B_y - F_{wood} + BC_y + A_y = 0 \quad [5]$$

$$\begin{aligned} \sum M_B = & -(F_{wood} \times 12.75 \cos(52.85)) + (D_x \times 13.5 \sin(52.85)) + \\ & (BC_y \times 16.75 \cos(52.85)) + (BC_x \times 16.75 \sin(52.85)) \\ & + (A_y \times 24.5 \cos(52.85)) - M = 0 \end{aligned} \quad [6]$$

Right Leg Equilibrium Equations:

$$\sum F_x = BC_x + D_x = 0 \quad [7]$$

$$\sum F_y = -BC_y + C_y = 0 \quad [8]$$

$$\begin{aligned} \sum M_c = & -(D_x \times 12.5 \sin(59.4)) - (BC_x \times 15.5 \sin(59.4)) \\ & + (BC_y \times 15.5 \cos(59.4)) + M = 0 \end{aligned} \quad [9]$$

Center Connector:

$$\sum F_x = D_x - D_x = 0 \quad [10]$$

$$\sum F_y = 0 \quad [11]$$

$$\sum M = 0 \quad [12]$$

Overall System:

$$\sum F_y = -F_{app} + B_y + C_y = 0 \quad [13]$$

Calculation of resultant forces

After drawing out the free body diagrams, an 8x9 matrix and a 1x9 matrix was created for the forces and $F_{applied}$ respectively using the above-mentioned equilibrium equations. The matrix left division operator was used to determine the resultant forces for each part. This can be seen in the attached appendix A.

Calculation of Izz

The right leg is a hollow rectangular prism and the moment of inertia must be calculated for further calculations. The moment was determined to be

$$I_{zz} = \frac{BH^3}{12} - \frac{bh^3}{12} \quad [14]$$

Where B and H are external measurements and b and h are internal measurements, and Izz is the moment of inertia. It can be noted that a separate Izz was calculated for the cutter as the cross section is not a hollow rectangular bar. It is a combination of a solid rectangle and a solid triangle. This can be seen in the MATLAB attached in appendix A.

Calculation of bending moment

The bending moment was calculated using the `cumtrapz(X, Y)` function in Matlab. X is a vector of coordinates of the lengths of the parts while Y is a vector of the forces applied to the part. The `cumtrapz(X, Y)` function computes the approximate cumulative integral of Y via the trapezoidal method with unit spacing. This effectively takes the area under the curve of the shear force diagram. This can be done by hand however this is more effective considering MATLAB was used for all calculations.

Calculation of bending stress

Now the bending stress needs to be calculated. This is simplified through MATLAB as well. The data points for the moment were created in the last section, and the following equation can be applied to this vector of moment values to create a vector of bending stress values. In order to calculate the bending stress, the following equation was used with the previously calculated I_{zz} values.

$$\sigma_{bend} = \frac{M(\frac{h}{2})}{I_{zz}} \quad [15]$$

Calculation of axial force

The axial force was calculated from the summation of forces along the member. The forces used are BC_x , BC_y , D_x , C_y . These forces do not line up perfectly axially with each part, therefore the applied forces need to be translated into partial forces that are in the axial direction. These are summed up taking into account their direction and sign. This was done in MATLAB, an example of this is the right legs ground force being converted to the axial direction in the below equation.

$$F_{C_y, Axial} = C_y \times \cos(30.6) \quad [16]$$

Calculation of axial stress

For the axial force, the sum of forces and the cross-sectional area were used. The area was calculated by subtracting the internal area from the external area. Using the axial force and the area, the stress can be calculated by the equation.

$$\sigma_{axial} = \frac{F_{axial}}{A} \quad [17]$$

This equation requires the cross-sectional area of each part being analyzed. These can be seen below, with the cross-sectional area for a hollow rectangular beam followed by the cross-

sectional area for the connector. The cutter has no axial forces so there is no stress in the axial direction for this part.

$$A_x = (B_{out} \times H_{out}) - (B_{in} \times H_{in}) \quad [18]$$

This equation represents the cross-sectional area A_x . Internal dimensions of the cross-section are represented by B_{in} and H_{in} . External dimensions for cross-sectional area are B_{out} and H_{out} .

Calculation of principal stresses

The principal stresses were calculated using the following equations which are used in the MSST factor of safety calculation.

