ME 383 Final Report: The Apple Peelers

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ME 383

Dr. Anthony Nix

Introduction:

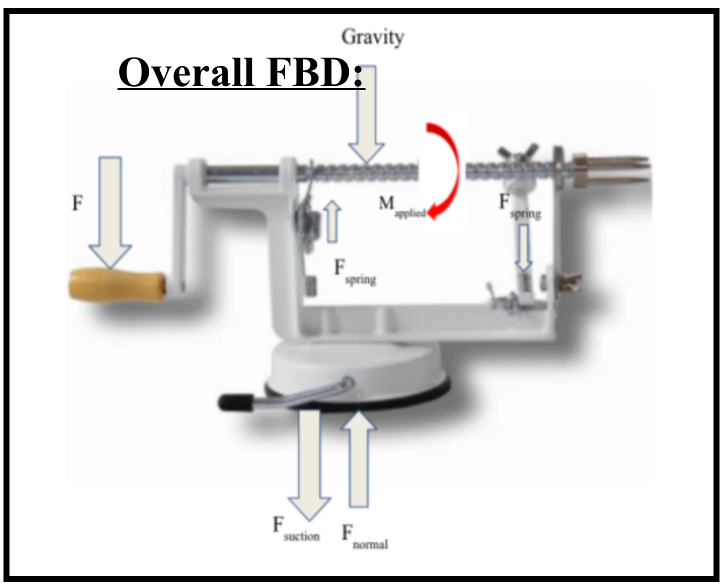
Our chosen product for this class is an apple peeler. As the name suggests, the main purpose the apple peeler serves is that it peels the skin off the apple by using a crank arm, shaft, and blade. The apple peeler works by placing the product on a surface, where the base will suction itself to remain stable. You can then place an apple into the sharp prongs at the edge and turn the crank arm to have the fruit peeled in a consistent direction along the shaft and blade. As for the target market, the product is mostly marketed towards the typical household consumer and household user that wants to cut their apple peeling time in half for cooking and eating.

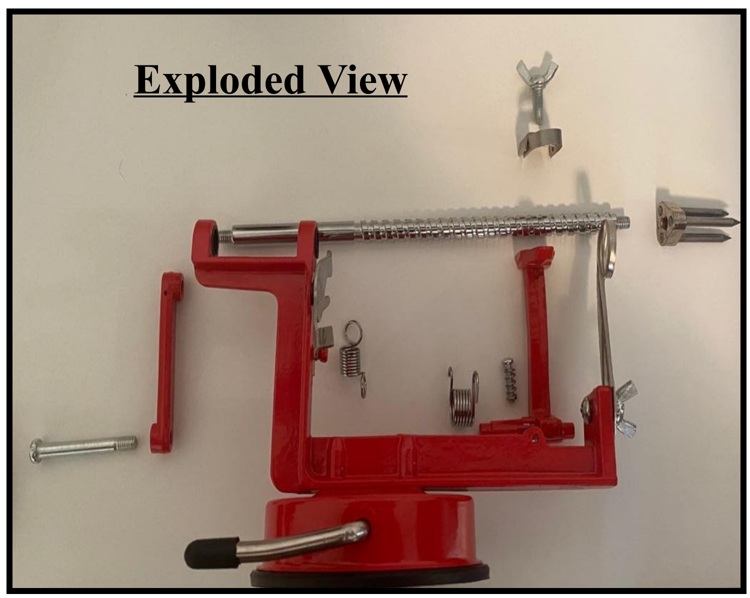
In this report we analysed the load, stress, and fatigue of 6 different mechanically driven parts of the apple peeler in order to determine how well each component performs in operation, under load. This is important as the design and functionality of our product relies heavily on these components not failing in regards to either a static overload, which will plastically deform the component, or fatigue failure where the component stops working after a small number of cycles. Specifically, each component was justified in their selection based on a strong relation to an engineering requirement, customer requirement, or FMEA. For the report, each individual worked on a component, undergoing a load analysis, stress analysis, and factor of safety examination for yielding and fatigue. The report below summarizes how we utilized mechanical component design equations in order to come to a conclusion on the values of each analysis, and design recommendations on ways to improve each element of the apple peeler.

FMEA:

Component Name	Function	Failure Mode	Failure Effect	Failure Cause	Severity [1 - 10]	Occurrence [1 - 10]	Detection [1 - 10]	Risk Priority Number
Tension Spring	Allow an outside force (by apple) to create tension and use its stored energy to pull the peeling arm back.	Fracture failure of spring/from corrosion	Can’t locate the crank rod	Fatigue and excessive load	7	3	7	147
Peeling Blade Screw	To fasten the peeling blade for support during peeling	Over bending	Bolt fractures in threaded areas	Overloaded by applied force	5	2	6	60

Crank Arm	To transfer applied load at the handle arm in order to rotate the shaft for peeling.	Bending of arm over time	Less power delivered to shaft making it harder to peel apples	Too much force applied to crank arm	7	3	6	126
Peeling Shaft	Shaft that rotates from force of handle to peel apples on the fork threads.	Plane bending	Crack growth on surface.	Excessive bending load.	7	3	5	105
Shaft Lever Spring	The spring helps to suspend the shaft lever, which performs the bolt screw function to give the shaft the linear movement also along the rotational.	Corrosion	Harder to peel different sizes.	Foregin materials entering the spring.	7	3	6	126
Handle Bolt	To fasten the crank arm.	Excessive bolt thread bending.	Harder to use and harder to move the handle/crank arm.	Overuse and potential misalign ments	7	3	5	105





Overall FBD:

From our last lab report we have removed the moment applied from the handle as asked for from lab 2. The forces found for the overall free body diagram were simplified and based on the critical forces solved for each of the components (Crank Arm, Shaft Lever Spring, Peeling Shaft, Peeling Blade Screw, and Tension Spring). Other force components included are gravity/normal force elements acting on the machine as a whole. For instance, we have a force that acts on the handle and forces from the springs. Also, we have an internal force that's applied on the shaft lever spring at a specific angle. We neglected forces we decided were not necessary like the force of friction from the peeling of the apple and any operation force errors.

Exploded view:

The exploded view includes the components we are analysing and how they relate to their aspect of the apple peeler. The components include the Crank Arm, Shaft Lever Spring, Peeling Shaft, Peeling Blade Screw, and Tension Spring.

Table of Customer and Engineering Requirements Correlated with each other:

#	Customer Requirements (CR)	Engineering Requirements (ER)
1	Easy to Clean	Cleaning Time (sec)
2	Easy to Use	Number of Tools Required for The Assembly (#)
3	Inexpensive/ Affordable	Cost to Manufacture ($)
4	Efficient Use of Peeling	Length (m)
5	Durable	Width (m)
6	Portable	Weight (kg)
7	Easy to Skin Apples of All Sizes	Height (m)
8	Limited Hand Strength For Operation	Operation Force (N)
9	Safe to Use	Safety Factor Number (#)
10	Easy to Assembly	Assembly time (sec)

House of Quality:

Customer Requirements (CR)	Customer Weights	Cost to Manufacture ($)	Low Operation Force (N)	Weight (N)	Assembly Time (sec)	Cleaning Time (sec)	Safety Factor Number (#)	Tools required (# of tools)	Height (m)	Length(m)	Width(m)	Edge Radius (m)
	Direction	Engineering Specifications (ES)										
Easy to Use	8		9	3	3	3		3				
Affordable	7	9		3								
Easy Assembly	4				9			9				1
Easy to Clean	5					9		3	3	3	3	3
Efficient	8		6		1	1	1					
Durable	6	3	3	1			3					
Portable	9	1		3					3	3	3	1
Can Peel Apples of All Sizes	8		3			1			1	1	1	
Limited Hand strength	9		9									
Safe to use	7		3	1			3					1
ES Ranking Calculations	Raw score	90	264	85	68	85	47	75	50	50	50	35
	Scaled	090909	1	969697	757576	969697	030303	909091	939394	939394	939394	757576
	Relative Weight	10%	29%	9%	8%	9%	5%	8%	6%	6%	6%	4%
	Rank	2	1	3	6	3	10	5	7	7	7	11

Analyzing and justification of apple peeler components based on customer and engineering requirements (FMEA):

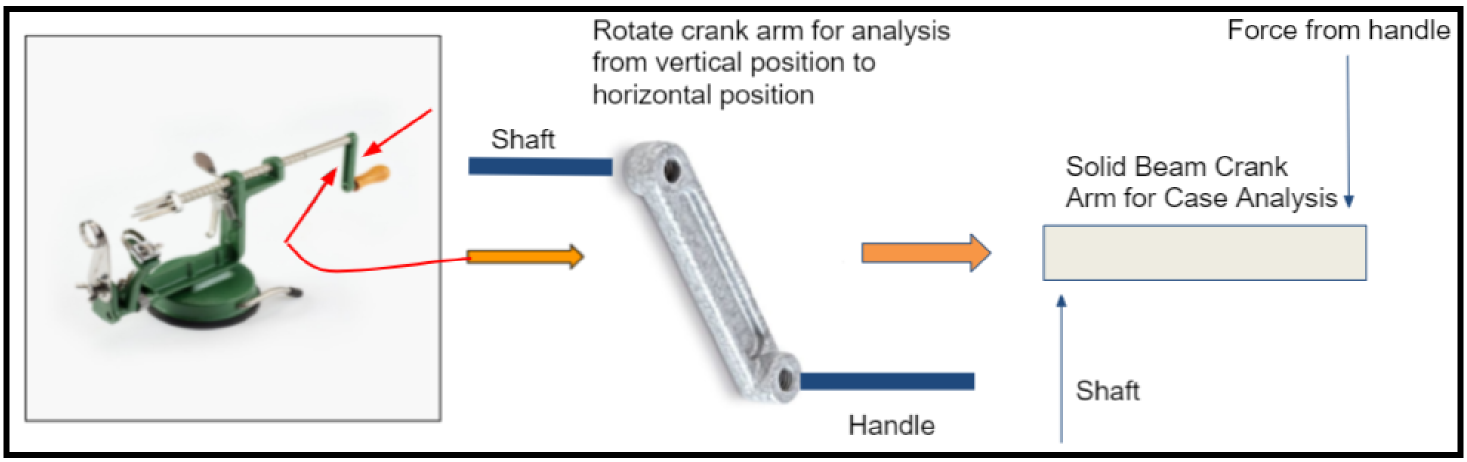
Name of the components	HOQ	FMEA
Crank Arm	Limited hand strength/low operation force	Bending failure
Tension Spring:	Limited hand strength/ cleaning time	Foregin substances entering the spring (Corrosion).
Handle Bolt:	Cleaning and assembly time	Excessive bending
Peeling blade screw:	Cleaning and assembly time	Foregin substances entering the Peeling Blade Screw (Corrosion).
Shaft lever spring:	Limited hand strength/ cleaning time	Foregin materials entering the spring (Corrosion).
Peeling Shaft:	Cleaning time/ Limited hand strength	Cyclical stresses and excessive load bending.

