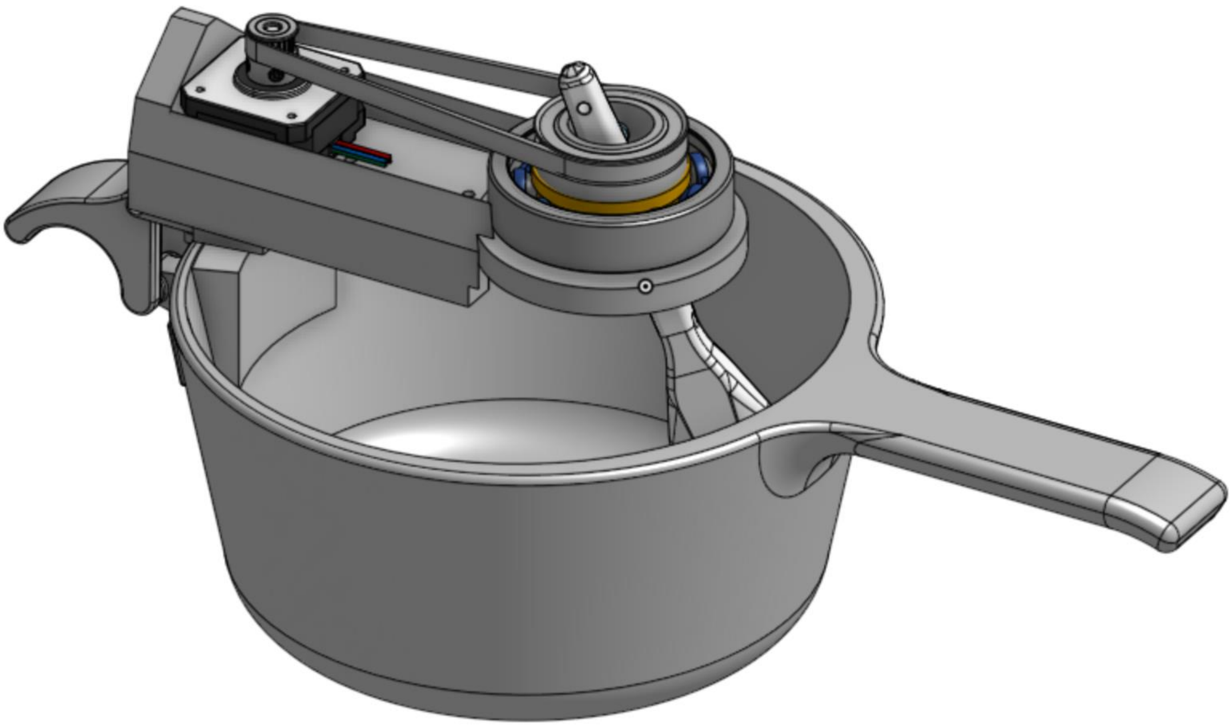
W.H.I.S.K. (Wireless Hands-free Intelligent Stirring Kitchen-aid) – Final Report



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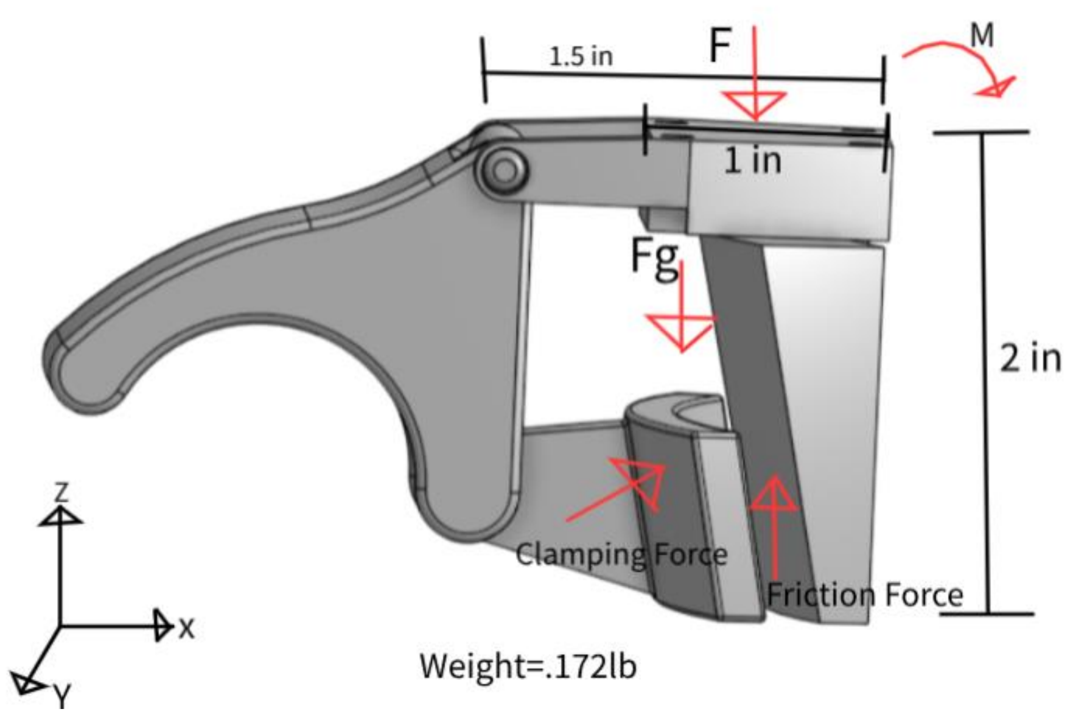
Introduction

