Motorized Can Crusher

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Abstract. Can crushing is not a task that should require a person's full attention or strength to operate. By powering a can crusher with a motor, people can crush their cans without having to stand by or work a lever or other mechanism. This can crusher uses a motor to power the crushing through a worm gear and a crank slider mechanism. Doing so creates a small and effective device which crushes cans to a more dense state ready for transportation and recycling.

1 Introduction

Transporting aluminum cans for recycling or practical purposes can be a cumbersome task. The extremely low density of these cans creates an issue due to the space they take up. Crushers have been designed and are sold by many companies to crush these cans into a smaller size for storage and transportation, but use human strength and attention. In large industrial situations crushers are built for items up to the size of cars and larger which require little attention and no human strength. The market is lacking a device for the average consumer to crush cans without using their own strength.

1.1 Problem Statement

Crushing pop cans can be a trivial task. Can crushers were made to help alleviate the effort required to crush cans, but they are not completely beneficial if a worker is required to spend time operating them. People have much more pressing matters to attend to so it would be most beneficial if they could leave the triviality and mechanical work of can crushing solely to a machine.

1.2 Conceptual Design

Our can crusher will be much like the design of existing conventional can crushers, but the input force will be transmitted by a motor. This design consists of a motor transmitting power to a shaft that drives a plunger into the crushing chamber. The sides of the crushing chamber have a track in which guides the plunger back and forth as it crushes cans. The motor will run at a constant speed to allows cans to be fed into the crushing chamber at an optimal pace. The cans will be fed through a feeding channel located above the crushing chamber such that a new can will fall in place as crushed cans are dumped out. Can are dumped out through a short opening at the far end of the crushing chamber sized so that only a crushed can will be able to fall through.

1.2.1 Features

This design uses a 230W motor to drive the plunger. The worm gear contains a gear ratio of 45:1. The feeding chamber can effectively hold up to 7 cans.

1.2.2 Requirements

For this machine to work well the cans must be crushed to a sufficient size, less than $\frac{3}{4}$in in height. Too slow of a machine would also be a problem. A machine which crushes a can in less than 5 seconds would be fine. Finally the machine should be able to crush most cans. While the crusher will be designed around a standard soda can it should fit other similar sizes of cans.

1.3 Design Effectiveness

Our design will be effective in allowing users to not worry about their can crushing needs. Once they dump their cans into the feeding channel, they can press on and walk away. The machine's high-cycle fatigue allows it to keep working without fail and ease the worries of the user. The simple and perfectly constrained design allows for cans to be crushed at an optimal rate. By only using 230W, the motor does not require much energy to power on.

2 Parts Analysis

Analysis is performed on each part using static and dynamic failure analysis equations from Shigley's Mechanical Engineering Design (**?**). All parts are designed with infinite life. The endurance limit of parts are calculated and modified accordingly using the (Marin Equation). All calculations are done using MATLAB code which can referenced in Figure 12.

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \qquad \text{(Marin Equation)}$$

$$k_a = aS_{ut}^b \qquad \text{(Surface Condition Modification Factor)}$$

$$k_b = \begin{cases} 0.879d^-0.107 & 0.11 \leqslant d \leqslant 2in \\ 0.91d^-0.157 & 2 \leqslant d \leqslant 10in \end{cases} \quad \text{(Marin Size Factor)}$$

2.1 Torsional Power Components

The first section of this design are the torsional power components that makeup the torsional power input into the system.

2.1.1 Worm Shaft

This shaft is one of the more complicated shafts with both torque around one axis, and force in all three axes. Because of this fact the standard Combined Distortion Energy and ASME Elliptic Curve equation, (DE-ASME Elliptic Diameter Equation), could not be used. Therefore the standard (DE Three-Dimensional Planar Stress Formula) was used to find both the alternating and midrange von Mises stress. These stresses were then used in the (ASME-elliptic Curve Equation). As is prudent the yield stress was also checked using the Langer line (shigley eq 6-49), in this case the yield stress was the limiting factor. Using AISI 1006 CD Steel this method showed a shaft with a $\frac{1}{8}$in diameter and a $\frac{1}{128}$in radius fillet to a $\frac{3}{16}$in diameter step to have a sufficient factor of safety.

$$\sigma' = \frac{1}{\sqrt{2}}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + \ldots \right.$$
$$\left. (\sigma_z - \sigma_x)^2 + 6\left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2\right)\right]^{\frac{1}{2}}$$
(DE Three-Dimensional Planar Stress Formula)

$$\left(\frac{n\sigma_a}{S_e}\right)^2 + \left(\frac{n\sigma_m}{S_y}\right)^2 = 1 \quad \text{(ASME-elliptic Line Equation)}$$

2.1.2 Worm Gear Shaft

The analysis for the worm gear shaft first required finding the yield strength, tensile strength, and the alternating and midrange moments and torques of the shaft. A factor of safety is first assumed to be 1.5. Using an assumed diameter for the shaft, the stress concentration and shear stress concentration factors are found using a standard table four round shafts with shoulder fillets in torsion (A-15-8 (?)). The assumed diameter is also used to solve the Marin factors and calculate the endurance limit of the shaft. The actual diameter of the shaft is then solved for using the calculated endurance limit and parameters using the (DE-ASME Elliptic Diameter Equation) The closest standard size shaft to the actual diameter calculated is used to solve for the stress and shear stress concentrations, and is also used to solve for the Marin factors and the endurance limit. The new endurance limit is used to solve for the factor of safety of the standard size shaft, using the same equation used to find the actual diameter.

2.1.3 Worm Gear Mesh

To convert rotational motion to translational motion, the design features a worm gear assembly. The worm gear runs with a ratio of 45:1 and a mean diameter of 1/2 of an inch. The gear teeth are individually capable of withstanding a tangential force of 25.62 lbf. To calculate this, we used the (Sliding Velocity) which involved determining various factors such as the (Velocity Factor) and the (Ratio Correction Factor).

$$(W^t)_{all} = C_s D_m^{0.8} F_e C_m C_v$$
(Allowable Tangential Force Equation)

$$C_v = \begin{cases} 0.659e^-0.0011 & V_s < 700\frac{ft}{min} \\ 13.31V_s^-0.571 & 700 \leqslant V_s \leqslant 3000\frac{ft}{min} \\ 65.52V_s^-0.774 & V_s > 3000\frac{ft}{min} \end{cases}$$
(Velocity Factor)

$$C_m = \begin{cases} 0.02\sqrt{-m_G^2 + 40m_G - 76} + 0.46 & 3 < m_G \leqslant 20 \\ 0.0107\sqrt{-m_G^2 + 56m_G + 5145} & 20 < m_G \leqslant 76 \\ 1.1483 - 0.00658m_G & m_G > 76 \end{cases}$$
(Ratio Correction Factor)

$$V_s = \frac{\pi n_w d_m}{12\cos\lambda} \quad \text{(Sliding Velocity)}$$

2.2 Translational Crushing Mechanism

The second part of this design is comprised of the translational motion of the crush plate.