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\frac{\sigma_x^2}{2} + \tau_{xy}^2} \quad [19]$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\frac{\sigma_x^2}{2} + \tau_{xy}^2} \quad [20]$$

σ_1 and σ_2 are the principal stresses, σ_x are the axial stresses, and τ_{xy} are the maximum bending stresses. This is needed because the legs both experience axial and bending stress.

Calculation of factor of safety

The factor of safety was calculated using the MSST failure theory. This is because the material that was selected is a ductile material and has a yield strength significantly lower than the ultimate strength. The factor of safety was calculated using S_y and the maximum shear stress as calculated above. The equation used is the following:

$$n = \frac{S_y}{\sigma_1 - \sigma_3} \quad [21]$$

The legs both require this equation, and for the connector which only has axial stress, the second stress value become zero and the equations becomes the below equation. This also applies to the cutter, which only has bending stress.

$$n = \frac{S_y}{\sigma_1} \quad [22]$$

Results

In this section, the resultant shear force, bending moment, and bending stress diagrams are presented. These were created in MATLAB; the code is included in the appendix [Appendix A]. These diagrams are all created using an applied force of 100 [lbf]. Below is the diagram for the cutter, or chopper.

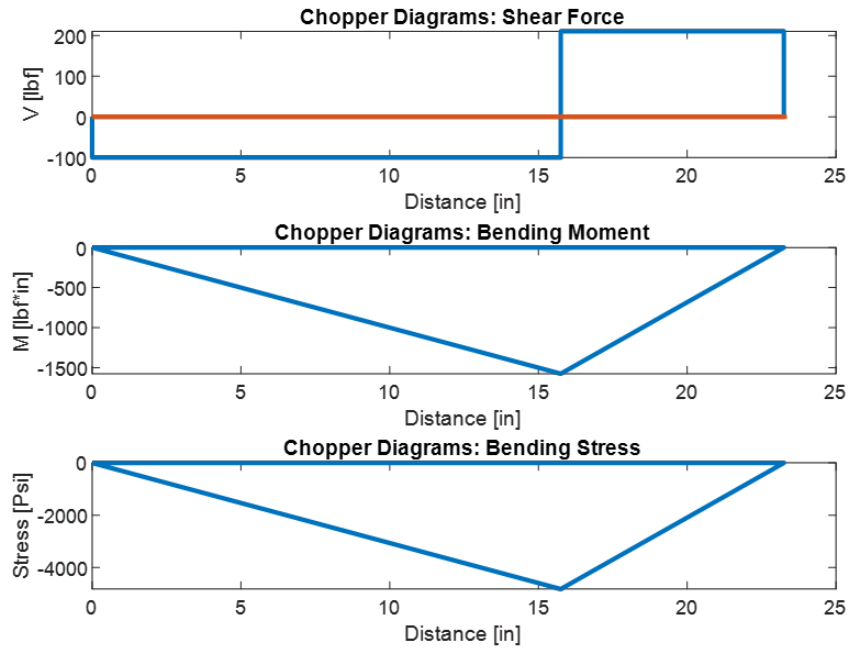


Figure 6: Cutter Shear, Moment, and Bending stress Diagrams

Below is the diagram for the left leg, the axis beginning at the leftmost point of the leg in the same orientation as the free body diagram. The forces are not perpendicular to the part, and therefore the relevant force parts were calculated using the angle of the leg, to find the partials of each force acting in the perpendicular direction.