In modern kitchens, chefs and home cooks face difficulty in multitasking and maintaining consistent stirring of pots and pans while preparing meals. This challenge often leads to uneven cooking, potential burning of ingredients, and the need for constant attention during the cooking process. Moreover, recipes requiring prolonged simmering or frequent stirring demand significant time and effort, limiting convenience and productivity in the kitchen. As a result, there is a growing demand for an automatic rotary utensil holder that can continuously stir dishes and attach different utensils for any meal. This will ensure even cooking, prevent sticking and burning, and allow cooks to focus on other aspects of meal preparation. Therefore, this device will not only save time and effort but also enhance cooking precision, leading to consistently delicious and perfectly cooked meals.

1. Component Analysis

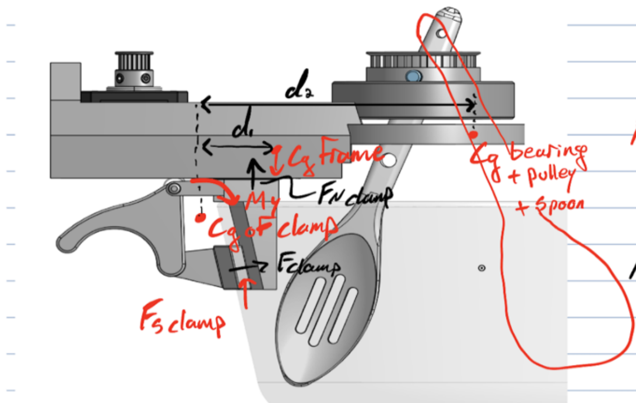
All parts used in the W.H.I.S.K. must adhere to strict FDA compliance. With proposed design requirements of high temperature use, lightweight, cleanable, and high viscous use, all parts have limited material selection ability. To reduce the cost of manufacturing, all parts will be made from Polyphenylenesulphide (PPS). PPS is a food-safe material that is approved by the FDA. It is a semicrystalline, high temperature thermoplastic and has high mechanical strength even at temperatures of 400°F. It also has low water absorption and low tendency to creep. PPS allows for the W.H.I.S.K. to operate in all cooking applications and temperatures while maintaining reliability. The material choice was validated through the following analysis.

1.1 Clamping handle



FBD:
Justification for analysis:

The clamp is important to analyze as it is one of the main load bearing parts of the device. The clamp is responsible for holding the weight of the stirring mechanism in place while it is stirring. If the clamp fails at any point during the cooking process the entire device will fail with it. The main concern for failure of the clamp is the bending moment in the Y direction that is applied by the weight of the stirring mechanism. This bending moment creates a need for the clamp to apply enough force to the pot to stay in place throughout the entire cooking process. That is why the clamping handle is essential to analyze.