2.2.1 Crush Plate Rolling Shaft

A 131 lb crushing force allows is optimal to crush cans of varying resistance while allowing the motor to maintain efficiently. We decided that using a distortion energy (DE) method would be most suitable for this calculation. In our initial design process, we shot for a factor of safety of 1.5. With this factor of safety, the optimal diameter was calculated to be 0.25676 of an inch using the (DE-ASME Elliptic Diameter Equation) and the (Marin Equation). The (Marin Equation) were influenced by the (Surface Condition Modification Factor) and the (Marin Size Factor). However, the calculated diameter value exceeded our intended 1/4 inch design so we increased our diameter size to 5/16 of an inch. A new factor of safety was recalculated to be 2.65.

$$d = \left\{ \frac{16n}{\pi} \left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + \ldots \right.\right.$$
$$\left.\left. 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(K_{fs} T_m S_y\right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$
(DE-ASME Elliptic Diameter Equation)

2.2.2 Torsional Shaft Short Extending Arms

The shorter extension arms are made of 1006 cold-drawn steel. These arms are the main parts in transforming the rotational motion of the motor into the translational motion of the crush plate. This part can be modeled as a beam with a force from the link on one end and with a fixed support on the other. Because they are only subject to bending forces, the analysis only required the calculation of the bending moment. The max moment can be used to find the stress in the beam. This stress can then be used in the (ASME-elliptic Curve Equation) to find the factor of safety.

$$\sigma = Mc/I \qquad \text{(Bending Moment)}$$

2.2.3 Torsional Shaft Long Extending Arms

The arms attaching the shaft to the motor are prone to buckling failure due to the central loading parallel to the sides. Therefore, they are able to be modeled as columns subject to central loading. Using the (Central Column Loading Equation), we estimated the critical load point of unstable bending. To provide infinite life, the arms are made of 1006 AISI cold-drawn steel which holds an ultimate tensile strength of 48 ksi.

$$\frac{P_{cr}}{A} = \frac{C\pi^2 E}{\left(\frac{l}{k}\right)^2} \qquad \text{(Central Column Loading Equation)}$$

A failure analysis was performed on the arm to determine the factor of safety. For failure analysis, the ultimate tensile strength of steel is used to determine the endurance limit. By applying Marin Factors to the endurance limit, a modified endurance limit can be calculated. The principal stresses are then calculated by dividing the force required to crush a can over the cross-sectional area. This was seen to have a higher factor of safety than bending. Therefore the part was designed using the (Central Column Loading Equation). The final iteration had a factor of safety of 1.71.

2.2.4 Crush Plate

A 1006 AISI hot-rolled steel crush plate will exert 131 lb of force on the cans. In calculating the stress of the plate, we treat the can as a distributed axial load on a beam. The plate is modeled as a beam as seen in Figure 1. Through use of shear force and bending moment diagrams, seen in Figure 2 , we calculated the maximum bending moment to be 45 lb-in.

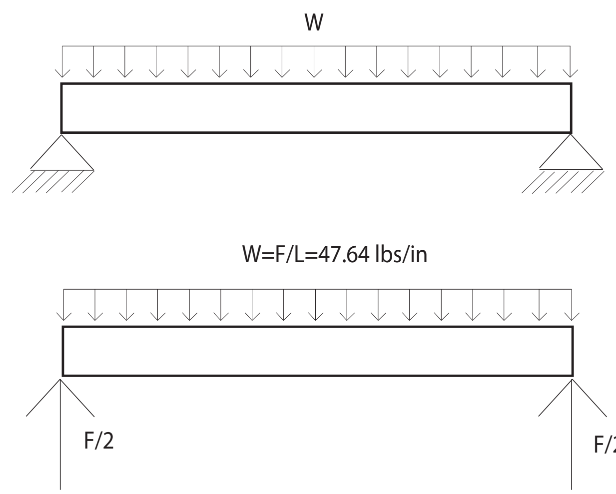

Figure 1: Crush plate beam model with distributed load.

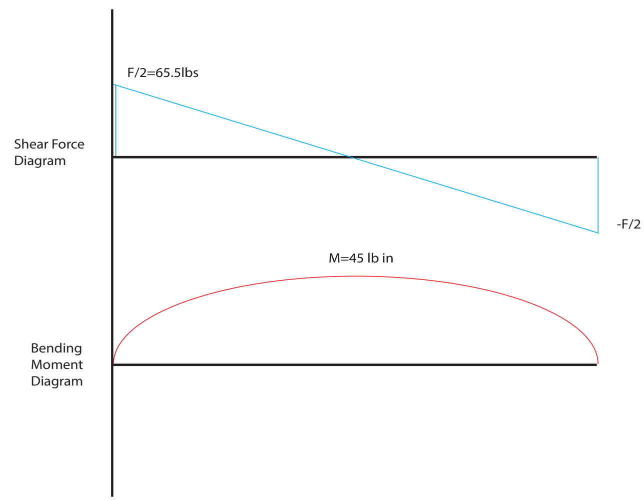

Figure 2: Shear force and bending moment diagrams for crush plate beam model.

$$\left(\frac{n\sigma_a}{S_e}\right)^2 + \left(\frac{n\sigma_m}{S_{ut}}\right)^2 = 1 \quad \text{(ASME-elliptic Curve Equation)}$$

2.3 Housing Shell & Mating

The shell housing is comprised of sheet metal with a thickness of 0.125 inches for increased durability. The parts are joined together using a combination of bolts and rivets.

2.3.1 Rivets

The thickness, grip, radius, and moment of inertia were used with the mechanical properties of steel (i.e. the ultimate tensile strength and the yield strength) to calculate the maximum force the rivet can withstand. Rivets will fail due to shear and the shearing force is defined as the product of the yield strength and the cross-sectional area, divided by the factor of safety. Because the factor of safety is used to calculate the shearing force, and the shearing force is used to calculate the factor of safety, an arbitrary value for the factor of safety was used. We started with a factor of safety of 1.5, yielding a total force of 1500 lbs. The maximum shearing force would occur when the factor of safety is less than one. Figure 3 represents the application of a rivet and the direction of forces applied. As seen in parts (a), (b), and (c) in Figure 3, the rivet will fail due to shear force. A failure analysis was then performed to validate the factor of safety determined previously. Similar to the failure analysis of the extending arm, many factors were considered. The ultimate tensile strength was used to determine the endurance limit. Marin Factors were then applied to the endurance limit to produce a modified endurance limit. The principal stresses were also calculated and are defined as the total force divided by the cross-sectional area. With the principal stresses and the endurance limit, the factor of safety can be calculated. The factor of safety was determined to be 1.44.