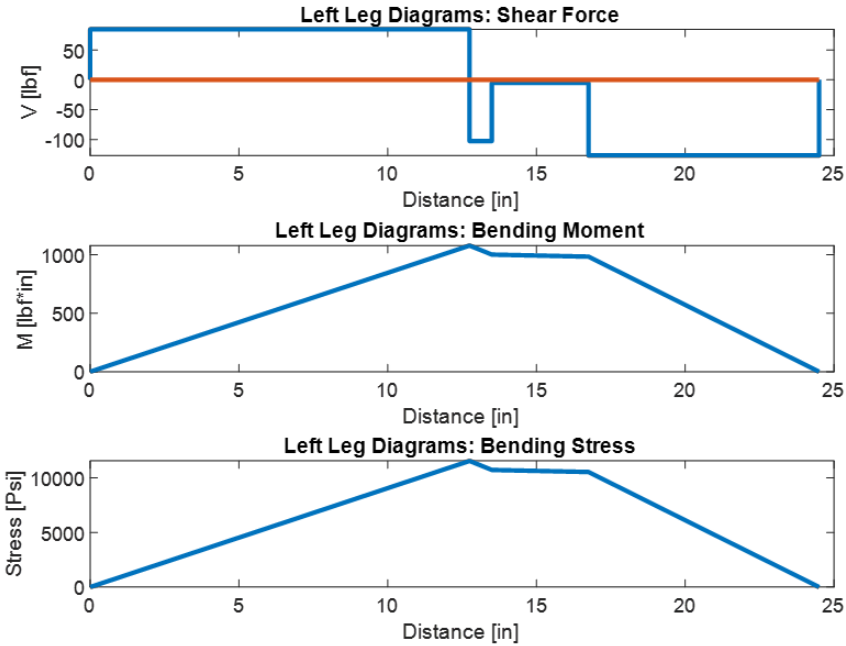


Figure 7: Left Leg Shear, Moment, and Bending stress Diagrams

Below is the diagram for the right leg, the axis beginning at the leftmost point of the leg in the same orientation as the free body diagram. The forces are not perpendicular to the part, and were treated in the same manner as the left leg.

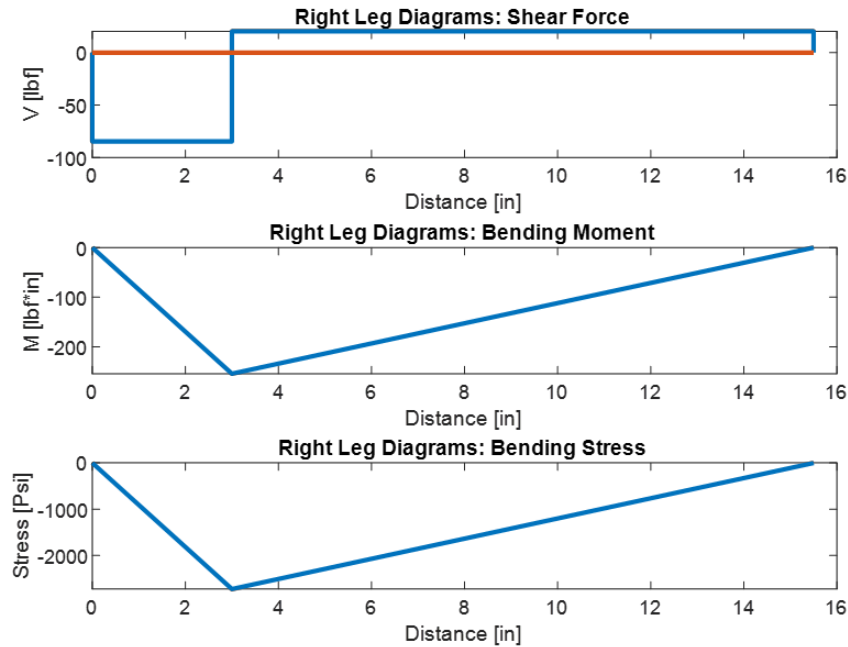


Figure 8: Right Leg Shear, Moment, and Bending stress Diagrams

The connector is a two-force member and does not have a shear force bending moment or bending stress diagram. The part only faces forces in the axial direction. This part's axial stress was calculated using the tensile force and the area of the member.

Below are the tabulated results of the resultant forces for each applied loading. These forces correlate to the free body diagrams in the above sections, and were used in the creation of the above shear, bending moment, and bending stress diagrams.

Table 1: Max Forces for each part and Where they Occur

Resultant Forces: All units in [lbf]								
Applied Force	F_{wood}	A_y	B_y	BC_y	D_*	BC_x	C_y	M
50 [lbf]	155	105	69.9	-19.9	61.0	-61.0	-19.97	0.00
75 [lbf]	232	157	104	-29.9	91.5	-91.5	-29.95	0.00
100 [lbf]	310	210	139	-39.9	122	-122	-39.94	0.00

Below are the tabulated results for the minimum factors of safety for each part and where on the part they occur. The location is given in distance correlating to the bending stress diagrams for ease of comparison. The visual location of each point can be gathered from the free body diagrams, and is covered in the discussion portion of the paper.

Table 2: Resulting Factors of Safety

Applied Force	Cutter		Left Leg		Right Leg		Connector	
	n	Location	n	Location	n	Location	N	Location
50 [lbf]	6.3	15.75 [in]	8.83	13.5 [in]	39.4	3 [in]	220	N/A
75 [lbf]	4.2	15.75 [in]	5.89	13.5 [in]	26.2	3 [in]	146	N/A
100 [lbf]	3.1	15.75 [in]	4.4	13.5 [in]	19.7	3 [in]	110	N/A

Discussions

Failure Theory & Critical Points

After drawing the free body diagrams of the structure. We were able to determine the effects of the user applied force on each of the members. We were then able to use static equilibrium laws to identify the forces, stresses and strains associated with our model and used the principal stresses to identify the critical points based on MSST failure theory.