$F_{s\,clamp} + F_{N\,clamp\text{-}z} = F_{rame} + F_{spoon\ attachment}$

$F_{s\,clamp} > 0.422\,lb$

For rubber on metal

$F_{clamp} = 1.6\ lb = F_N$

$F_{s\,clamp} = F_N \mu_s$

$F_s\,clamp = 1.6\,lb\,(0.7)$

$F_s\,Clamp = 1.13$

$M_y = m_{frame} \cdot g \cdot d_1 + m_{(spoon + bearing + pulley + clamp)}\, g \cdot d_2$

$= 0.564\,lb\,(2.11\,in) + (0.422)(5.17\,in) = 3.3718\ lb\text{-}in$

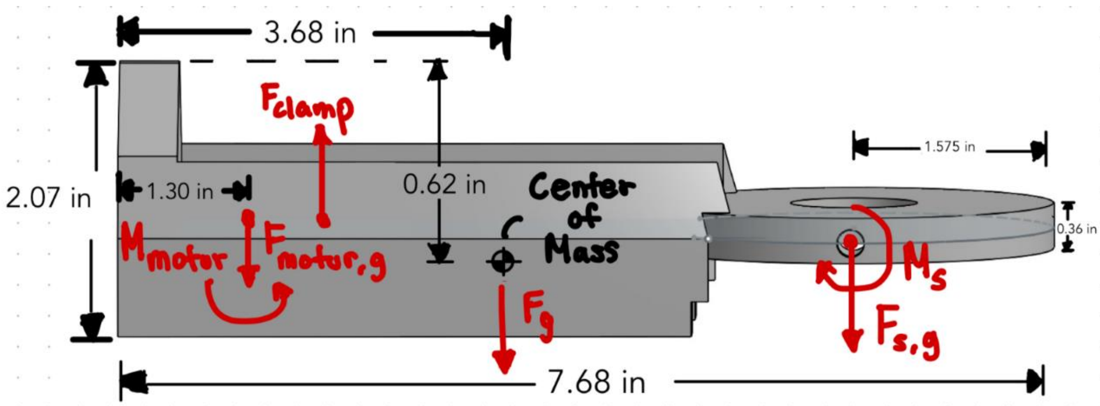
$F_{max,\,spoon} = 8.39\,lb \longrightarrow$ Largest normal Force on Spoon (through mollasses)

$M_z = (8.39)(5.17\,in) = 43.3763\ lb\text{-}in$

Force Calculations:

When calculating these forces, it was important to assume the highest possible loading scenario so that we can ensure our design is reliable and safe. In this design, it was assumed that the spoon was at the far end of the pot. This adds more moment to act on the clamp as it is now farther away which increases the moment arm. The force from the clamp was calculated by using the weight values from the other parts of the assembly. This value was important for calculating the force of static friction between the clamp and pot to ensure it was enough to hold the machine in place. Through this calculation it was clear that a material such as rubber was necessary to hold up the device since it has a high coefficient of static friction. The moments in both the Y and Z directions were calculated, with the Y moment being calculated from the weights of various parts on the device and the Z moment being calculated by assuming the spoon is stirring a viscous fluid, which in this case was molasses. This was done to ensure the clamp can handle even the most viscous fluids which are harder to stir.