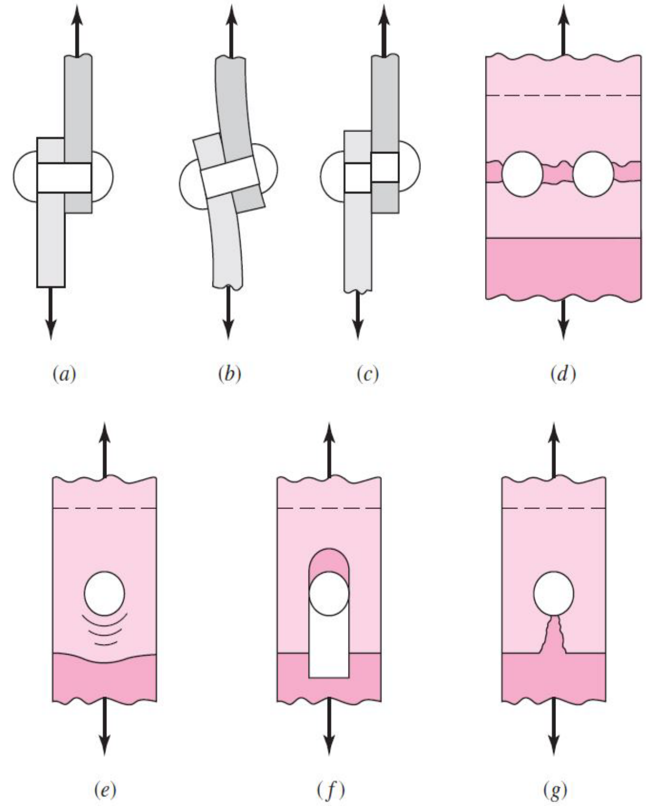

Figure 3: Rivets in shear failure (?).

2.3.2 Bolts

There are many parameters to consider when analyzing a bolt. The pitch is the distance between adjacent threads measured parallel to the thread axis. The major diameter d is the largest diameter of the threaded section as seen in Figure 4. The minor diameter dr is the smallest diameter of the threaded section. The differences in diameters results in a nominal diameter, in which the nominal area can be calculated. Along with parameters regarding the shape of the bolt, mechanical properties also need to be considered. The minimum tensile strength and minimum proof strength are used to calculate the preload within the bolt. The bolt stiffness and member stiffness is also needed and is defined by the length of the threaded portions. The deflection of the bolt is defined as the quotient of the force in the bolt over the bolt stiffness. The deflection of the member is equivalent to the deflection of the bolt, therefore the force in the member can be expressed as a function of the force in the bolt. The force applied to the bolts is equivalent to the force required to crush a can. The total force to the bolt and member is a sum of the two. A script was written in MATLAB that calculates the preload, total force, shear stress, and factor of safety. The factor of safety is defined as the ratio of the preload to the total force. In our analysis, a factor of safety of 1.78 was calculated.

References

Crusher.pdf

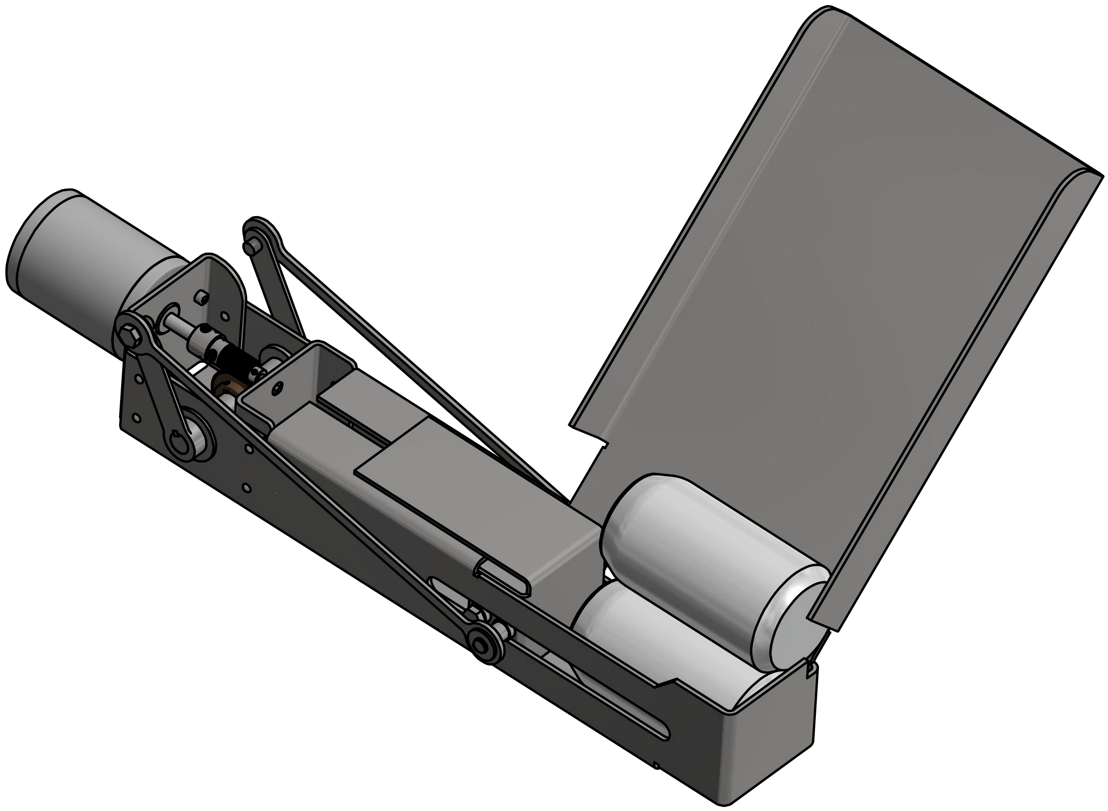

Figure 5: 3D parametric model of finished design.

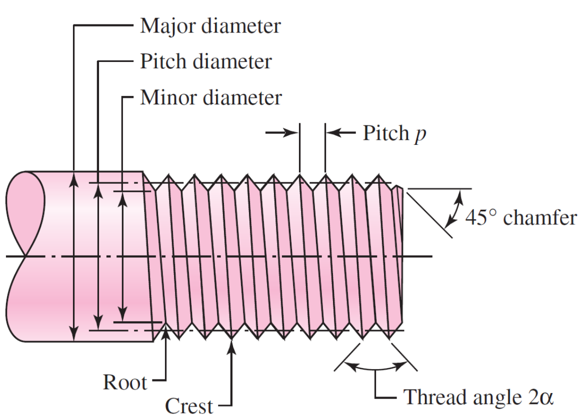

Figure 4: Graphic design of standard AISI bolt (?).