Crank Arm:

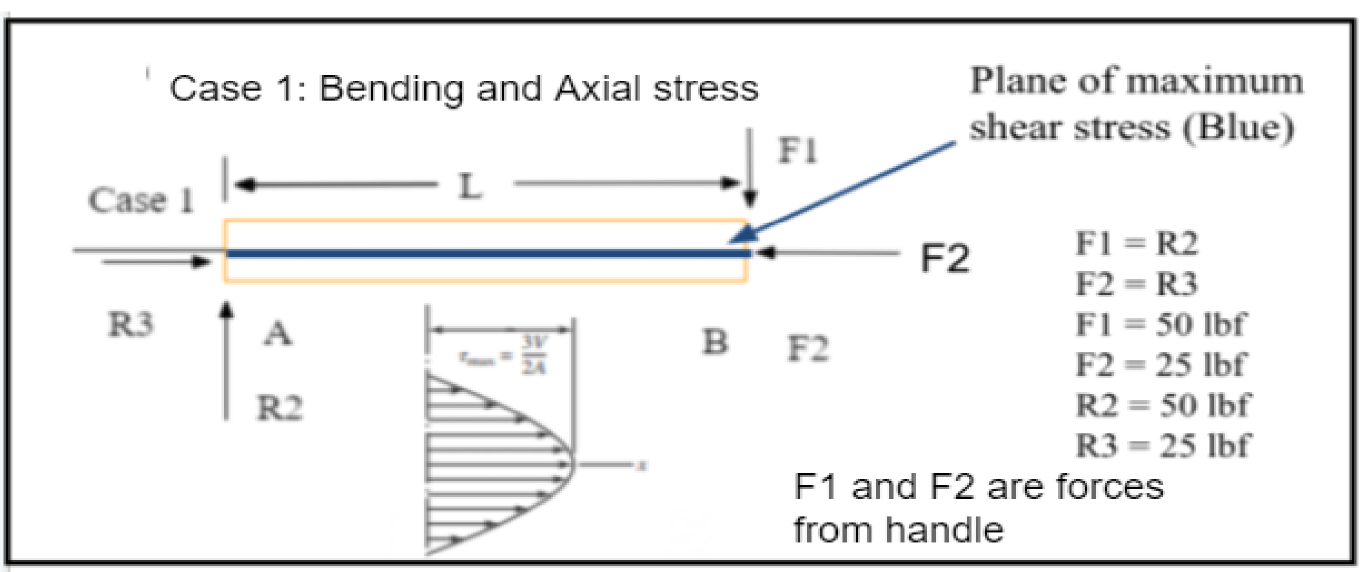
Material Selection: For this analysis of the crank arm the material chosen is 303 stainless steel. 303 stainless steel was selected because of its ductility due to being a low carbon steel and its cheap price tag relative to other steels available. 303 stainless steel is also very common in components such a crank arm as the steel is highly machinable. The material being stainless and made using chromium allows for less corrosion and wear over long periods of time which is perfect for the apple peelers application. The mechanical properties of 303 stainless steel are 35 kpsi for yield strength and 87.3 kpsi for ultimate tensile strength.

Key Assumptions:

1) Point forces act at the edges of crank arm
2) Treating crank arm like a uniform beam
3) Ignoring complex geometry at ends of crank arm
4) Analyzed at an angle normal to the handle



Case 1 FBD:



Initial Givens:

$A_{crank\ arm} = h * b = 0.125in^2$, $V = \frac{M}{L} = \frac{137.5(lbf-in)}{2.75in} = 50lbf$

Stress Calculations:

$\sigma_y = \frac{(50\ lbf)}{A_{beam}} = 400\ psi$, $\sigma_x = \frac{(25\ lbf)}{A_{beam}} = 200\ psi$, $\tau_{xy(Max)} = \frac{3V}{2A} = 600\ psi$

Planar Stress Calculations:

$$\sigma_1\sigma_2 = \frac{\sigma_{x+}\sigma_y}{2} +- \sqrt{(\frac{\sigma_x-\sigma_y}{2})^2 + \tau_{xy}{}^2} = (908.276, -308.276)psi$$

Static Safety Factor:

Sy = 35 kpsi, Sut = 87.3 kpsi, $\tau_{xy} = 600\ psi$, $\sigma_{bmax} = \frac{Mc}{I} = \frac{137.5lbf-in*0.2in}{0.00166in^4} = 16566psi = 16.566kpsi$

$\sigma_{ax} = -308\ psi$, $\sigma' = ((\sigma_{ax} + \sigma_{bend})^2 + 3\tau_{xy})^{\frac{1}{2}} = 16.313kpsi$, $\eta_{DE} = \frac{S_y}{\sigma'} = \frac{35\ kpsi}{16.313\ kpsi} = 2.145$

(Distortion energy for ductile materials such as 303 stainless steel)

Fatigue Safety Factor: (Using no alternating stress)

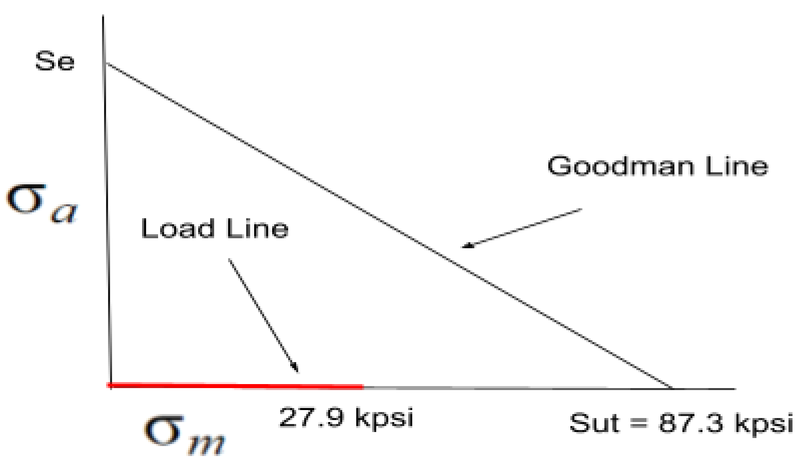
$\sigma_{mb} = 16.566\ kpsi, \sigma_{mA} = -308.276\ psi, \tau_m = 600\ psi$, $\sigma_m' = ([k_f\sigma_{mb} + k_f\sigma_{mA}]^2 + 3[k_{fs}\tau_m]^2)^{\frac{1}{2}} = 27.9\ kpsi$

$k_f = 1 + q(k_t - 1) = 1.715$, $k_{fs} = 1 + qs(k_{ts} - 1) = 1.576, k_t = 2.1, k_{ts} = 1.8,\ q = 0.65,\ qs = 0.72$

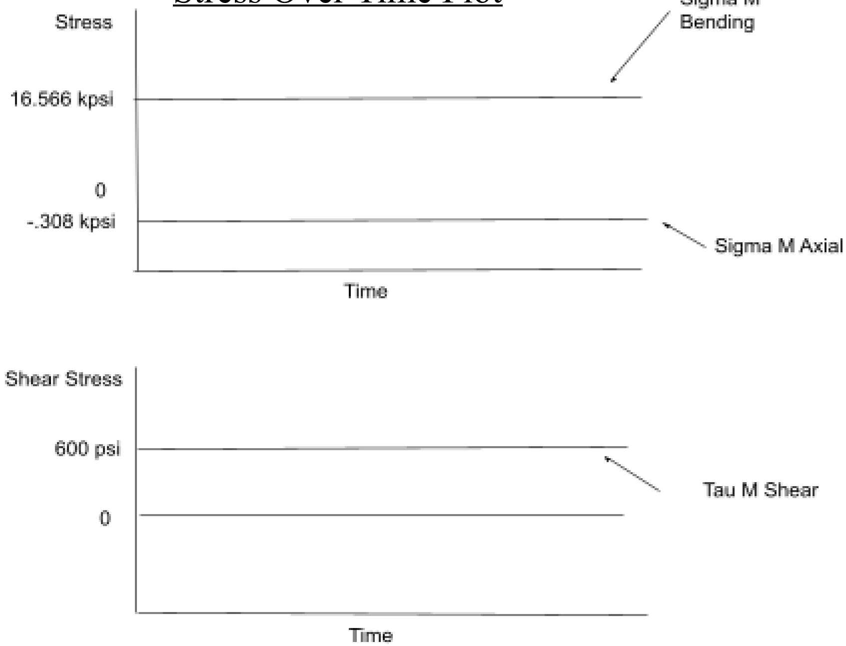
$n_f = (\frac{\sigma'_a}{Se} + \frac{\sigma'_m}{Sut})^{-1} = 3.126$ (Goodman fatigue factor of safety using no alternating stress)

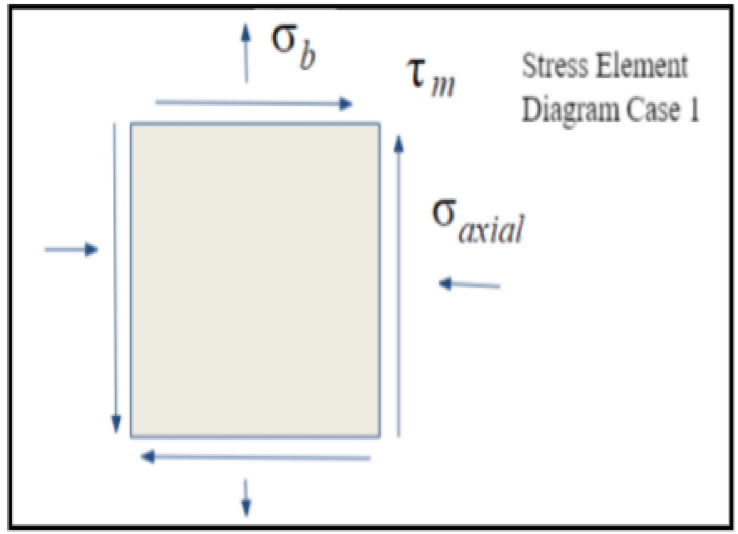
Sigma A-Sigma M plot:

- No alternating stress in plots due to forces being a constant transmission of stress on the crank arm during operation by the user.