All of the critical points in our design were determined based on the location where the bending stress is the highest and inherently where the factor of safety would be the least. For the cutter, the critical point was located 15.75 inches away from the applied load (to the right). At this location where the wood meets the cutter, our minimum factor of safety was 3.1, and the stress was 4.3 kpsi. For the left leg of our design, the critical point was found to be 13.5 inches from the base of the leg, which is where the wood rests as it is chopped. With our maximum loading, the minimum factor of safety at this point was found to be 4.4 and the associated stress was 12.3 kpsi. For the right leg, the critical point was found to be located 3 inches from its base. With our maximum loading applied, the minimum factor of safety at this location was found to be 19.7, and the associated stress was 2.8 kpsi. Lastly, the connector piece was taken to be a two-force member, with only two equal and opposite forces being applied along the same axis in the member. Due to this, there is no axial or bending stress felt by the member, and therefore thus no critical points along the connector. With our maximum load applied, the minimum factor of safety for the connector was calculated to be 110.

Design Improvements

After analyzing the critical points and determining the factor of safety of each member, it was clear that the 2-point connection could be made thinner to save material cost and machine weight without considerably lowering the factor of safety from its current value of 110. A second design improvement which we can manipulate after analyzing our data is the factor of safety for our right and left leg. This can be accomplished by adding 7 in to cutter arm which brings the factor of safety of the left leg down to 3.3 instead of 4.4 and the right leg down to 10.2 from 19.7. Increasing the horizontal arm length will allow better leverage for the user to utilize when cutting the wood. This design change is based in the fact that an increased moment arm radius will allow the user to cut the wood using less force. The factor of safety is not majorly affected on the cutting arm as a result of this change, which makes this design change reasonable.

Conclusions

The most valuable lesson the team has learned is how to apply engineering models to design an item with real world applications. We were able to use static equilibrium equations to obtain a complete computational model of our design. Our computational model includes calculations for axial stress, bending stress, factor of safety, and much more. We considered multiple design criteria such as material, applied load, and different brittle failure theories, and were able to view in real time how proposed design changes impacted our structure. This implementation of iterative design provided us with real-life experiences in applying theories we had only ever seen in coursework. One of the key points of the design process was using reasonable and proper assumptions to justify the models we used in our design calculations. The team needed to discuss which assumptions were reasonable, and the implications and limitations associated with each of them. Another key point was coming together as a team to define our selection criteria for the tested load values, structure material, appropriate failure theory, and in checking all of the calculations for accuracy. Lastly, the most crucial point of the project was working on a team using engineering models, assumptions, and a subsequent computational model to design and validate a structure, all while presenting our findings in a professional manner. In the transition from coursework to industry, we will be required to design projects in the real world. These projects are more ill-defined, teamwork-oriented, and all calculations and design choices need to be professionally presented and validated. We feel that all aspects of the work done in this project have supplied us with valuable practice in applying our skills in a comparable situation to those we will find in industry. Much like in industry, there were multiple reasons and times where the team needed to adjust our design path. After using static equilibrium equations to define our system, we created a computational model which could use inputs such as elastic modulus, member dimensions, compressive, ultimate, and tensile yield strength, and would output the internal and external forces and stresses of our entire system. We began to use the model, but our calculations were inconclusive, and we needed to reconsider our assumptions in creating our equilibrium equations. In the early stages of our design, we had assumed that the meeting point between the left and right leg would act as a fixed support, which would create a reactionary moment, as well as reactionary forces. After re-examining this assumption and our computational model, we realized that the “connection” between the right and left leg is more accurately modeled as a pin connection, containing only reactionary forces and no reactionary moment. We changed our design path to include this assumption, and were able to successfully represent our system using the computational model that we created.