Crusher Parts 2.pdf

PARTS LIST				
ITEM	QTY	PART NUMBER	DESCRIPTION	MATERIAL
1	1	217-3371-STEP	Motor	Generic
5	1	base	Base	Steel, Mild
6	1	motor mount	Motor Mount	Steel, Mild
7	1	bearing mount	Worm Shaft Bearing Mount	Steel, Mild
8	1	crush plate	Can Crushing Plate	Steel, Mild
9	2	arm	Rotation to Translation Arm	Steel, Mild
10	2	link	Arm-Crush Plate Link	Steel, Mild
11	2	Can	Soda Can	Aluminum 6061
12	4	bearing	5/16 X 1/2 X 5/32 Bearing	Steel, Mild
13	4	flanged bearing	5/16 X 1/2 X 5/32 Flanged Bearing	Steel, Mild
14	2	main shaft bearing	1/2 X 1 1/8 X 5/16 Flanged Bearing	Steel, Mild
15	2	spacer	Crushing Plate Shaft Spacer	Steel, Mild
16	1	crush plate shaft	Crushing Plate Shaft	Steel, Mild
17	2	crush plate washer	Crushing Plate Shaft Washer	Steel, Alloy
19	4	crush plate bolt	5/16 X 1/2 Bolt	Steel, Alloy
20	2	97633A150	5/16 Retaining Ring	Steel, Mild
24	1	worm gear shaft	Worm Gear Shaft	Steel, Mild
25	1	s1c86z-p064b090d	90 tooth 64 Diametrical Pitch Worm Gear	Bronze, Cast
26	1	s51fczm040080	8mm to 1/8in Fairloc Coupling	Stainless Steel
27	1	s1d94z-p064sd	2 Lead 64 Diametrical Pitch RH Thread Worm	Stainless Steel
28	1	worm shaft	Worm Shaft	Steel, Mild
29	1	worm shaft bearing	1/8 X 5/16 X 9/64 Flanged Bearing	Steel, Mild
30	3	Key way	1/8 X 1/8 Key	Steel, Mild
31	8	CNS 7339 - 3 x 12	Rivet	Steel, Mild
32	2	Unbrako - 8 x 1 1/4	Hexagon Socket Head Cap Screw	Steel, Mild

Figure 6: Bill of materials for can crusher design.

Crusher Numbers.pdf

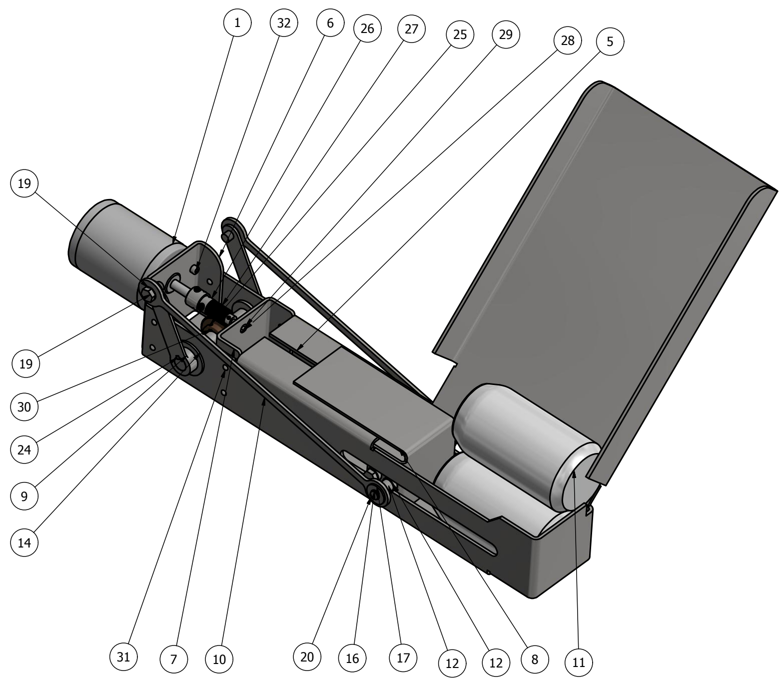

Figure 7: Labeled 3D parametric model of finished design.

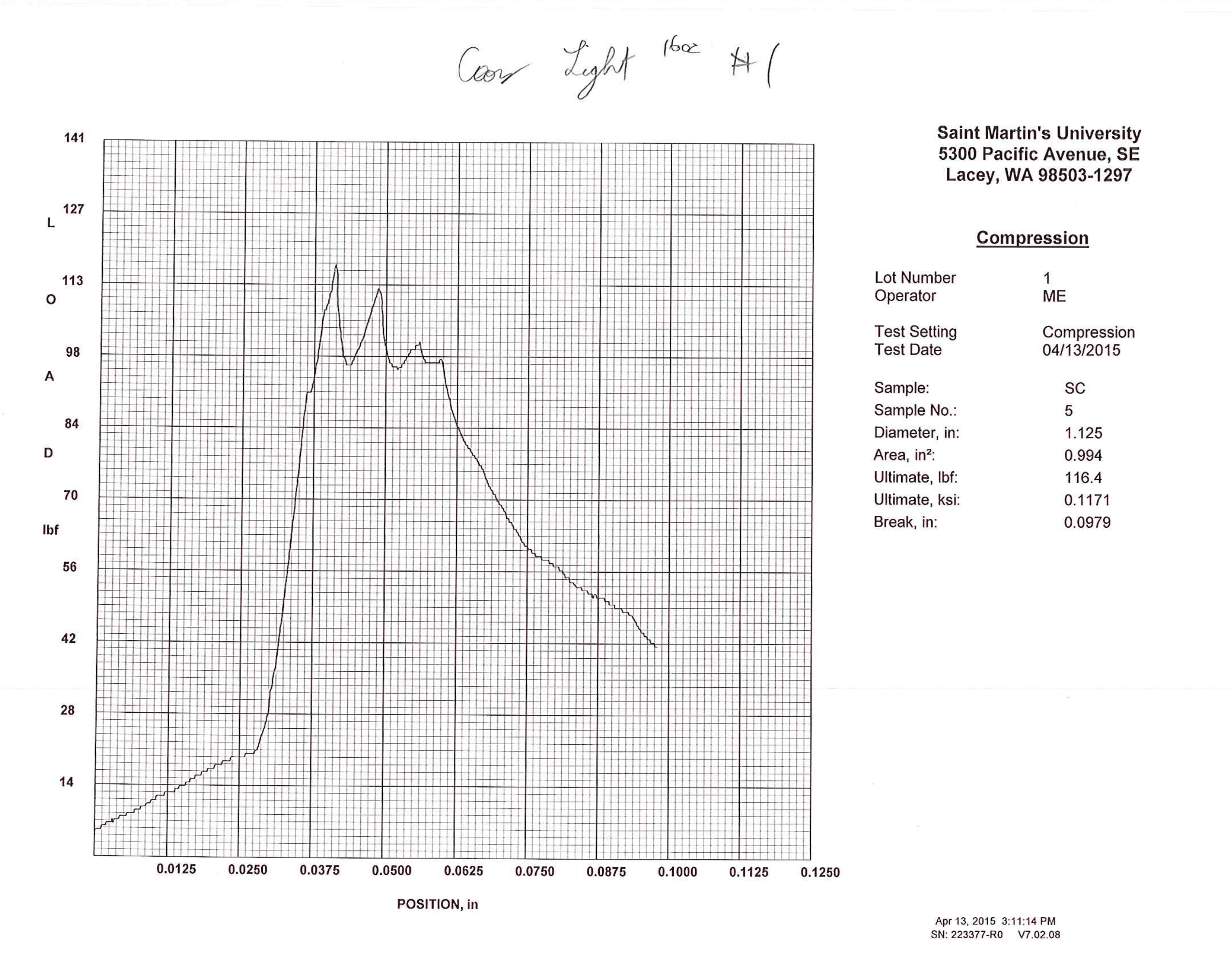

Figure 8: Coors Light Can Crushing Test #1.