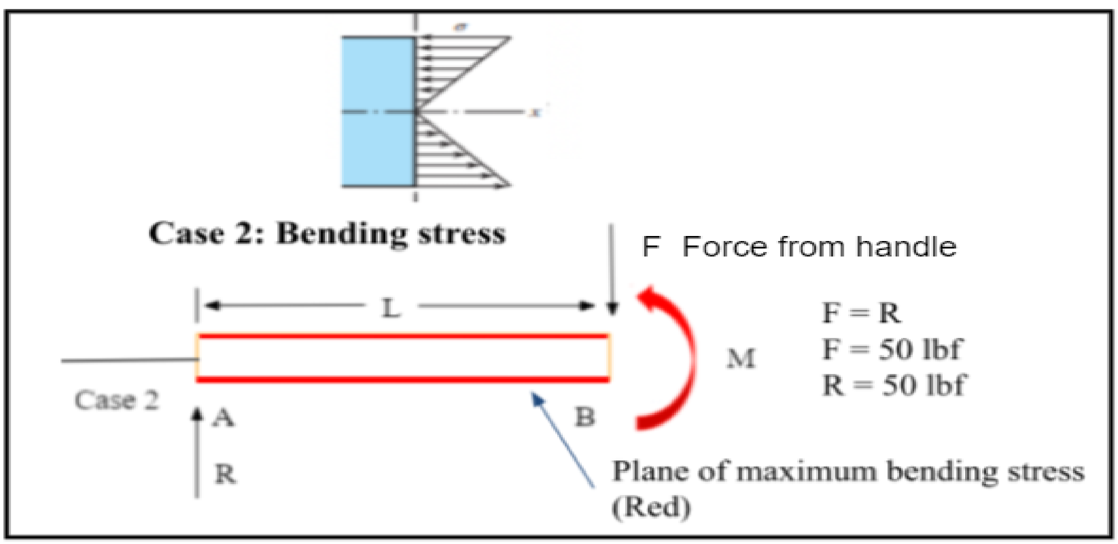


Stress Over Time Plot





Case 2 FBD:



Initial Givens:

L = 2.75 in , $A_{crank\ arm} \ = \ h * b$ = 0.125in^2 , F = 50lb, $\Sigma Fy \ = \ 0, \ F \ = \ R$
R = 50lb, $\Sigma M_A \ = \ 0, \ M_A \ = \ L * F \ = \ 137.5 lbf - in$

Stress Calculations:

$\sigma_{bmax} \ = \ \frac{Mc}{I} \ = \frac{137.5lbf-in*0.2in}{0.00166in^4} \ = 16566psi \ = \ 16.566kpsi,$

$I = \frac{1}{12}bh^3 = 0.00166in^4$, h = 0.4in, b = 0.313in, c = 0.2in,

Static Safety Factor:

Sy = 35 kpsi, Sut = 87.3 kpsi, $\eta_{MSS} = \frac{S_y}{\sigma_{mp} - \sigma_{mn}} = \frac{35\ kpsi}{16.566\ kpsi} = 2.113$

(Maximum shear stress theory for ductile materials such as 303 stainless steel)

Fatigue Safety Factor:

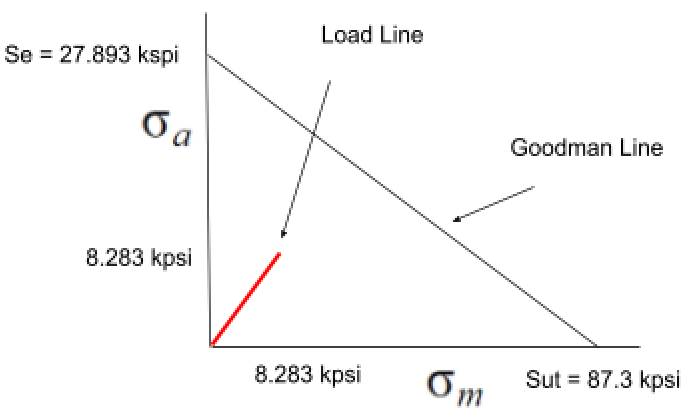
$\sigma_a = 8.283\ kpsi,\ \sigma_m = 8.283kpsi,\ Se' = .5 * Sut = 43.75kpsi,\ K_e = 0.868\ at\ 95\%$

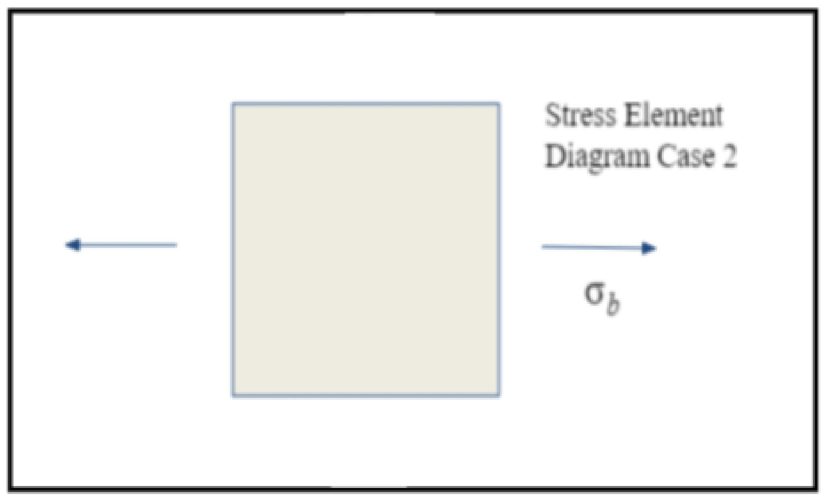
$K_a = aSut^b = 2 * Sut^{-0.217}(Cold\ Drawn) = 0.758,\ K_b = .879d^{(-.107)} = 0.969,\ K_c = 1(bending)$

$K_b = .879d^{(-.107)} = 0.969,\ K_c = 1(bending),\ Se = Se'K_aK_bK_cK_e = 27.893kpsi$

$n_f = (\frac{\sigma_a}{Se} + \frac{\sigma_m}{Sut})^{-1} = 2.55$ (Goodman fatigue factor of safety)

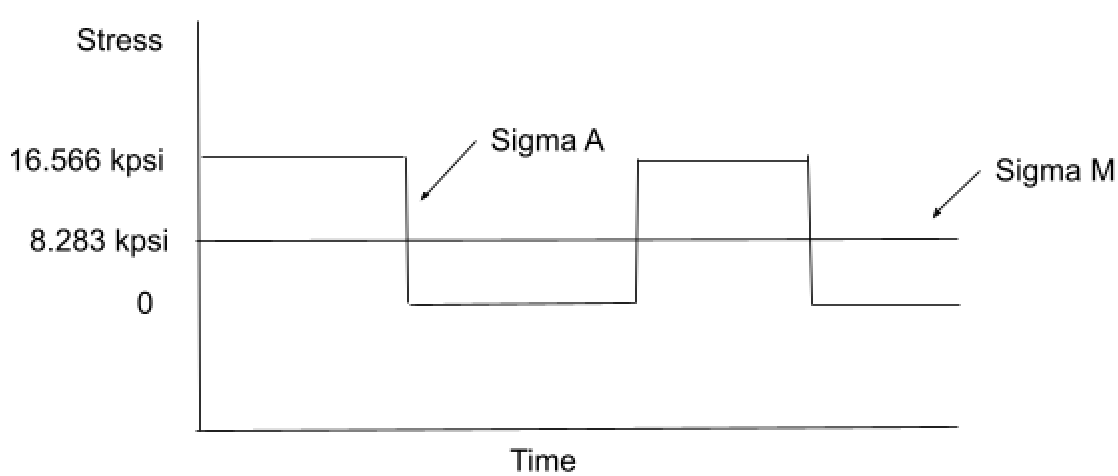
Sigma A-Sigma B plot:





Stress Over Time Plot:

- Step function plot based on stress applied by the user as the apples are peeled. A maximum stress is applied for a time interval and then a drop to zero stress acting on the crank arm as the user stops operation and then the cycle continues as more apples are peeled.



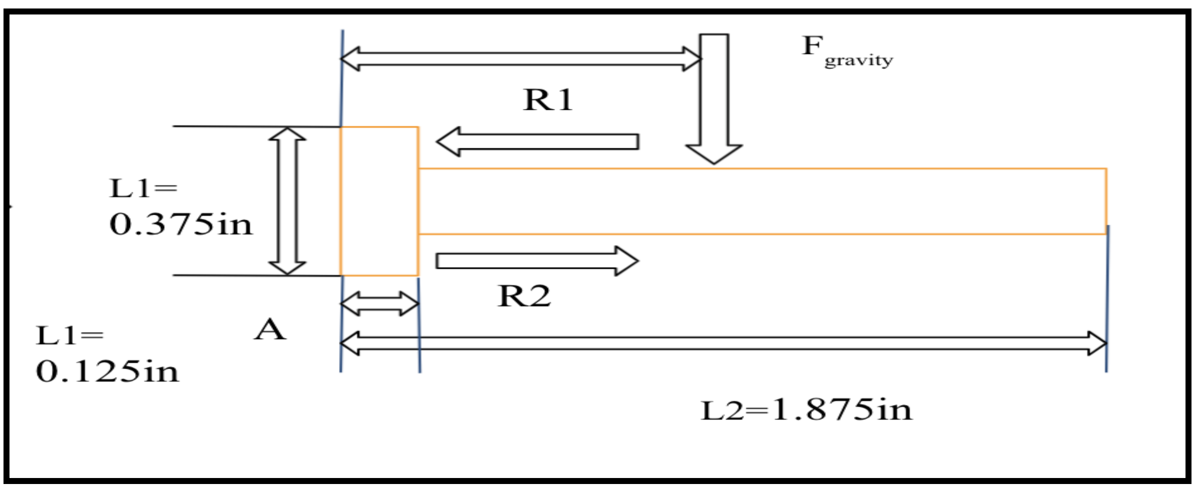
Handle Bolt:

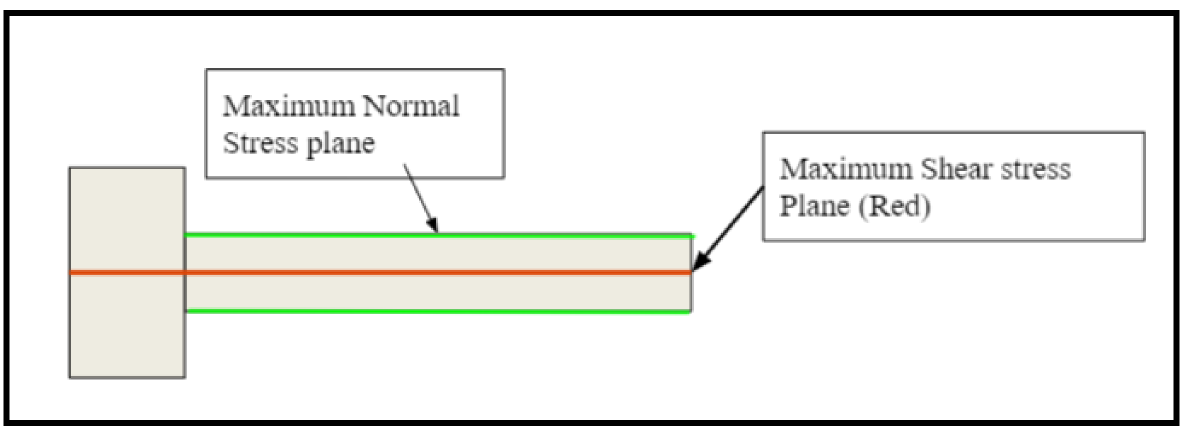
Material Selection: The material used for the handle bolt analysis is 1015 cold drawn steel. This type of low carbon steel was chosen because it is common in bolts of this type and has strong mechanical properties such as a yield strength of 47 kpsi and an ultimate tensile strength of 56 kpsi.

Assumptions:-

- The system is frictionless.
- System at equilibrium.
- The Fg (Gravity Force) is acting at the middle of part because it's symmetric
- Treating handle bolt like a bending beam
- Two reaction forces.