References

- [1] *AISI 1018 CD Steel*. AISI 1018 Steel, Cold drawn. (n.d.). Retrieved March 13, 2023, from https://www.matweb.com/search/datasheet_print.aspx?matguid=3a9cc570fbb24d119f08db22a53e2421
- [2] *The healthy journal - gluten, dairy, sugar free recipes, interviews and health articles*. <https://www.thehealthyjournal.com>. (n.d.). Retrieved March 14, 2023, from <https://www.thehealthyjournal.com/frequently-asked-questions/how-much-force-can-a-human-exert>

Appendix

Appendix A: MATLAB Code

```
clear all

clc

%% Solving for forces-CORRECT

F = 100;%This is the applied force

% Force Order: Fwood, Ay, By, BCy, Dx, BCx, Cy, M, Below matrix made using
% my equations, posted as an image in box

Forces = [1,-1,0,0,0,0,0,0;
7.5,0,0,0,0,0,0,0;
-1,1,1,1,0,0,0,0;
0,0,0,0,-1,-1,0,0;
-12.75*cosd(52.85),24.5*cosd(52.85),0,16.75*cosd(52.85),13.5*sind(52.85),16.75*sind(52.85),0,-1;
0,0,0,-1,0,0,1,0;
0,0,0,0,1,1,0,0;
0,0,0,15.5*cosd(59.4),-12.5*sind(59.4),-15.5*sind(59.4),0,1;
0,0,1,0,0,0,1,0];

Fapplied = [F;23.25*F;0;0;0;0;0;F];

Result = Forces\Fapplied;

Fwood = Result(1);

Ay = Result(2);

By = Result(3);

BCy = Result(4);

Dx = Result(5);

BCx = Result(6);

Cy = Result(7);

Ma = Result(8);

% Stress Pre calculations
```

```

b = 1.5;
h = 1;
b_internal = 1.38;
h_internal = 0.88;
lzz = b*h^3/12 - (b_internal*h_internal^3/12);
%% Chopper - CORRECT - Needs a different lzz value using the shape of the chopper head
x = [0, 0, 15.75,15.75,23.25,23.25,0];
Fv = [0,-F,-F,-F+Fwood,-F+Fwood,-F+Fwood-Ay,-F+Fwood-Ay];
N = length(Fv);
lzz = 0.16325;
zero = zeros(N,1);
M = cumtrapz(x,Fv);
sigbend = M*(h/2)/lzz;
%%No axial stress in the chopper
%Fax = 0;
%Ax = ((b*h)-(b_internal*h_internal));
%stax = Fax/Ax;
chopmaxV = max(Fv);
chopmaxM = max(M);
chopmaxTao = max(abs(sigbend));
%chopstax = stax;
figure
subplot(3,1,1)
plot(x,Fv,x,zero,'linewidth',2)
ylabel('V [lbf]')
xlabel('Distance [in]')
title('Chopper Diagrams: Shear Force')
subplot(3,1,2)
plot(x,M,'linewidth',2)

```

```

ylabel('M [lbf*in]')
xlabel('Distance [in]')
title('Chopper Diagrams: Bending Moment')
subplot(3,1,3)
plot(x,sigbend,'linewidth',2)
ylabel('Stress [Psi]')
xlabel('Distance [in]')
title('Chopper Diagrams: Bending Stress')

%% Left leg - CORRECT

b = 1.5;
h = 1;
b_internal = 1.38;
h_internal = 0.88;

lzz = b*h^3/12 - (b_internal*h_internal^3/12);

x = [0,0,12.75,12.75,13.5,13.5,16.75,16.75,24.5,24.5];

Fv = [0,By*sind(37.15),By*sind(37.15),By*sind(37.15)-Fwood*sind(37.15),By*sind(37.15)-
Fwood*sind(37.15),By*sind(37.15)-Fwood*sind(37.15)+Dx*cosd(37.15),By*sind(37.15)-
Fwood*sind(37.15)+Dx*cosd(37.15),By*sind(37.15)-
Fwood*sind(37.15)+Dx*cosd(37.15)+BCy*sind(37.15)+BCx*cosd(37.15)+Ma,By*sind(37.15)-
Fwood*sind(37.15)+Dx*cosd(37.15)+BCy*sind(37.15)+BCx*cosd(37.15)+Ma,By*sind(37.15)-
Fwood*sind(37.15)+Dx*cosd(37.15)+BCy*sind(37.15)+BCx*cosd(37.15)+Ay*sind(37.15)+Ma];

N = length(Fv);
zero = zeros(N,1);
M = cumtrapz(x,Fv);
sigbend = M*(h/2)/lzz;

Fax = -By*cosd(37.15)+Fwood*cosd(37.15)+Dx*sind(37.15)+BCx*sind(37.15)-BCy*cosd(37.15)+Ay*cosd(37.15);
Ax = ((b*h)-(b_internal*h_internal));
stax = Fax/Ax;

leftmaxV = max(Fv);
leftmaxM = max(M);
leftmaxTao = max(sigbend);