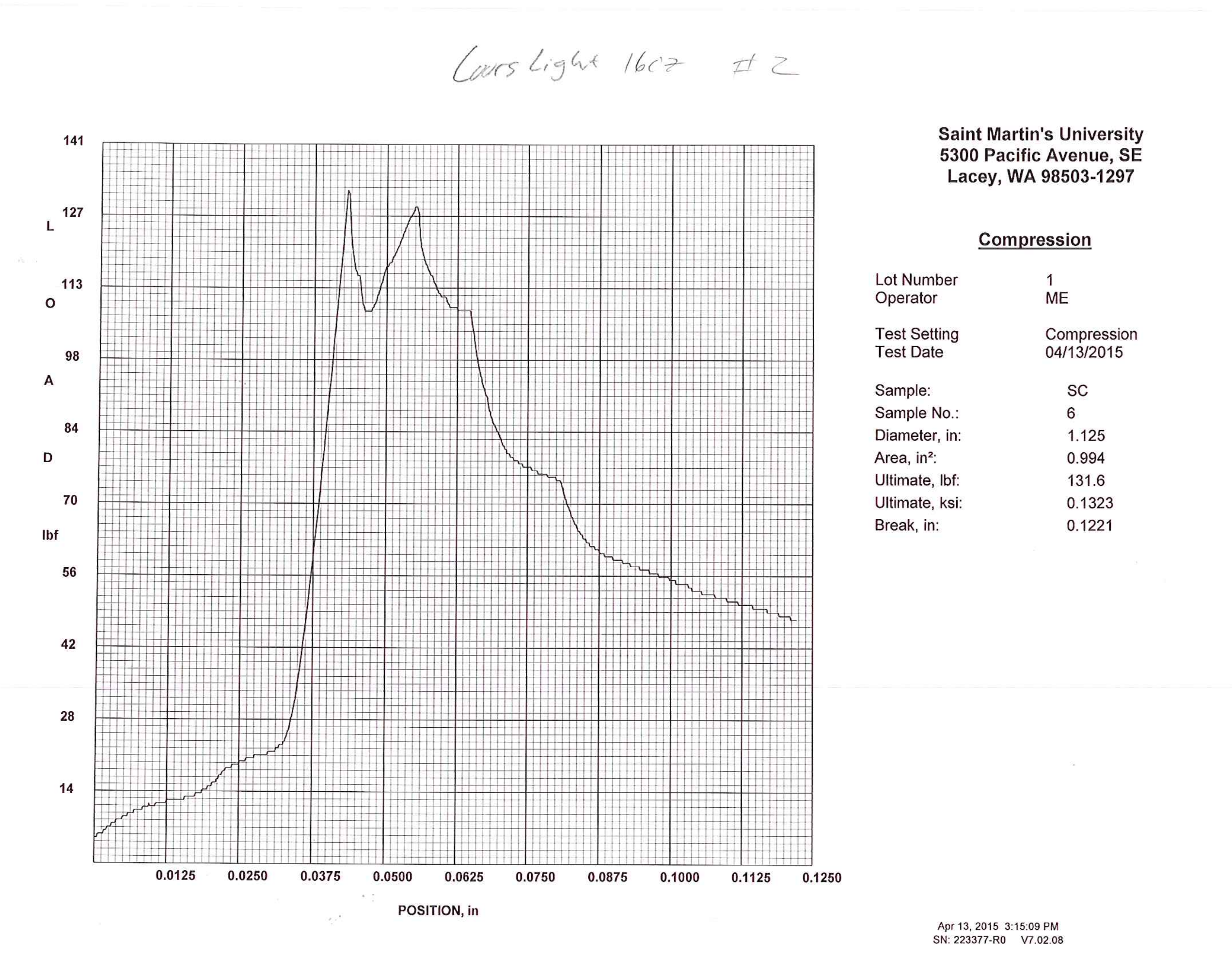

Figure 9: Coors Light Can Crushing Test #2.