FBDS:





Handle Bolt Calculations:

Initial Givens:

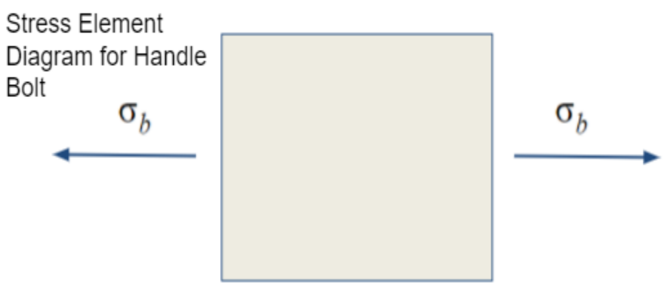
F = 24lbf, L = 1.875 in, R1 and R2 = 24lbf, $\Sigma M_A = 0,\ M_A = L * F = 45lbf - in$

$A = .125in * 1.875in = 0.234in^2$

Stress Calculations:

I = $\frac{1}{12}bh^3 = 0.00088in^4$ **,** h = 0.375in, b = 0.2in, c = 0.1875in

$\sigma_{bmax} = \frac{Mc}{I} = \frac{45lbf-in*0.1875in}{0.00088in^4} = 9588psi = 9.588kpsi$



Static Safety Factor:

Sy = 47 kpsi, Sut = 56 kpsi

$\sigma_{bmax} = \frac{Mc}{I} = \frac{45lbf-in*0.1875in}{0.00088in^4} = 9588psi = 9.588kpsi$

$\eta_{MSS} = \frac{S_y}{\sigma_{mp}-\sigma_{mn}} = \frac{47\ kpsi}{9.588\ kpsi} = 4.9$

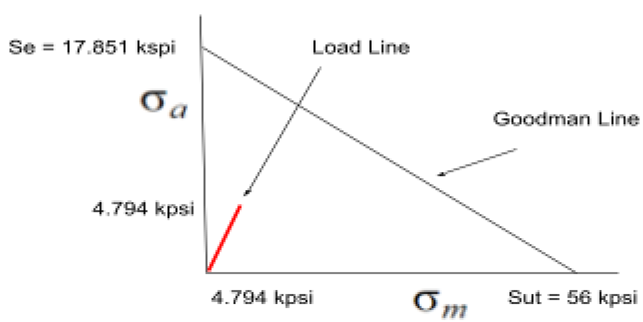
Fatigue Safety Factor:

$\sigma_a = 4.794\ kpsi,\ \sigma_m = 4.794\ kpsi, Se = Se'K_aK_bK_cK_e = 17.851\ kpsi$

$Se' = .5 * Sut = 28\ kpsi,\ K_e = 0.868\ at\ 95\%, K_a = aSut^b = 2 * Sut^{-0.217}(Cold\ Drawn) = 0.758$

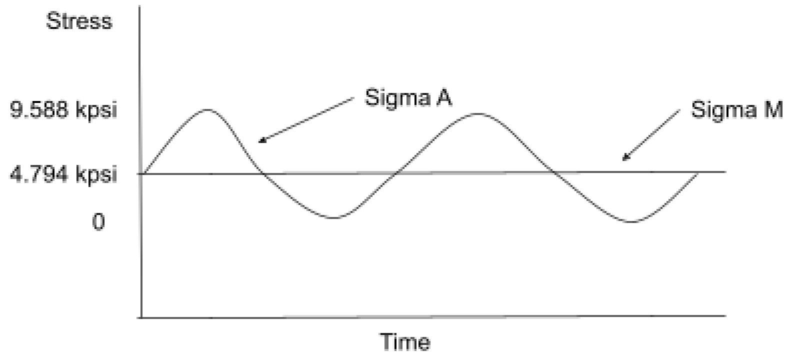
$K_b = .879d^{(-.107)} = 0.969,\ K_c = 1(bending),\ \ n_f = (\frac{\sigma_a}{Se} + \frac{\sigma_m}{Sut})^{-1} = 2.824$

Sigma A-Sigma M plot:



Stress Over Time Plot:

- Alternating and mean stress in the handle bolt over time

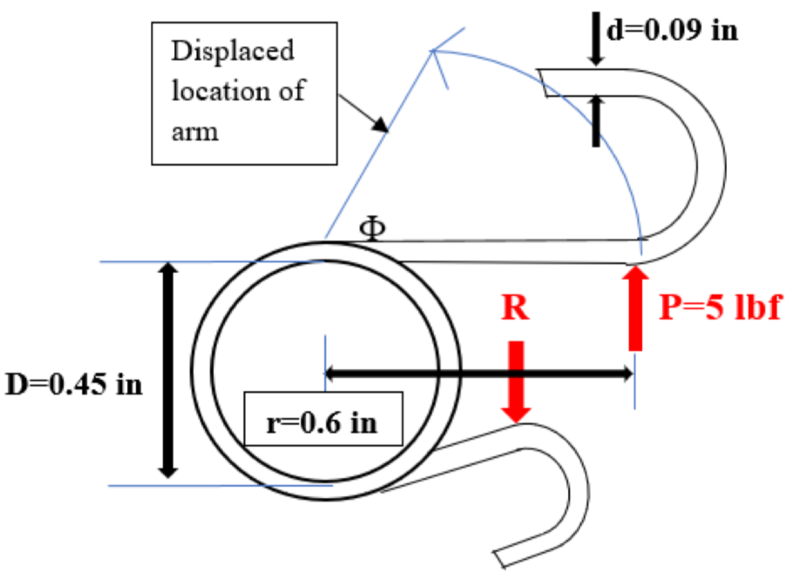


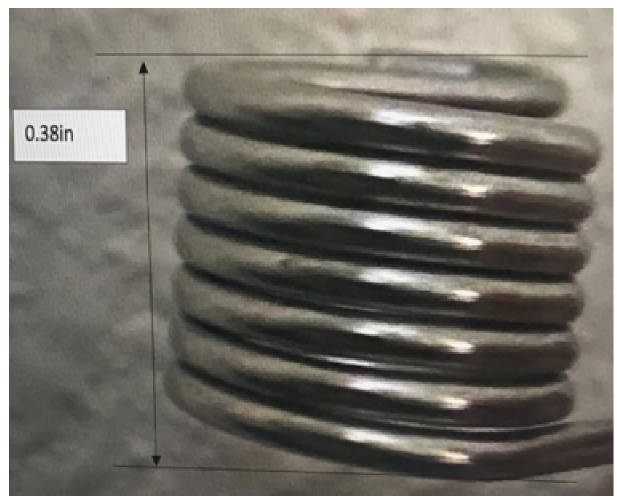
Torsion/Tension Spring

Function: Peeling blade stays intact with the apple's surface due to torsion spring.Tension springs are used to attach two components and bring them together when they try to detach. The spring stretches to a specific length as the load/force is applied, absorbing energy to create a resistance to the pulling force.Tension spring creates tension in the peeling arm as it is mounted on the main peeler body, whereas one of its arms is attached with a peeling arm.

Material choice: The selected material for analysis is 301 Stainless Steel. 301 stainless steel was used because it is rust resistant, durable and robust, has a high ductility and excellent fatigue life, high tensile strength and elastic limit, and overall inexpensive.

FBD is shown in figure below:





Assumptions:

1. Dynamic loading is taken as static for analysis
2. Negligible length of one arm
3. On average 5lbf of force is acting on spring arm due to apple
4. One arm is accounted for by the force exerted by the apple and the other arm is subjected

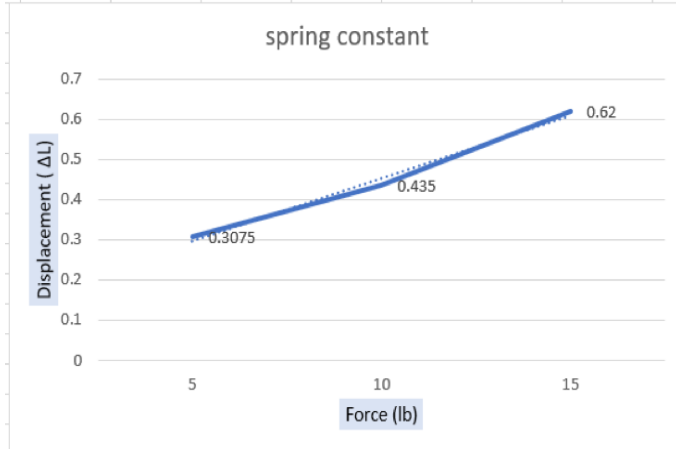
- **Calculations for Stress & Fatigue failure analysis**

1. $Active\ spring\ length = L = \pi(0.5)(7) = 11in$
2. $Torsion\ Moment = M = P \times r = 5 \times 0.6 = 3\ lbf.in$
3. $Torsion\ angle = \emptyset = \frac{64(3)(0.745)(7)}{(29007547)(0.09)^4} = 0.53°$
4. $Spring\ rate = K_T = \frac{(29007547)(0.09)^4}{64(0.745)(7)} = 5.702$

- **Known or measured data**

1. Spring wire diameter, d=0.09 in
2. Pin diameter, D_{pin}=0.45 in
3. Coil outer diameter, D_{out}=0.745 in
4. Spring arm length, r=0.6 in
5. Starting Force, P=5 lbf
6. Active number of coils, N=7
7. Material = Stainless Steel
8. Modulus of Elasticity, E=29007547.5psi

Test no.	Force (lb)	Initial length (Li)	Final length (Lf)	Change in length, (in) ΔL= (Lf-Li)
1	5	0.38	0.6875	0.3075
2	10	0.38	0.815	0.435
3	15	0.38	1	0.62
Slope = (y2-y1)/(x2-x1)				
Slope:	0.03125			

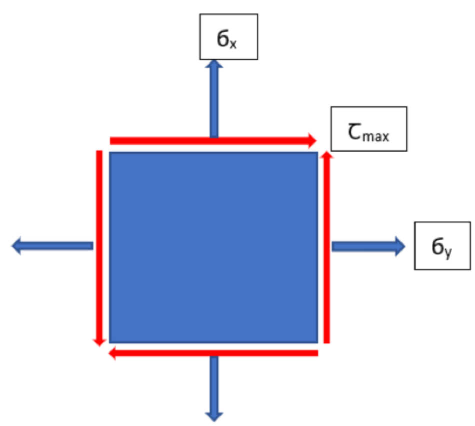


In order to calculate the spring constant, k, we considered three forces; that are 5, 10 and 15 lbf and on applying each force we measured the respective displacements in spring. After that we constructed a graph between force and displacement, from where we determined the slope which is actually the spring constant; 0.03125.