1.2 Mechanism frame

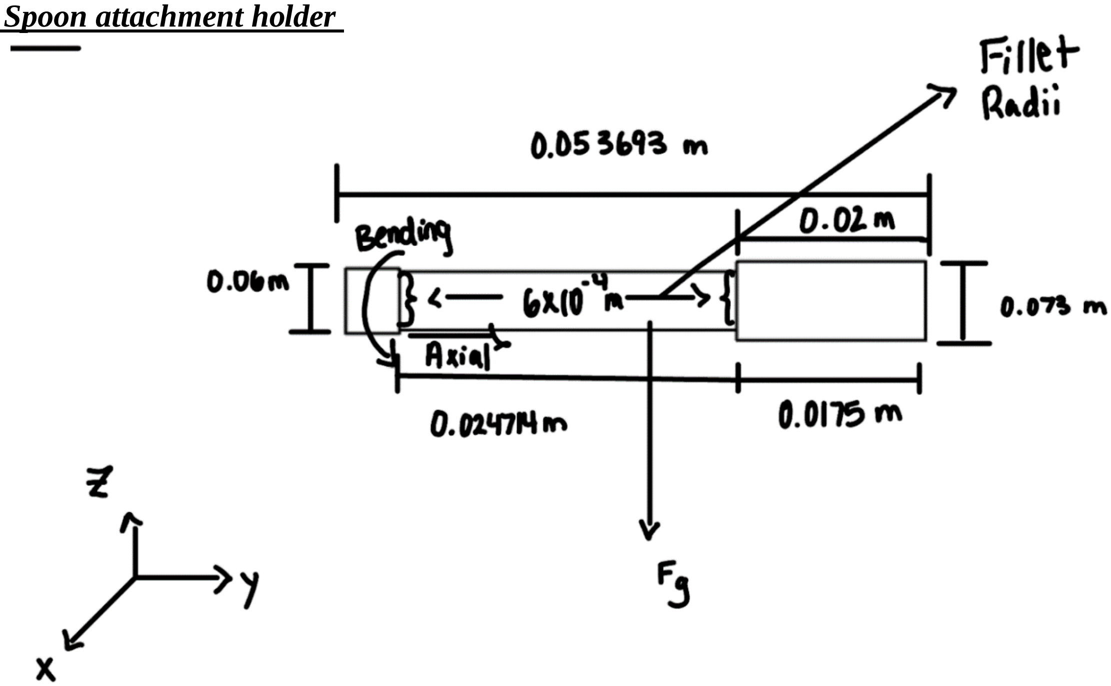


FBD:

Justification for analysis:

The mechanism frame is responsible for the secure attachment of the clamp handle and the spoon attachment holder. As it hosts vital components for actuating and rotating the utensil, most of the loads are not evenly distributed. This creates a “cantilever beam” moment where significant stresses and possible deflection may occur in the mechanism frame. As a result, analysis of the mechanism frame is crucial as there would be bending moments and shear forces acting on the frame as the forces are not distributed about its center of mass. Additionally, there is no risk of buckling based on the expected loading conditions. Since the motion of the mixing attachment is rotational, all stresses on the associated parts will be due to torsion and moment. Without the necessary axial stresses, there is no risk of buckling in this part.

1.3: Spoon attachment holder



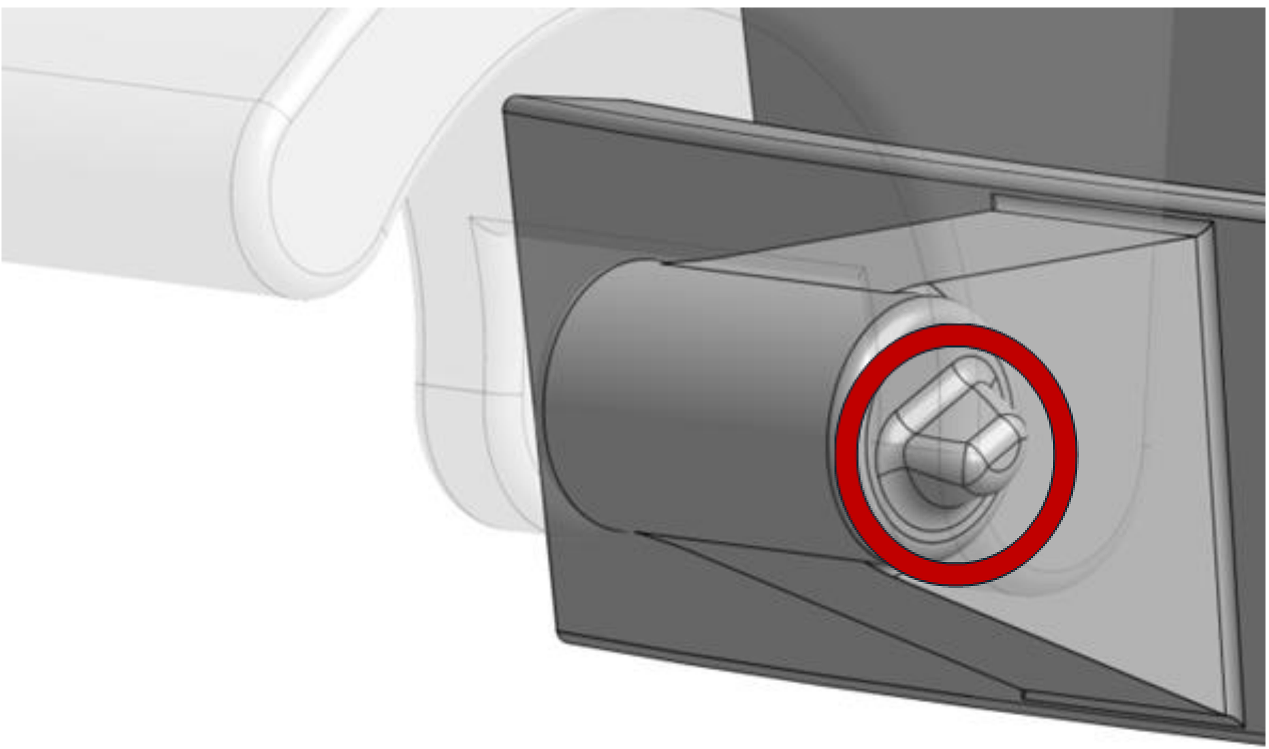
FBD:

Justification for analysis:
The spoon attachment holder is important to analyze since it directly impacts the functionality of the automatic stirrer. It needs to securely hold the stirring mechanism in place while it is moving so that there is an efficient and consistent stirring of the ingredients. Additionally, the mechanical stability of the holder is a main concern in preventing wobbling or excessive movement during the stirring process. The stability would ensure that the stirring motion is uniform and won't cause any damage to the stirring mechanism or even the holder.

To determine if the clamp provided enough force to hold the utensil in place, the analysis above was done. The force of the clamp is going in the y-axis, and the opposing force was coming from the horizontal component of the spoon's weight. This analysis showed that the required force to keep the spoon in place was 0.0219 lb, which will be achievable by our chosen spring.

1.4 Summary of Results

1.4.1 Clamping handle



Critical point identified:
This part of the clamp was chosen to be analyzed because it is a part that will experience high forces and has a stress concentration factor. This part is responsible for holding the majority of the weight of the device, so it is important to consider the bending moment that will be applied at this point.

Results of Static Analysis

$$D = 10\,mm,\quad r = 1\,mm,\quad d = 5.263\,mm \approx .2\,in$$

$$\frac{D}{d} = \frac{10}{5.268} = 1.9 \qquad \frac{r}{d} = \frac{1}{5.268} = .19$$

$$K_t\,\text{bending} \approx 1.4 \qquad \sigma = K_t\left(\frac{Mc}{I}\right)$$

$$\sigma_{bend} = 1.4 \cdot \frac{3.37 \cdot .1}{\frac{\pi \cdot .2^4}{64}} = 6007.1\,Psi$$

$$S_y = 13.7\,ksi > 6007.1\,Psi$$

To determine the bending stress on the part the moment in the Y direction had to be considered. To calculate the bending stress, we first had to determine the stress concentration factor at the point which through tables was found to be around 1.4. With this the bending stress could be calculated to be 6007 psi.

$$\sigma_{max} = \frac{6007.1}{2} + \sqrt{6007.1^2} = \underline{9010.65\,Psi}$$

$$\frac{S_y}{2n} = \frac{\sigma_{max} - \sigma_{min}}{2} \qquad \sigma_{max} = 9010.65 \quad \sigma_{min} = 0$$

$$\frac{13.7 \cdot 10^3}{2n} = \frac{9010.65}{2} \qquad n = 1.52$$

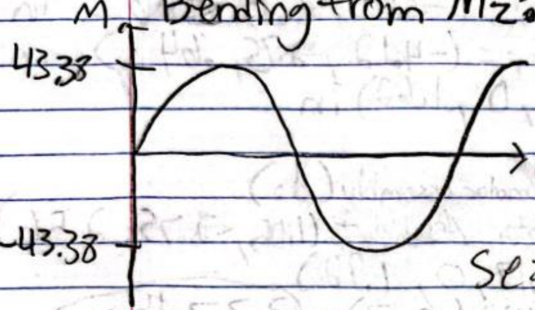
Once the bending stress was calculated the maximum principal stress was determined. This maximum principal stress came out to be 9010 psi. Through using maximum shear theory, the safety factor could be calculated, and it came out as 1.52, which is above the threshold of 1.

Buckling did not need to be considered when analyzing the clamping mechanism. This is because there is very little compression in the clamp and there are also no columns that could easily buckle. Since the part has a wide geometry and no columns buckling was not considered.