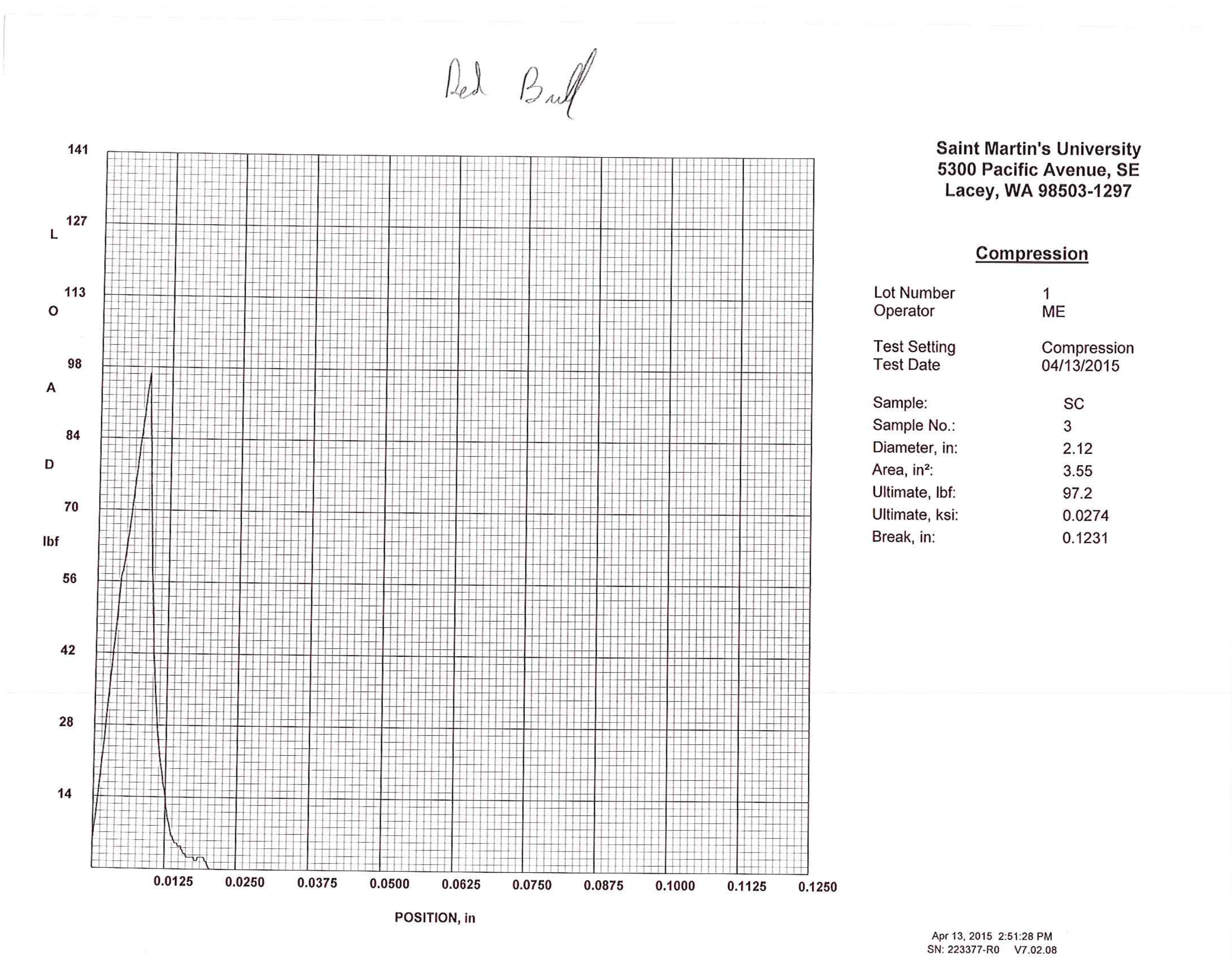

Figure 10: Red Bull Can Crushing Test.

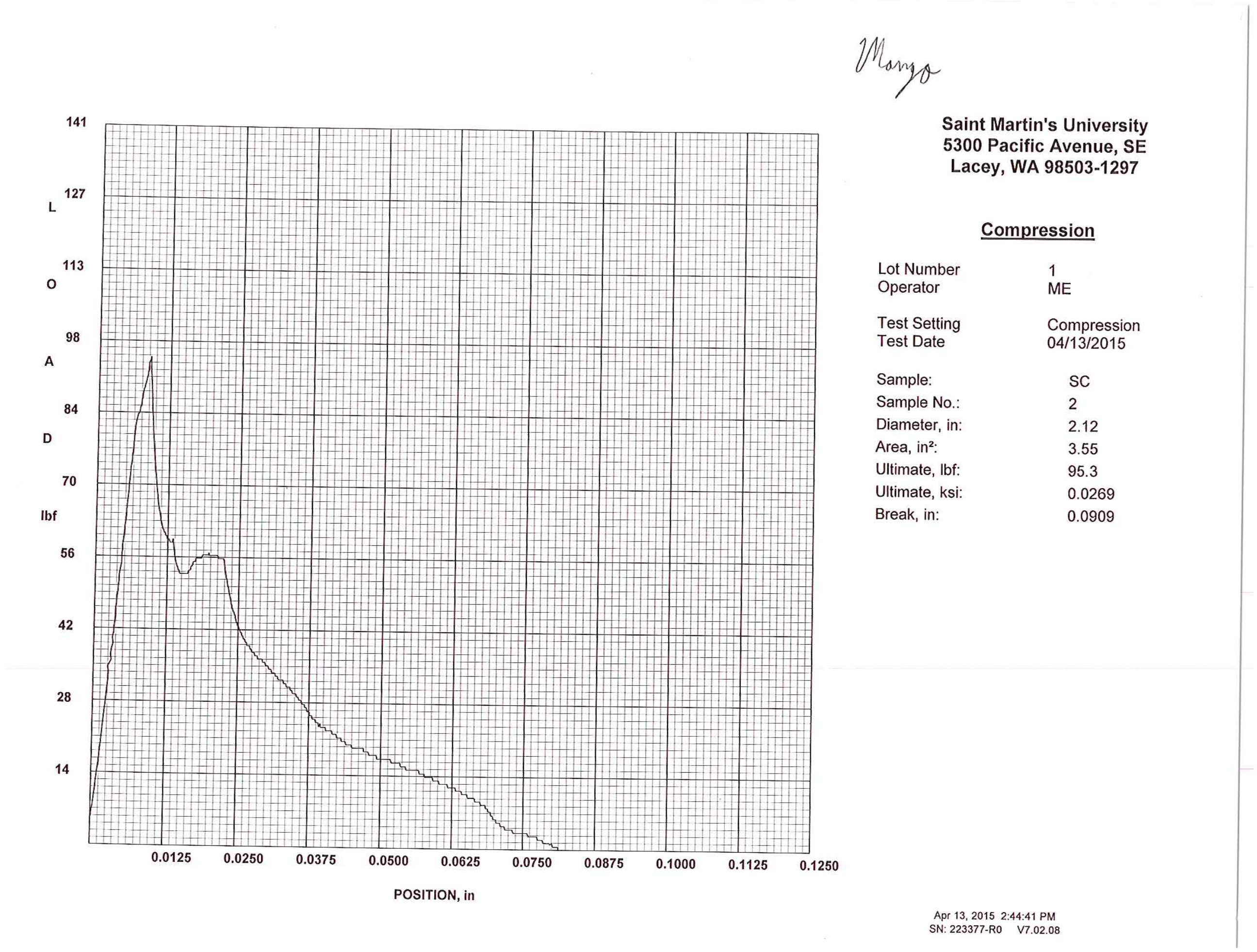

Figure 11: Mango Can Crushing Test.

Table of Contents


```
%MATLab Code
```

Extending Arm Failure

```
clear all; close all; clc;

Sy=41*10^3; %Yield Strength, psi
W=1; %Width of Can, inches
h1=.125; %Height, inches
A1=W*h1; %Cross Sectional Area
Pcr=(A1*(Sy/2))/1000; %Maximum Force

B=.25; %Thickness, inches
h2=.125; %Height, inches
I=(1/12)*B*h2^3; %Moment of Inertia, inches^3
A=B*h2; %Cross Sectional Area, inches^2
k=sqrt(I/A); %Radius of Gyration, inches
E=30*10^6; %Modulus of Elasticity, psi
L2=10.375; %Length, inches
C=1; %End-Condition Constant
Pcr2=((C*pi^2*E*A)/((L2/k)^2)) %Critical Force to Buckle, lbs
F=131/2; %Minimum Force to Crush Can
N2=Pcr2/F  %More Reliable Factor of Safety

% Failure Analysis
Sut=48*1000; %Ultimate Tensile Strength, Ksi
Se_prime=0.5*Sut; %Endurance Limit
a=2.7; %Surface Finish Factor
b=-0.265; %Exponent

%Marin Factors
ka=a*(Sut/1000)^b; %Surface Condition Modification Factor
kb=1; %Size Modification Factor
kc=0.85; %Load Modification Factor
kd=1; %Temperature Modification Factor
ke=1; %Reliability Factor
kf=1; %Miscellaneous-Effects Modification Factor
Se=ka*kb*kc*kd*ke*kf*Se_prime; %Modified Endurance Limit
```