Stress Analysis

First of all the ultimate tensile stress S_{ut} is calculated using ‘A’ and ‘m’ from table 10.4 for stainless steel wire of spring. Then yield strength ‘S_y’ is calculated from equation 10.57. Bending correction factor ‘K_i’ from equation 10.43. After this maximum torque is determined by rearranging the stress formula from equation 10.44, here stress will be considered to be yielding strength ‘S_y’. Now considering no factor of safety, we calculated the number of turns of the coil ‘Φ_c’, using equation 10.54 and table 10.5.For stainless steel wire, from Table 10–4 we find that A = 169 kpsi · in^m and m = 0.146. Therefore,

1. *Ultimate tensile strength,* $S_{ut} = 253.3 kpsi$
2. *Yield Strength,* $S_y = 0.78 S_{ut} = 197.6\ kpsi$
3. *Mean coil diameter* $= D = D_{out} - d = 0.655\ in$
4. *Spring index* $= C = D/d = 8.277$
5. *Bending stress-correction factor,* $K_i = 1.0989$
6. *Active number of turns, Na = 7.22 turns*
7. *Maximum Bending Stress,* $6\sigma = K_i \frac{32Fr}{\pi d^3} \Rightarrow 6\sigma(1.0989)\frac{32\times5\times0.6}{\pi\times0.09^3}$ *46.1kpsi*

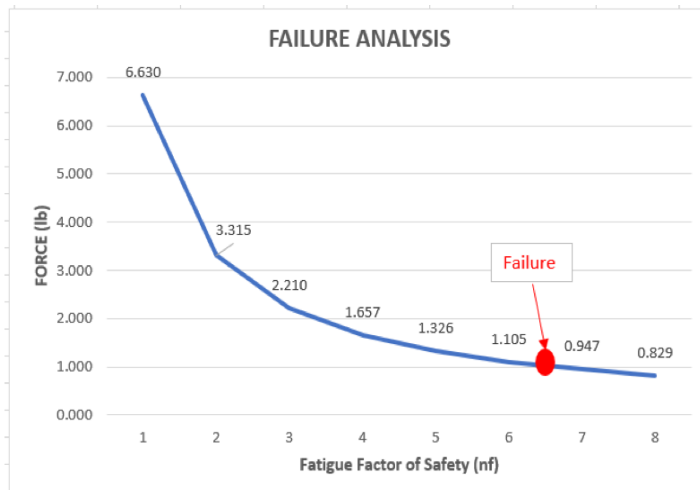


Fatigue Failure Analysis: For fatigue failure analysis we will subject the spring under different loadings; starting from 11 lbf. For each loading we will be calculating the fatigue factor of safety so that when the value of fatigue FOS becomes lower than one it means that on respective force the spring would probably fail.

1. $S_r = 126.65\ Kpsi$
2. *Bending endurance limit,* $Se = 67.546\ kpsi$
3. *The amplitude component of the strength,* $S_a = 60.44\ kpsi$
4. *Fatigue factor of safety,* $\eta_f = \frac{60.44}{46.1} = 1.31$

Now we will check fatigue factor of safety under different loadings to determine where the system will fail.

force(lb)	stress(psi)	Sa(psi)	nf
1	9116.616	60440	6.630
2	18233.23	60440	3.315
3	27349.85	60440	2.210
4	36466.46	60440	1.657
5	45583.08	60440	1.326
6	54699.69	60440	1.105
7	63816.31	60440	0.947
8	72932.92	60440	0.829



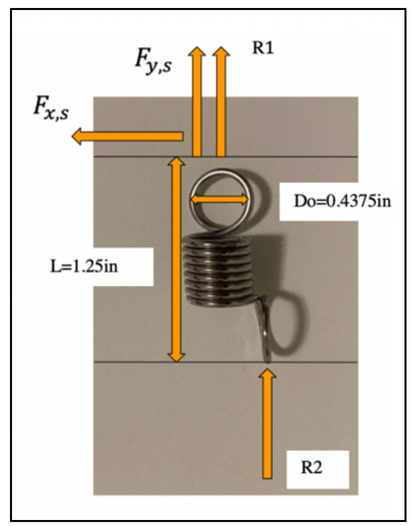
By compiling the data in excel and applying the formulas we came to know that the spring would fail around 7lb of force as the fatigue factor of safety falls under ‘1’. As shown in the graph.

Shaft Lever Spring

Material Selection:- For the shaft lever spring, we chose the music wire as our material, and it is a high-carbon steel alloy that's commonly used because of its high tensile strength. Also, due to high tensile strength, it has a better fatigue life than other materials. Having a high tensile strength, one of the most important reasons to select the music wire as our materials is due to the tension that the shaft lever spring is already experiencing in the apple peeler. The music wire is the toughest and it's used for small springs because it can withstand higher stresses under repeated loading and handle severe bends.

Assumptions:

- The spring is massless
- The system is at equilibrium
- Maximum Tensile Force is *7lb.*
- Initial Tensile Force is 2.5lb
- The spring is unpeened



FBD And Force Analysis:

The important assumption we made here is that the spring has an initial force equal to 2.5lb because the spring is already experiencing tension in the apple peeler.

Locations of Highest Stresses

In Figure-1, the stress at A is maximum due to axial and bending loading.

In Figure- 2, there is another potential stress which is torsional stress at B, so it must be included.

Figure1

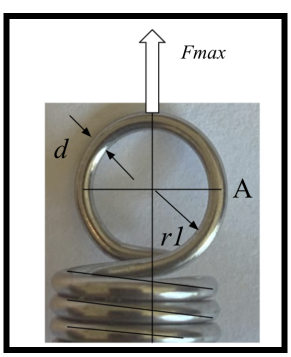
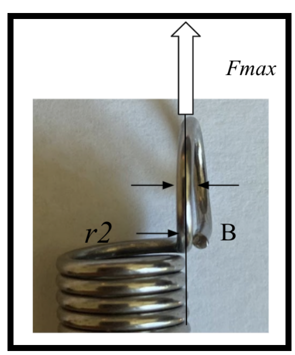


Figure2



Calculating the maximum tensile stress at A and maximum torsional stress at point B

For point A:

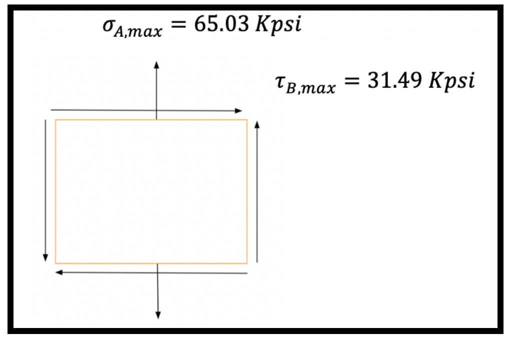
$$(K)_A = \frac{4C_1^2 - C_1 - 1}{4C_1(C_1-1)} = \frac{4(6)^2-(6)-1}{4(6)(6-1)} = 1.14 \text{ , } C_1 = \frac{2r_1}{d} = \frac{2(0.1875)}{0.0625} = 6 \text{ (Bending correction factor)}$$

$F_{max} = 7lb$
D=0.375in
d=0.0625in
$r_1 = 0.1875in$
$r_2 = 0.1875in$

$$\sigma_{A,max} = F_{max}\left[(K)_A \frac{16D}{\pi d^3} + \frac{4}{\pi d^2}\right]$$

$$= 7\left[(1.14)\frac{16(0.375)}{\pi(0.0625)^3} + \frac{4}{\pi(0.0625)^2}\right](10^{-3}) = 65.03\ Kpsi$$

For Point B:



$$(K)_B = \frac{4C_2 - 1}{4C_2 - 4} = \frac{4(6)-1}{4(6)-4} = 1.15 \text{ (stress} - \text{correction factor)}$$

$$C_2 = \frac{2r_2}{d} = \frac{2(0.1875)}{0.0625} = 6$$

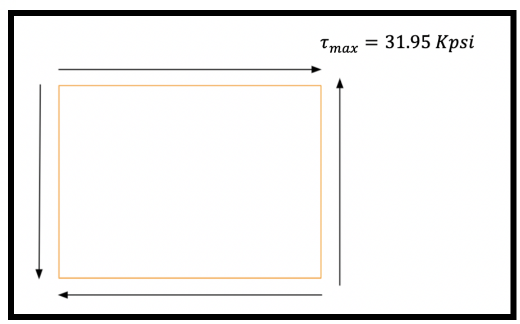
$$\tau_{B,max} = (K)_B \frac{8F_{max}D}{\pi d^3} = \left[(1.15)\frac{8(7)(0.375)}{\pi(0.0625)^3}\right](10^{-3}) = 31.49\ kpsi$$

Next, we started to analyze the helical spring by taking a section cut. As a result, we found out the direction of the direct shear force, Then, calculating the maximum tensile stress in the wire, and we used the stress correction factor

$$\tau_{max} = K_s \frac{8F_{max}D}{\pi d^3} = \left[(1.167)\frac{8(7)(0.375)}{\pi(0.0625)^3}\right](10^{-3}) = 31.95\ Kpsi$$

$$C = \frac{D}{d} = \frac{0.375}{0.0625} = 6$$

$$K_s = \frac{2C+1}{2C} = \frac{2(6)+1}{2(6)} = 1.167$$



Factor of Safety

For The Hook:
To find the factor of safety at A, we used maximum tensile stress and minimum tensile strength using the values provided in Table-1 and torsional yield strength.

$$S_{ut} = \frac{A}{d^m} = \frac{201}{0.0625^{0.145}} = 300.46\ Kpsi$$

$$S_y = 0.75S_{ut} = 0.75(300.46) = 225.34\ kpsi$$

Table 1

Material	ASTM NO.	Exponent (m)	A (*kpsi-in*)
Music Wire	*A228*	*0.145*	*201*

$$(n_y)_A = \frac{S_y}{\sigma_{A,max}} = \frac{225.345}{65} = 3.4$$

To find the factor of safety at B, using the maximum torsional stress and the yield strength

$$S_{sy} = 0.45S_{ut} = 0.45(300.46) = 135.207\ kpsi$$

$$(n_y)_B = \frac{(S_{sy})_B}{\tau_{B,max}} = \frac{135.207}{31.49} = 4.29$$

For The Body Coil:
Here we determined the Factor of Safety For The Body-Coil. First we used the alternating, mean forces, and initial force.

$$F_m = \frac{F_{max} + F_{min}}{2} = \frac{7+4}{2} = 6lb \qquad F_a = \frac{F_{max} - F_{min}}{2} = \frac{7-4}{2} = 1.5lb$$

Next, we calculate the alternating, mean and initial torsional shear stresses. Also, we used the load line for the body coil.

$$(K)_s = 1.167$$

$$\tau_{max} = (K)_s \frac{8F_{max}D}{\pi d^3} = 31.95\ kpsi$$

$$\tau_a = (K)_s \frac{8F_aD}{\pi d^3} = 6.84\ kpsi$$

$$\tau_m = (K)_s \frac{8F_mD}{\pi d^3} = 27.38\ kpsi$$

$$\tau_{min} = (K)_s \frac{8F_{min}D}{\pi d^3} = 18.25\ kpsi$$

$$\tau_i = \left(\frac{F_i}{F_a}\right)\tau_a = \left(\frac{1.5}{2.5}\right)(6.84) = 4.104\ kpsi$$

$$(S_{sa})_y = \left[\frac{r}{(r+1)}\right](S_{sy} - \tau_i) = \left(\frac{0.292}{0.292+1}\right)(135.207 - 4.104) = 29.63kpsi$$

$$r = \frac{\tau_a}{(\tau_m - \tau_i)} = \frac{6.7}{(26.9 - 4.02)} = 0.292$$

$$(n_y)_{body} = \frac{(S_{sa})_y}{\tau_a} = \frac{29.63}{6.84} = 4.33$$

<u>Fatigue Factor</u>

For The Hook (Tension at A)

$$\sigma_{A,max} = 65\ kpsi$$

$$\sigma_{A,a} = F_a\left[(K)_A \frac{16D}{\pi d^3} + \frac{4}{\pi d^2}\right] = 1.5\left[(1.14)\frac{16(0.375)}{\pi(0.0625)^3} + \frac{4}{\pi(0.0625)^2}\right](10^{-3}) = 13.865\ kpsi$$

$$\sigma_{A,m} = F_m\left[(K)_A \frac{16D}{\pi d^3} + \frac{4}{\pi d^2}\right] = 6\left[(1.14)\frac{16(0.375)}{\pi(0.0625)^3} + \frac{4}{\pi(0.0625)^2}\right](10^{-3}) = 55.46\ kpsi$$

$$\sigma_{A,min} = F_{min}\left[(K)_A \frac{16D}{\pi d^3} + \frac{4}{\pi d^2}\right] = 4\left[(1.14)\frac{16(0.375)}{\pi(0.0625)^3} + \frac{4}{\pi(0.0625)^2}\right](10^{-3}) = 35.68\ kpsi$$