```

```

leftstax = stax;
figure
subplot(3,1,1)
plot(x,Fv,x,zero,'linewidth',2)
ylabel('V [lbf]')
xlabel('Distance [in]')
title('Left Leg Diagrams: Shear Force')
subplot(3,1,2)
plot(x,M,'linewidth',2)
ylabel('M [lbf*in]')
xlabel('Distance [in]')
title('Left Leg Diagrams: Bending Moment')
subplot(3,1,3)
plot(x,sigbend,'linewidth',2)
ylabel('Stress [Psi]')
xlabel('Distance [in]')
title('Left Leg Diagrams: Bending Stress')
%% Right Leg - CORRECT
b = 1.5;
h = 1;
b_internal = 1.38;
h_internal = 0.88;
lzz = b*h^3/12 - (b_internal*h_internal^3/12);
x = [0,0,3,3,15.5,15.5];
Fv = [0,BCx*cosd(30.6)-BCy*sind(30.6),BCx*cosd(30.6)-BCy*sind(30.6),BCx*cosd(30.6)-
BCy*sind(30.6)+Dx*cosd(30.6),BCx*cosd(30.6)-BCy*sind(30.6)+Dx*cosd(30.6),BCx*cosd(30.6)-
BCy*sind(30.6)+Dx*cosd(30.6)+Cy*sind(30.6)];
N = length(Fv);
zero = zeros(N,1);
M = cumtrapz(x,Fv);

```



```

sigbend = M*(h/2)/Izz;
Fax = BCx*sind(30.6)+BCy*cosd(30.6)+Dx*sind(30.6)-Cy*cosd(30.6);
Ax = ((b*h)-(b_internal*h_internal));
stax = Fax/Ax;
rightmaxV = max(Fv);
rightmaxM = max(M);
rightmaxTao = max(abs(sigbend));
rightstax = stax;
figure
subplot(3,1,1)
plot(x,Fv,x,zero,'linewidth',2)
ylabel('V [lbf]')
xlabel('Distance [in]')
title('Right Leg Diagrams: Shear Force')
subplot(3,1,2)
plot(x,M,'linewidth',2)
ylabel('M [lbf*in]')
xlabel('Distance [in]')
title('Right Leg Diagrams: Bending Moment')
subplot(3,1,3)
plot(x,sigbend,'linewidth',2)
ylabel('Stress [Psi]')
xlabel('Distance [in]')
title('Right Leg Diagrams: Bending Stress')
%% Connector - CORRECT
% There is no shear, moment, or stress diagram as this is a 2 force member
constax = Dx/(0.25*1); % only axial stress or whatever this means, no other stresses
%% Stresses - Already calculated with each plot, this is here as an afterthought if needed
% Izz = b*h^3/12 - (b_internal*h_internal^3/12)

```

```

%bending stress tao_xy = M_max*(h/2)/Izz
b = 1.5;
h = 1;
b_internal = 1.38;
h_internal = 0.88;
Izz = b*h^3/12 - (b_internal*h_internal^3/12);
tao_xy_leftleg = leftmaxM*(h/2)/Izz;
%% Principal Stresses
%cutter
chopmaxTao;
%Left leg
lefts1 = leftstax/2 + ((leftstax/2)^2+leftmaxTao^2)^(1/2);
lefts2 = leftstax/2 - ((leftstax/2)^2+leftmaxTao^2)^(1/2);
%Right leg
rights1 = rightstax/2 + ((rightstax/2)^2+rightmaxTao^2)^(1/2);
rights2 = rightstax/2 - ((rightstax/2)^2+rightmaxTao^2)^(1/2);
%Connector
constax;
%% Factor of Safety
Sut = 53700;
ncutter = Sut/chopmaxTao;
nleft = Sut/lefts1;
nright = abs(Sut/rights2);
% nleft = 1/(lefts1/Sut - lefts2/Suc);
%nright = 1/(rights1/Sut - rights2/Suc);
nconnector = Sut/constax;
%% Strain
E = 29E6;
ecutter = chopmaxTao/E;

```

$e_{left} = \text{lefts1}/E;$

$e_{right} = -\text{rights2}/E;$

$e_{connector} = \text{constax}/E;$