1

Figure 12: Matlab Code Page 1.

```
sigmaMAX=F/A; %Maximum Stress due to Crushing
sigmaA=sigmaMAX/2; %Alternating Stress
sigmaM=sigmaA; %Minimum Stress is Zero
N1=sqrt((sigmaA/Se)^2+(sigmaM/Sy)^2);
n=1/N1 %Factor of Safety
```

```
Pcr2 =

  111.9267


N2 =

    1.7088


n =

   16.9750
```

Link Arm Failure

```
clear all; close all; clc;

Sy=41*1000; %Yield Strength, psi
Sut=48*1000; %Ultimate Tensile Strength, psi
M=180.125; %Moment
width=.125; %Width, inches
height=.75; %Height, inches
c=height/2; %Centroid
I=width*height^3/12; %Moment of Inertia, inches^3
sigma=M*c/I; %Stress
sigma_a=sigma/2; %Alternating Stress
sigma_m=sigma/2; %Midrange Stress

%Marin Factors
Ka=2.7*(Sut/1000)^-0.265; %Surface Condition Modification Factor
d=0.808*(height*width)^(1/2); %Diameter
Kb=0.879*(d)^-0.107; %Size Modification Factor
Kc=1;  %Load Modification Factor
Kd=1; %Temperature Modification Factor
Ke=1; %Reliability Factor
Kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf; %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety

width=.125; %Width, inches
height=11/16 %Height, inches
c=height/2; %Centroid
I=width*height^3/12; %Moment of Inertia, inches^3
```

2

Figure 13: Matlab Code Page 2.

```
sigma=M*c/I; %Stress
sigma_a=sigma/2; %Alternating Stress
simga_m=sigma/2; %Midrange Stress
d=0.808*(height*width)^(1/2); %Diameter
Kb=0.879*(d)^-0.107; %Size Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf; %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety

width=.09375; %Width, inches
height=11/16 %Height, inches
c=height/2;  %Centroid
I=width*height^3/12; %Moment of Inertia, inches^3
sigma=M*c/I; %Stress
sigma_a=sigma/2; %Alternating Stress
simga_m=sigma/2; %Midrange Stress
d=0.808*(height*width)^(1/2); %Diameter
Kb=0.879*(d)^-0.107; %Size Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf; %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety
n=Sy/sigma %Factor of Safety
```

```
n =

    2.6707


height =

    0.6875


n =

    2.3405


height =

    0.6875


n =

    1.8593


n =

    1.6810
```

3

Figure 14: Matlab Code Page 3.

Link Arm Bolt Analysis

```
clear all; close all; clc;

d=.3125; %Diameter
A=(pi/4)*d^2; %Original Area
Anom=A-.045; %Nominal Area (effective area)
Sp=36*10^3; %Minimum Proof Strength
kb=1.79; %Bolt Stiffness
km=11.33; %Member Stiffness
C=.114; %Stiffness Joint Factor
Fi=0.75*Anom*Sp; %Preload on Bolt
Pb=65.5; %Force on Bolt, lbs (min force to crush can over 2 bolts)
Fb=Pb+Fi; %Loading on Bolt
Pm=Pb*(km/kb); %Force on Member
Fm=Pm-Fi; %Loading on Member
Total_Force=(Pm+Pb) %Total Force, ksi

%Failure Analysis
Factor_Of_Safety=Fi/Total_Force %Factor of Safety
Shear_Stress=Pb/A %Shear Stress, Ksi
```

```
Total_Force =

  480.0894


Factor_Of_Safety =

    1.7827


Shear_Stress =

  853.9872
```

Rivet Analysis

```
clear all; close all; clc;

t=.125; %Thickness of Sheet Metal
tg=t*2; %Grip Thickness
r=.25; %Radius of Rivet
%Sy=41*10^3 %Yield Strength, psi
Sy=36000; %Yield Strength, psi
A=r*2*t; %Area of Rivet
I=(pi/4)*r^4; %Moment of Inertia
C=r; %Centroid
n1=1.5; %Factor of Safety
Total_Force=(Sy*A)/n1 %Maximum Shear Force
Max_Bending_Moment=(Total_Force*t)/2 %Maximum Bending Moment
```

4

Figure 15: Matlab Code Page 4.

```
%The maximum shear stress is Sy/2, where n=1

%Failure Analysis
Sut=48*1000; %Ultimate Tensile Strength, Ksi
Se_prime=0.5*Sut; %Endurance Limit
a=2.7; %Surface Finish Factor
b=-0.265; %Exponent

%Marin Factors
ka=a*(Sut/1000)^b; %Surface Condition Modification Factor
kb=1; %Size Modification Factor
kc=0.85; %Load Modification Factor
kd=1; %Temperature Modification Factor
ke=1; %Reliability Factor
kf=1; %Miscellaneous-Effects Modification Factor
Se=ka*kb*kc*kd*ke*kf*Se_prime; %Modified Endurance Limit

sigmaMAX=Total_Force/A; %Maximum Stress
sigmaA=sigmaMAX/2; %Alternating Stress
sigmaM=sigmaA; %Minimum Stress is Zero
N=sqrt((sigmaA/Se)^2+(sigmaM/Sy)^2);
Factor_Of_Safety=1/N %Factor of Safety
```

```
Total_Force =

        1500


Max_Bending_Moment =

   93.7500


Factor_Of_Safety =

    1.4427
```

Crush Plate Failure

```
clear all; close all; clc;

Mm=45/2 %Midrange Moment
Ma=45/2 %Alternating Moment
width=2.55; %Width, inches
height=0.125; %Height, inches
I=(width*height^3)/12; %Moment of Inertia, inches^3
c=height/2; %Centroid
sigma_m=Mm*c/I; %Midrange Stress
sigma_a=Ma*c/I; %Alternating Stress
Sut=43*1000; %Ultimate Tensile Strength, psi
Sy=24*1000; %Yield Strength, psi
```