$S_{su} = 0.67 S_{ut} = 0.67(300.46) = 201.3\ Kpsi$

Here, we assumed it is unpeend because we are going to calculate Zimmerlia Data. the using the values of unpeened that is provided in Table-2

$$S_{se} = \frac{S_{sa}}{1-\left(\frac{S_{sm}}{S_{su}}\right)^2} = \frac{35}{1-\left(\frac{55}{201.3}\right)^2} = 37.82\ kpsi$$

Table-2

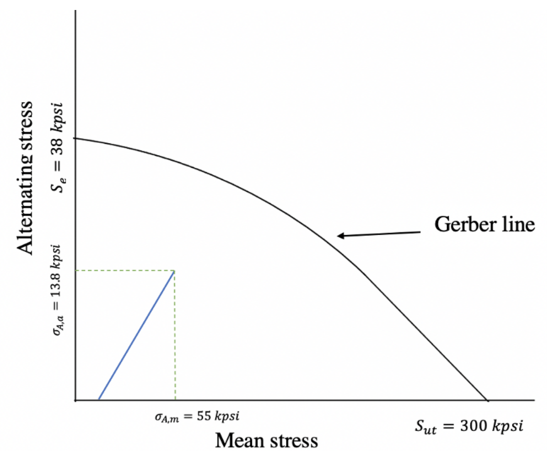
Unpeened
$S_{sa} = 35\ kpsi$
$S_{sm} = 55\ kpsi$

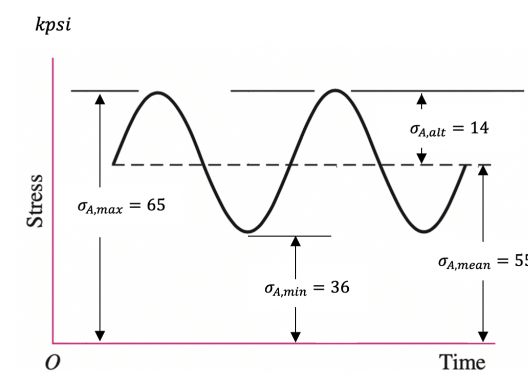
And to find the tensile endurance limit, we used the distortion-energy theory, and Gerber equation.

$$S_e = \frac{S_{se}}{0.577} = \frac{37.82}{0.577} = 65.55\ kpsi$$

$$(n_f)_A = \frac{1}{2}\left(\frac{S_{ut}}{\sigma_{A,m}}\right)^2 \frac{\sigma_{A,a}}{S_e}\left[-1+\sqrt{1+\left(2\frac{\sigma_{A,m}}{S_{ut}}\frac{S_e}{\sigma_{A,a}}\right)^2}\right] = \frac{1}{2}\left(\frac{300.46}{55.46}\right)^2 \frac{13.86}{65.55}\left[-1+\sqrt{1+\left(2\frac{55.46}{300.46}\frac{65.55}{13.86}\right)^2}\right] = 3.13$$

Here the stress over time and Gerber line for tension at A





For the Hook (Torsion at B)

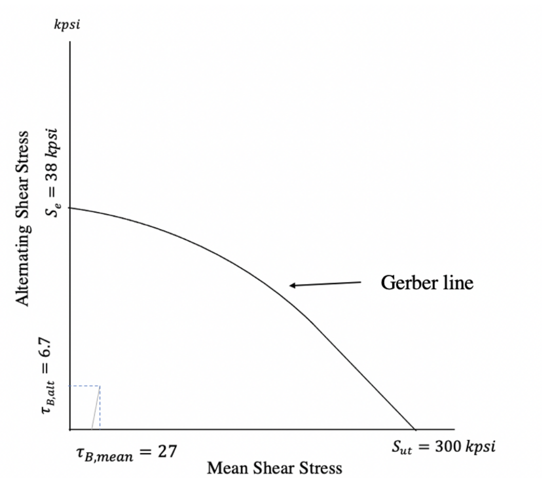
$\tau_{B,max} = 31.49\ Kpsi$, $\tau_{B,a} = 6.7\ kpsi$, $\tau_{B,m} = 26.9\ kpsi$, $\tau_{B,min} = 17.99\ kpsi$

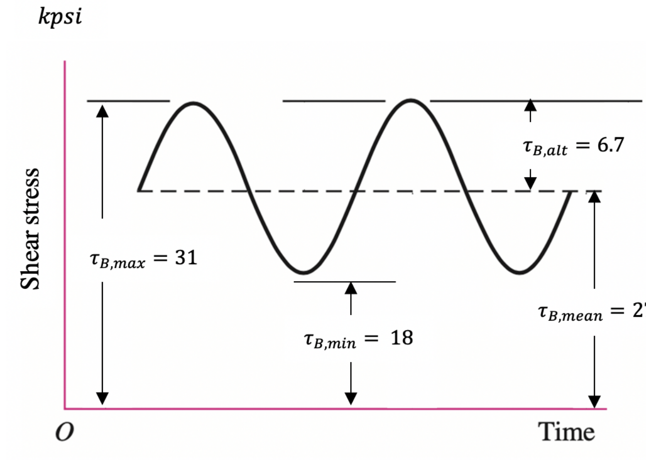
$S_{se} = 37.82\ kpsi$

$S_{su} = 201.308\ kpsi$

$$(n_f)_B = \frac{1}{2}\left(\frac{S_{su}}{\tau_{B,m}}\right)^2 \frac{\tau_{B,a}}{S_{se}}\left[-1+\sqrt{1+\left(2\frac{\tau_{B,m}}{S_{su}}\frac{S_{se}}{\tau_{B,a}}\right)^2}\right] = \frac{1}{2}\left(\frac{201.308}{26.9}\right)^2 \frac{6.7}{37.82}\left[-1+\sqrt{1+\left(2\frac{26.9}{201.308}\frac{37.82}{6.7}\right)^2}\right] = 4.0$$

Here the stress over time and Garber line for torsion at B





For Body Coil

$(K)_s = 1.167$

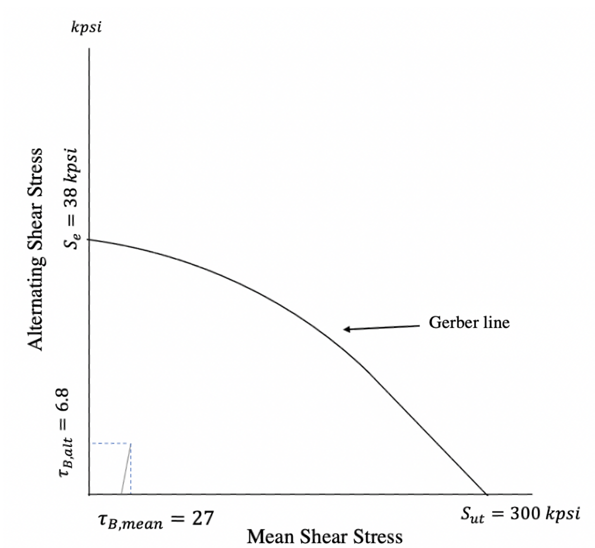
$\tau_{max} = 31.95\ kpsi$, $\tau_a = 6.84\ kpsi$, $\tau_m = 27.38\ kpsi$, $\tau_{min} = 18.25\ kpsi$

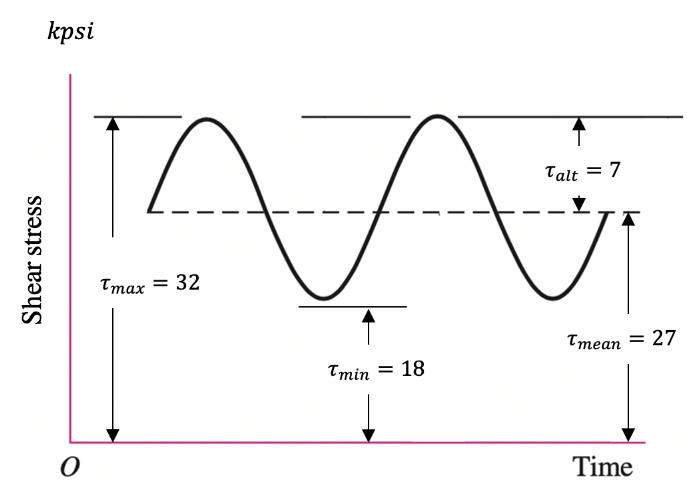
$S_{se} = 37.82\ kpsi$

$S_{su} = 201.308\ kpsi$

$$(n_f)_{Body} = \frac{1}{2}\left(\frac{S_{su}}{\tau_m}\right)^2 \frac{\tau_a}{S_{se}}\left[-1+\sqrt{1+\left(2\frac{\tau_m}{S_{su}}\frac{S_{se}}{\tau_a}\right)^2}\right] = \frac{1}{2}\left(\frac{201.3}{27.38}\right)^2 \frac{6.84}{37.82}\left[-1+\sqrt{\left(2\frac{27.38}{201.3}\frac{37.82}{6.84}\right)^2}\right] = 3.93$$

Here the stress over time and Gerber line for body coil



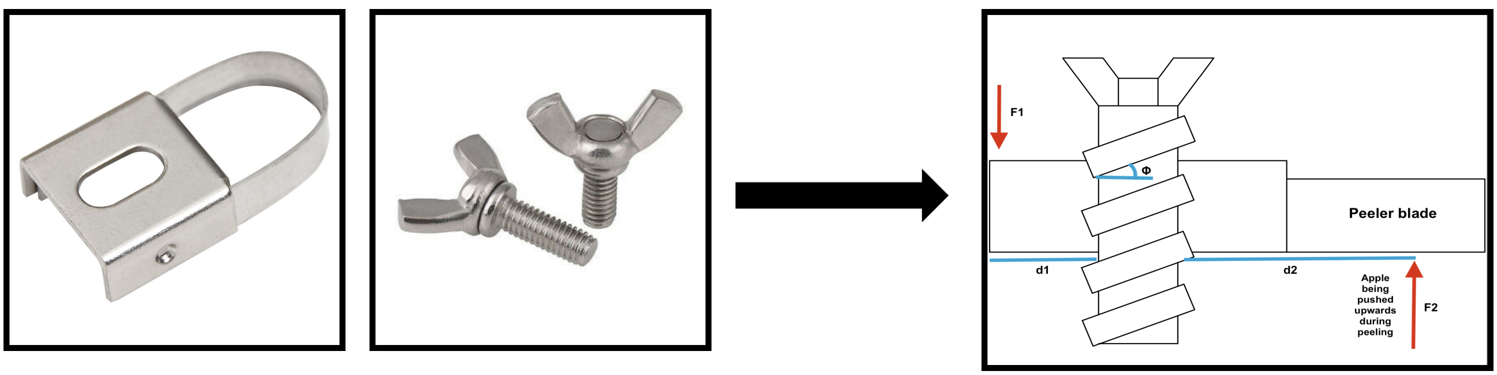


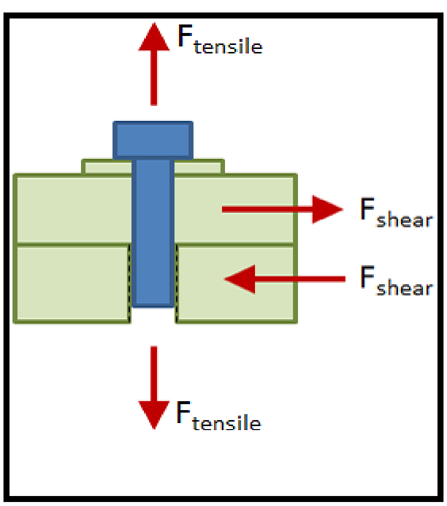
Peeling Blade Screw

Material Selection: For this analysis of the peeling blade screw, we decided on using 316 Stainless Steel. Other than it being commonly used, it costs roughly \$0.78/lb and has a minimum density of 7.87 lb/ft^3, with high tensile strength of 579 MPa. The material also has a better corrosion resistance compared to similar alloy such as 304 and 314 stainless steel. Furthermore, 316 stainless steel has a melting range between 2500ºF to 2550ºF, with a maximum use temperature of around 1472ºF. Due to its higher level of corrosion resistance, especially in salt water, 316 stainless steel is unsurprisingly used most frequently in wet surroundings. Since it has frequent contact with an apple, where the peeling blade is located at, the material has the potential to last longer compared to a lower corrosion resistance material.

Assumptions:

1) The screw is in a perpendicular (90º) position to the peeler
2) Friction is negligible
3) Mass of screw is negligible
4) During analysis the blade and screw is in equilibrium





To calculate the max force, θ=45º
P = M/R = F tan (Φ + θ), where Φ = arctan μ
100/(10*10^-3)= F tan (8.53+45)
F = 7.4lbf (max)

Given that

F = 7.4 lbf, L = 0.629 in (16mm), A = 1.2in, b = 0.2 in, c = 0.1875 in,

h = 0.375 in

M = FL = 4.65 lbf.in

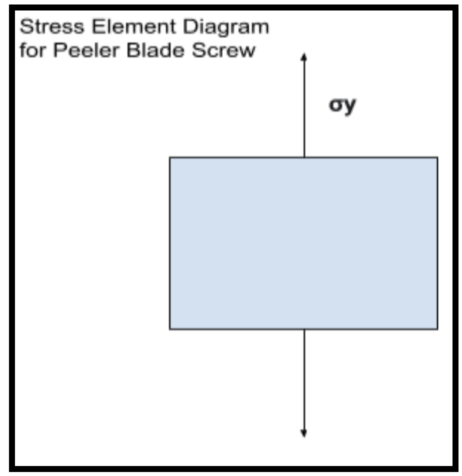
R1 = R2 = 7.4 lbf

V = M/L = 4.65/0.629 = 7.39 lbf

I = 1/12(b)(h^3) = 1/12 (0.2) (0.375)^3 = 8.8*10^-4

$\delta_2 = F/A = 6.17\ psi$

$\delta_{max} = MC/I = 991\ psi$



The von Mises stress is calculated by

$\sigma' = [(\sigma_1 + \sigma_2)^2 + 3\tau_{xy}]^{1/2}$

$\sigma' = 423.5$ psi

Yield factor of safety

For this, we use the maximum distortion theory to get our allowable stress of 118.4 MPa.

$Stress\ \delta = 21.7\ kpsi$

Maximum distortion energy theory

$\delta_1{}^2 - \delta_1\delta_2 + \delta_2{}^2 = \delta_{allow}{}^2$

$21.7^2 - (21.7)(-106) + (-106)^2 = \delta_{allow}{}^2$

Yield safety factor

$$n = \frac{ultimate\ stress}{allowable\ stress}$$

$$= \frac{144}{118.4}$$

$$= 1.21$$

Fatigue factor of safety

$$Given\ \delta_{max} = 21.7kpsi,\ \delta_{min} = -8.6\ kpsi,$$

$$S_{ut} = 42\ kpsi,\ S_y = 24.4\ kpsi,\ S_e = 18\ kpsi,$$

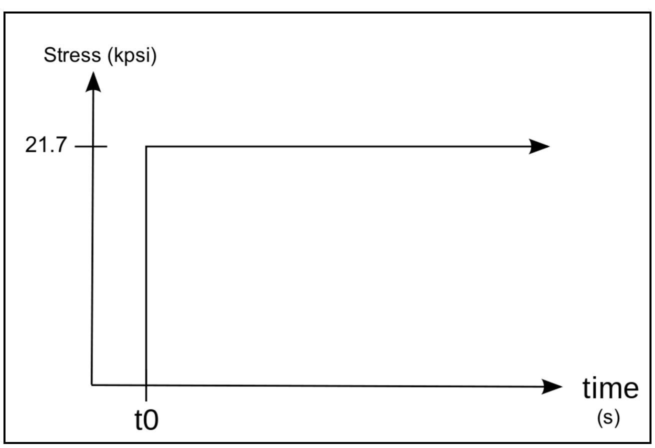
$$\delta_a = \frac{21.7 - (-8.6)}{2} = 15.15\ kpsi$$

$$\delta_m = \frac{21.7 + (-8.6)}{2} = 6.55\ kpsi$$

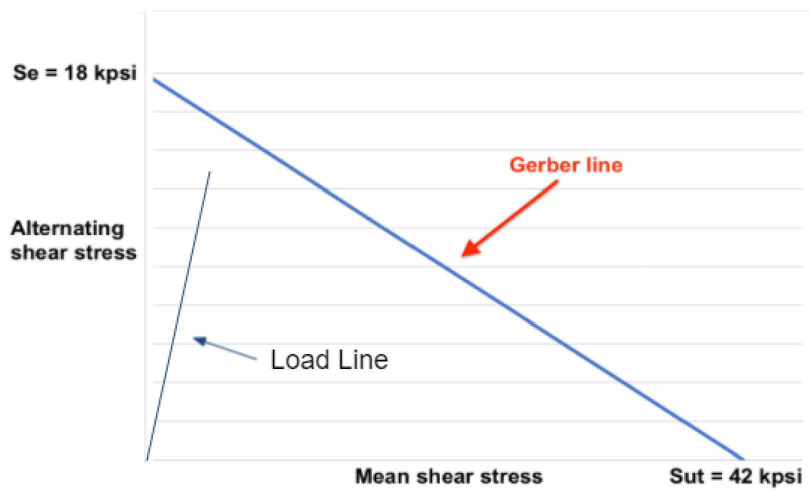
$$n_f = \frac{1}{\frac{\delta_a}{S_e} + \frac{\delta_m}{S_{ut}}}$$

$$n_f = \frac{1}{\frac{15.55}{18} + \frac{6.55}{42}} = 0.98$$

Stress over time plot



Sigma A-Sigma M plot



Peeling Shaft:

Material Selection: For the material selection of the peeling shaft, we decided to use AISI 1006 cold-drawn steel. This is because this material will give us a lot of value for its price, as it is cheap, low carbon, and has a high enough ultimate tensile strength of 41 kpsi provided for the induced stress. Also, since most steels have relatively the same modulus of elasticity, we don't need to get an expensive material since deflection will be nearly the same across all steel types. Also, since the shaft is only 0.3125 inches, it is optimal for it to be cold-drawn steel as cold-drawn is typically used for shafts with diameters under 3 inches.

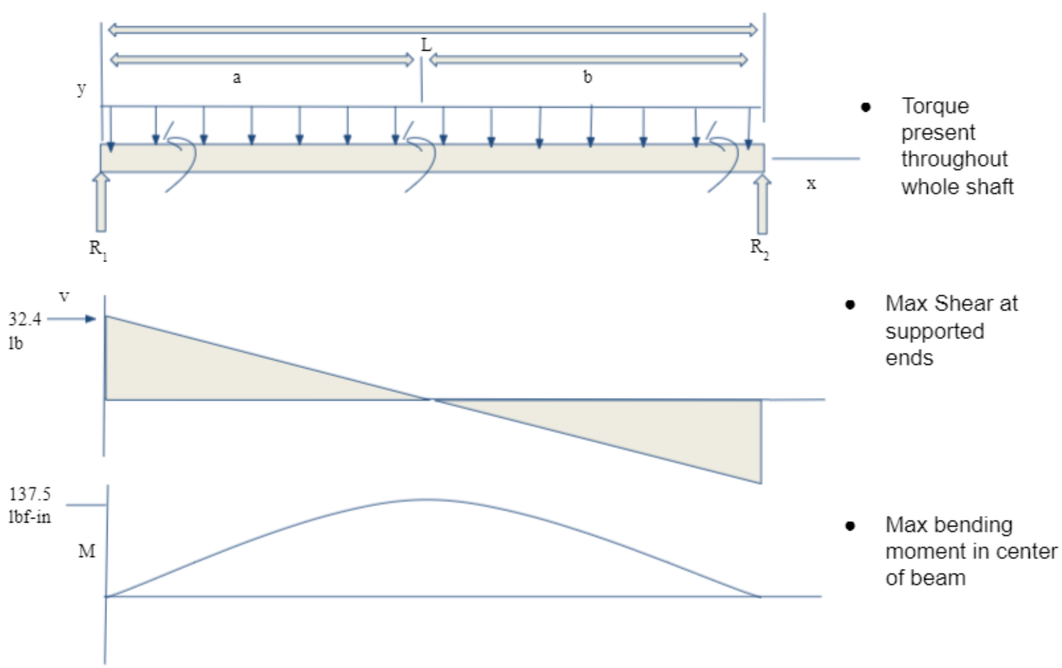
Assumptions:

- Uniform bar of steel with a circular cross section
- Analyze the stress in the shaft at critical locations, such as the surface where max stress occurs
- Shaft is transmitting torque throughout its body
- Neglecting axial forces as they are negligible.

Simplifications:

- Simply supported beam with a moment load
- Two reaction forces

Free Body Diagram Analysis:



This shows the max shear and max bending moment, and where they occur in the shaft.