5

Figure 16: Matlab Code Page 5.

```
%Marin Factors
Ka=14.4*(Sut/1000)^-0.718; %Surface Condition Modification Factor
de=0.808*(height*width)^(1/2); %Diameter
Kb=0.879*de^-0.107; %Size Modification Factor
Kc=1;  %Load Modification Factor
Kd=1; %Temperature Modification Factor
Ke=1; %Reliability Factor
Kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety

height=0.0625; %Height, inches
I=(width*height^3)/12; %Moment of Inertia, inches^3
c=height/2; %Centroid
sigma_m=Mm*c/I; %Midrange Stress
sigma_a=Ma*c/I; %Alternating Stress
de=0.808*(height*width)^(1/2); %Diameter
Kb=0.879*de^-0.107; %Size Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety

height=0.09375; %Height, inches
I=(width*height^3)/12; %Moment of Inertia, inches^3
c=height/2; %Centroid
sigma_m=Mm*c/I; %Midrange Stress
sigma_a=Ma*c/I; %Alternating Stress
de=0.808*(height*width)^(1/2);
Kb=0.879*de^-0.107; %Size Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf %Modified Endurance Limit
n=1/sqrt((sigma_a/Se)^2+(sigma_m/Sy)^2) %Factor of Safety
n=Sy/(sigma_a+sigma_m) %Factor of Safety
```

```
Mm =

   22.5000


Ma =

   22.5000


Se =

   1.9881e+04


n =

    4.5186
```

6

Figure 17: Matlab Code Page 6.

```
Se =

   2.0632e+04


n =

    1.1544


Se =

   2.0189e+04


n =

    2.5649


n =

    1.9922
```

Crush Plate Shaft Failure

```
clear all; close all; clc;

Ta=0; %Alternating Torque
Tm=0; %Midrange Torque
Ma=22.73; %Alternating Moment
Mm=22.73; %Midrange Moment
n=1.5; %Factor of Safety
Sy=41*1000; %Yield Strength, psi
Sut=48*1000; %Ultimate Tensile Strength, psi

%Marin Factors
Ka=0.968; %Surface Condition Modification Factor
Kb=1.13; %Size Modification Factor
Kc=1;  %Load Modification Factor
Kd=1; %Temperature Modification Factor
Ke=1; %Reliability Factor
Kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf %Modified Endurance Limit

d=((16*n/pi())*((4*(Ma/Se)^2)+(3*(Ta/Se)^2)+(4*(Mm/Sy)^2)+(3*(Tm/Sy)^2))^(1/2))^(1
d=5/16 %Assumed Diameter
Kb=1.107; %Size Modification Factor
Se=(Sut/2)*Ka*Kb*Kc*Kd*Ke*Kf %Modified Endurance Limit
n=1/((16/(pi()*d^3))*((4*(Ma/Se)^2)+(3*(Ta/Se)^2)+(4*(Mm/Sy)^2)+(3*(Tm/Sy)^2))^(1/
```

7

Figure 18: Matlab Code Page 7.

```
Se =

   2.6252e+04


d =

    0.2504


d =

    0.3125


Se =

   2.5718e+04


n =

    2.8717
```

Motor Failure

```
clear all; close all; clc;

T=180.125*2; %Torque Needed at Arm
Tm=12.4; %Motor Torque
n=1.5; %Factor of Safety
N=T*n/Tm %Gear Ratio


N =

   43.5786
```

Worm Gear Shaft Failure

```
clear all; close all; clc;

Sy=41*1000; %Yield Strength, psi
Sut=48*1000; %Ultimate Tensile Strength, psi
Ma=56.5/2; %Alternating Moment
Mm=56.5/2; %Midrange Moment
Ta=180.125/2; %Alternating Torque
Tm=180.125/2; %Midrange Torque
n=1.5; %Factor of Safety

d=0.25; %Assumed Diameter
```

8

Figure 19: Matlab Code Page 8.

```
Kf=1.5; %Stess Concentration Factor
Kfs=1.5; %Shear Stress Concentration Factor

%Marin Factors
ka=2.7*((Sut/1000)^-0.265); %Surface Condition Modification Factor
kb=0.879*(d^-0.107); %Size Modification Factor
kc=1;  %Load Modification Factor
kd=1; %Temperature Modification Factor
ke=1; %Reliability Factor
kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*ka*kb*kc*kd*ke*kf; %Modified Endurance Limit

d=((16*n/pi())*(4*((Kf*Ma/Se)^2)+3*((Kfs*Ta/Se)^2)+4*((Kf*Mm/Sy)^2)+3*((Kfs*Tm/Sy)

d=.5; %Assumed Diameter
Kf=2.78; %Stess Concentration Factor
Kfs=3.36; %Shear Stress Concentration Factor

%Marin Factors
ka=2.7*((Sut/1000)^-0.265); %Surface Condition Modification Factor
kb=0.879*(d^-0.107); %Size Modification Factor
kc=1;  %Load Modification Factor
kd=1; %Temperature Modification Factor
ke=1; %Reliability Factor
kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*ka*kb*kc*kd*ke*kf; %Modified Endurance Limit

n=1/((16/(pi()*d^3))*(4*((Kf*Ma/Se)^2)+3*((Kfs*Ta/Se)^2)+4*((Kf*Mm/Sy)^2)+3*((Kfs*

d=.625; %Assumed Diameter
Kf=2.96; %Stess Concentration Factor
Kfs=3.48; %Shear Stress Concentration Factor

%Marin Factors
ka=2.7*((Sut/1000)^-0.265); %Surface Condition Modification Factor
kb=0.879*(d^-0.107); %Size Modification Factor
kc=1;  %Load Modification Factor
kd=1; %Temperature Modification Factor
ke=1; %Reliability Factor
kf=1; %Miscellaneous-Effects Modification Factor
Se=(Sut/2)*ka*kb*kc*kd*ke*kf; %Modified Endurance Limit

n=1/((16/(pi()*d^3))*(4*((Kf*Ma/Se)^2)+3*((Kfs*Ta/Se)^2)+4*((Kf*Mm/Sy)^2)+3*((Kfs*
```

```
d =

    0.4526


n =

    1.6052
```

9

Figure 20: Matlab Code Page 9.

Worm Gear

Gear ratio 45:1

```
C=0.953; %gear d to D distance (in)
dm=0.5; % mean worm diamter good proportion; 0.319 <= d <= 0.599;
dm>=C^0.875/3
dm<=C^0.875/1.6
Dm=1.4063; %mean gear diameter

Cs=270+10.37*C^3; % materials factor; C <= 3 in
lambda=3.58; %lead angle
nw=6200; %gear velocity rpm
Vs=pi*nw*dm/(12*cosd(lambda)); %sliding velocity; ft/min
Cv=13.31*Vs^-0.571 %velocity factor
Fe=0.1875; %effective face width; Fe < 0.67dm; Fe <
Fe<0.67*dm
mG=45/1; %gear ratio
Cm=0.0107*sqrt(-mG^2+56*mG+5145); %ratio correction factor; 20 < mG <= 76
Wt=Cs*Dm^0.8*Fe*Cm*Cv %lb, allowable tangential force on worm-gear teeth

clc
```

```
ans =

     1


ans =

     1


Cv =

    0.2900


ans =

     1


Wt =

   16.0147
```

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10

Figure 21: Matlab Code Page 10.