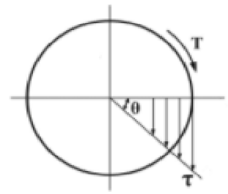
Peeling Shaft Stress Analysis:

Calculating Principal Stresses:

$\boldsymbol{\sigma}_{1,2} = (\boldsymbol{\sigma}_x + \boldsymbol{\sigma}_y)/2 \pm \sqrt{[((\boldsymbol{\sigma}_x - \boldsymbol{\sigma}_y)/2)^2 + \tau_{max}{}^2]}$

Calculating $\boldsymbol{\sigma}_x + \boldsymbol{\sigma}_y$

- Since Stress increases linearly from within the circular cross section, $\boldsymbol{\sigma}_x = \boldsymbol{\sigma}_y$



$\boldsymbol{\sigma}_{x,y} = My/I$

$M = 136.5$ lbf-in

$y = r = D/2 = 0.3125/2 = 0.15625$ in

$I = \pi D^4 / 64 = \pi(0.3125)^4 / 64 = 0.000468$

$\boldsymbol{\sigma}_{x,y} = My/I = (136.5)(0.15625) / 0.000468 = 45.6$ psi

$\tau_{max} = Tr/J$

$T_m = 0.194$ lbf-in

$r = 0.15625$ in

$J = \pi D^4 / 32 = \pi(0.3125)^4 / 32 = 0.000936$

$\tau_{max} = Tr/J = (0.194)(0.15625)/0.000936 = 32.4$ psi

Value matches up with our shear value in shaft found during load analysis

$\boldsymbol{\sigma}_{1,2} = (\boldsymbol{\sigma}_x + \boldsymbol{\sigma}_y)/2 \pm \sqrt{[((\boldsymbol{\sigma}_x - \boldsymbol{\sigma}_y)/2)^2 + \tau_{max}{}^2]} = 78, -13.2$

Principal Stresses: 78, -13.2 psi

- Rotating shaft so there is a constant bending moment and steady torsion.
  $M_m = T_a = 0$

Calculating Alternating Stress:

$\boldsymbol{\sigma}_a = [(32k_f M_a/\pi d^3)^2 + 3(16K_{fs}\cancel{T_a/\pi d^3}^{\,0})^2]^{½} = [(32k_f M_a/\pi d^3)^2]^{½}$

$k_f = 1 + q(k_t - 1) = 1.4$, $M_a = 137.5$ lbf-in , $d = 0.3125$ in

$\boldsymbol{\sigma}_a = [((32(1.4)(137.5) / \pi(0.3125)^3)^2]^{½} = 64.3$ psi

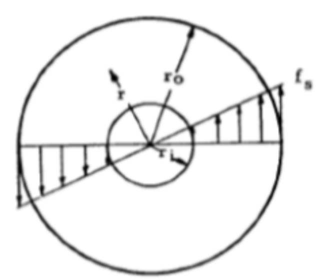


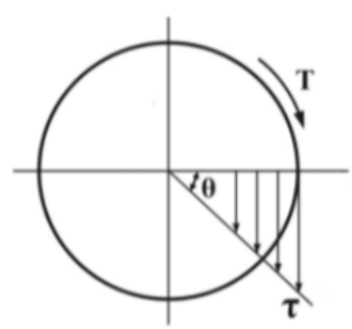
Figure 1-43. Stress Distribution of Circular Beam in Torsio

Calculating Midrange Stress:

$\boldsymbol{\sigma}_m = [(\cancel{32k_f M_m/\pi d^3})^2 + 3(16K_{fs}T_m/\pi d^3)^2]^{½} = [3(16K_{fs}T_m/\pi d^3)^2]^{½}$

$K_{fs} = 1 + q_s(K_{ts} - 1) = 1.2$, $T_m = \tau\pi d^3/16 = 0.194$ lbf-in

$\boldsymbol{\sigma}_m = [3(16(1.2)(0.194)/\pi(0.3125)^3)^2]^{½} = 67.3$ psi



Calculating Von Mises Stress:

$\boldsymbol{\sigma}_{max} = [(\boldsymbol{\sigma}_a + \boldsymbol{\sigma}_m)^2 + 3(16K_{fs}(T_m + T_a)/\pi d^3)^2]^{½} =$ **17.3 kpsi**

This max stress will occur on the surface of the shaft due to a combination of torsion and bending

Peeling Shaft Failure Analysis:

Yield Factor of Safety

$\mathbf{n_y = S_y / \boldsymbol{\sigma}_{max}}$

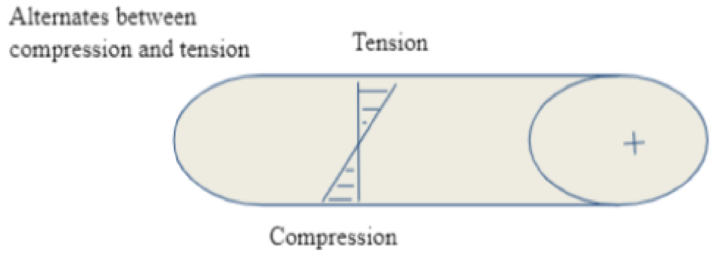
$\sigma_{max} = [(\sigma_a + \sigma_m)^2 + 3(16K_{fs}(T_m+T_a)/\pi d^3)^2]^{½}$

$\sigma_{max} = [(64.3 + 67.3)^2 + 3(16(1.2)(0.194+0)/\pi(0.3125)^3)^2]^{½} = 17.3$ kpsi

$S_y = 41$ kpsi (as given in table A-20 in textbook)

$n_y = S_y / \sigma_{max} = 41\text{kpsi} / 17.3\text{kpsi}$

$\mathbf{n_y = 2.37}$



Fatigue Factor of Safety - using Goodman

$\mathbf{n_f = (\boldsymbol{\sigma}_a/s_e + \boldsymbol{\sigma}_m/s_{ut})^{-1}}$

$S_e = S_e' k_a k_b k_c$

$k_a = aS_{ut}^{\ b}$, $a = 2.70$, $S_{ut} = 48$ kpsi, $b = -0.265$ (Given in tables)

$k_a = (2.70)(48)^{-0.265} = 0.968$

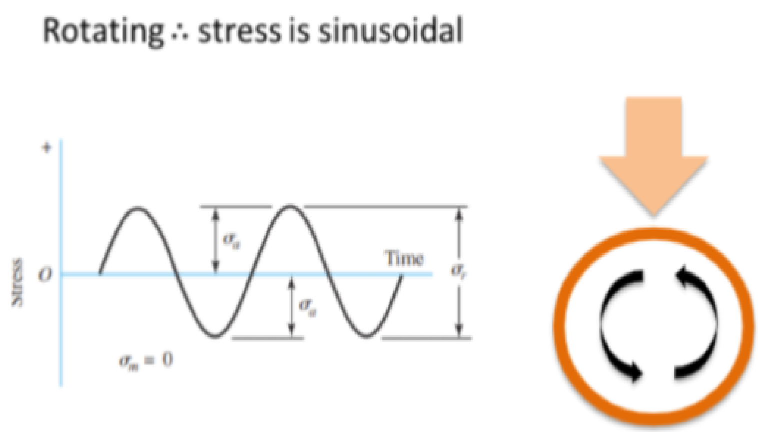
$k_b = 0.879d^{-0.107} = 0.9955$

$k_c = 1$ (bending)

$S_e = 24(0.968)(0.9955)(1) = 23.13$

$n_f = (\sigma_a/s_e + \sigma_m/s_{ut})^{-1} = (64.3/23.13 + 67.3/48)^{-1}$

$\mathbf{n_f = 2.045}$



<u>Evaluation of Design:</u> After the extensive analysis of the apple peeler and its components a full evaluation can now be complete. The main function of the apple peeler is to peel apples at an efficient speed and provide a low discomfort for the user. Using this functionally the components in the apple peeler examined are critical for providing an efficient service of peeling. The crank arm and handle bolt had good safety factors between two and three as expected because if they were to fail the entire apparatus would be compromised and broken. For these components resistance to bending and their ability to not break under harsh loading conditions is critical for the customer requirements of safe operation and low operation force. The shaft lever spring has an excellent factor of safety that ranges from 3 to 4.5, so it can maintain for a long period of time without failing due to tension, and if it fails because of tension, the spring will weaken for an extended period of time. For springs that have a good ability to last longer without weakening is essential for our customer requirements of durability. For the peeling blade screw, it has a safety factor of 1.21, which is relatively strong enough to withstand a continuous pressure exerted from the peeling blade. It is crucial for the screw to be able to last because when the peeling blade is malfunctioning, the entire apparatus will be compromised. In regards to the peeling shaft, the safety factor values found were around 2 which provide enough material strength for the stresses. As the shaft is one of the most important mechanical components in the apple peeler, it is important that the material can provide enough support to help the shaft maintain a long life cycle. This is especially important when looking at surface concentrations as most shafts tend to develop cracks before breaking in fatigue. For the tension spring the safety factor makes sense of 1.31 as we would expect the spring to be between the range of 0.8 and 1.5 in this case. The spring is critical for the operation of the apple peeler and therefore needs to be reasonably high so that the safety of the apple peeler is intact from the FMEA and that the apple peeler is easy to use for the user at a relatively low operation force.

<u>Recommendations and Alternate Design for Components:</u>

Crank Arm: With the Safety factors above one in both cases we can reduce the area of the crank arm and save money on the material cost of manufacturing the component and still have confidence the crank arm will not fail. Another recommendation is to make the crank arm out of a cheaper stainless steel such as 304. The crank arm could also be elongated to a greater length to improve the moment and force generated on the saft for more efficient peeling of the apples. The crank arm in both cases will not fail under the given loading conditions as seen in the analysis. The analysis showed that 303 stainless steel is justified numerically because the crank arm does not fail based on the mechanical properties of 303 stainless steel. The safety factors found are reasonable because they are between two and three at a comfortable margin as predicted in each case.

Handle Bolt: With the Safety factors above one we can reduce the area of the crank arm and save money on the material cost of manufacturing the component and still have confidence the handle bolt will not fail. Another recommendation is to make the crank arm out of a cheaper steel such as 1018 cold drawn steel. The handle bolt will not fail under the given loading conditions as seen in the analysis. The analysis showed that 1015 cold drawn steel is justified numerically because the handle bolt does not fail based on the mechanical properties of 1015 cold drawn steel. The safety factors found are reasonable because they are between 0.5 and 5 as predicted.

Torsion/ Tension Spring: Our recommendation on making the wire diameter larger, the spring will become stronger and if it is smaller, it will be weaker. This is because, by making the wire diameter larger, you are also making the spring's coils tighter which reduces the spring index. The opposite effect is done when you make it smaller, the spring index increases so it isn't as tight and it is under less stress. This adjustment will not only affect your spring's force but it will also affect its elastic limit like it does when adjusting the outer or inner diameter. Make the outer diameter smaller to make it stronger but also add a coil or two to release stress. You may also make the wire diameter larger to a point where it is stronger than you need but then make the outer diameter a bit larger to release tension and also achieve more travel.

Shaft lever spring: We recommend reducing the wire diameter since it can handle high tension. We also recommend changing the material since the factor of safety and fatigue factor is high enough to consider it'll last longer, and the material we recommend to use is Stainless steel 302 because it can last longer than wire music due to its corrosion resistance which is an important property that any apple peeler must have since it'll be exposed to acidic liquid and different substances. Also, if we compare the prices of music wire and stain steel, stain steel is cheaper and commonly used as well.

Peeling Blade Screw: With a reasonable safety factor, the screw is unlikely to fail with continuous use. We can switch to 316 stainless steel from its current 314 stainless steel with only a slight increase in price, but better corrosion resistance. Another recommendation is that to increase the contact surface, this will ensure a more stable and sturdy condition for the peeling blade. Another option is that we can permanently attach the peeling blade to the apple peeler main body to ensure a failure-free component. This however limits the depth of apple skin during peeling.

Peeling Shaft: For our peeling shaft, the values found in the analysis seem reasonable, given our loading scenario and how much force is being applied. In regards to design improvements for the peeling shaft, there are a number of things we can change to reduce costs and maximize value out of our apple peeler. For one, we can reduce the diameter of the shaft to reduce material costs as much as the factor of safety will allow. Another possibility is finding a new material that is cheaper and will still allow for us to achieve a factor of safety value over 1, since our material is stronger than required. Lastly, we can possibly look at adding surface treatments to the shaft in order to increase the life cycles.