Results of fatigue analysis

Cyclic Loads

Bending from M_z:



$K_a = a S_{ut}^b = 1.21 \cdot 52^{-.067} = .929$

$K_c = 1 \quad K_e = .868$

$K_D = 1$

$K_g = 1 \quad K_b = .879 \cdot 1^{-.107} \approx 1$

$S_e = .868 \cdot .929 \cdot (.5 \cdot S_{ut}) = .403 S_{ut}$

$\sigma_B = \frac{Mc}{I} = \frac{43.38 \cdot 1/2}{.0104} = 2085.58 \text{ Psi} \qquad I = \frac{bh^3}{12} = \frac{1 \cdot .5^3}{12} = .0104$

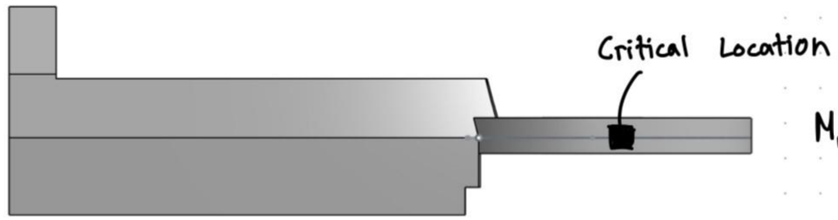
$\frac{2085.58}{.403 S_{ut}} + \frac{2085.58}{S_{ut}} = \frac{1}{2}, \quad \boxed{S_{ut} = 14.5 \text{ Ksi}}$

PPS is valid

Fatigue analysis was done on the top of the clamp to ensure that it could withstand stirring molasses. The top of the clamp was chosen because it is the part that will be experiencing the most forces from this motion. The motion was represented by a cyclic curve with positive and negative 43.38 representing the spoon stirring in a full rotation clockwise and counterclockwise. To calculate the stress from this moment, K values had to be calculated. These values were derived from tables and the geometry of the top of the clamp. Once the K values were determined the Se value could be calculated in terms of Sut. This allowed for Sut to be calculated using Goodman's equation. Through calculations it was found that Sut had to be at least 14.5 ksi, which is less than the 52 ksi that PPS has, which means that it is a valid material to use for the clamp and is not a concern for cyclic failure.

1.4.2 Mechanism frame

Results of Static Analysis



$$M_{max} = 52.35 \text{ lb-in}$$

Principle Stress Calculations

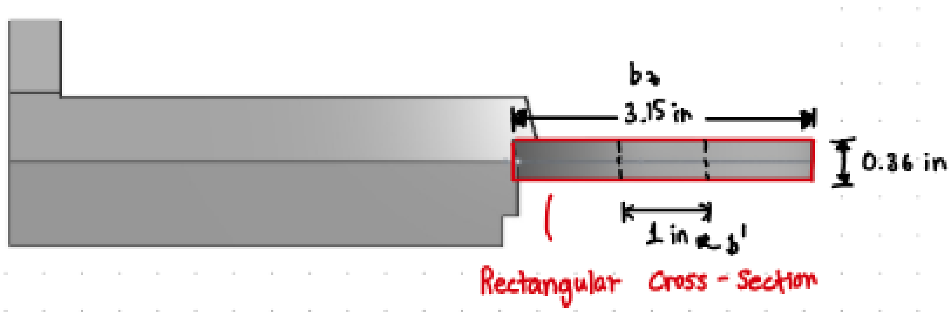
- $\sigma_A = \sigma_1 = \dfrac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\dfrac{\sigma_x + \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \dfrac{16.805 + 49.58}{2} + \sqrt{\left(\dfrac{16.805 + 49.58}{2}\right)^2 + 49.58^2} \approx 85.41 \text{ psi}$
- $\sigma_B = \sigma_2 = \dfrac{16.805 + 49.58}{2} - \sqrt{\left(\dfrac{16.805 - 49.58}{2}\right)^2 + 49.58^2} = -19.026 \text{ psi}$
- $\sigma_3 = 0$

Max Shear Theory

$$\sigma_{p,max} - \sigma_{p,min} = \frac{S_y}{2n} \Rightarrow n = \frac{13.7 \times 10^3}{85.41 + 19.03} \approx 131.18$$

For the static analysis, a critical location was selected to be the outer diameter of the frame where the utensil would be rotated. The principal stresses were determined by determining the bending and torsion on spoon attachment holder and the parts responsible for rotating it. The distances were determined from the center of the ball bearing to the edge of the spoon for the vector components to calculate the torque. From finding the torque using these distances, the bending and torsion components were found and used to calculate the principal stresses at the critical location shown above. After the principal stresses were found, max shear theory was used to calculate the factor of safety, n.

Results of Fatigue Analysis



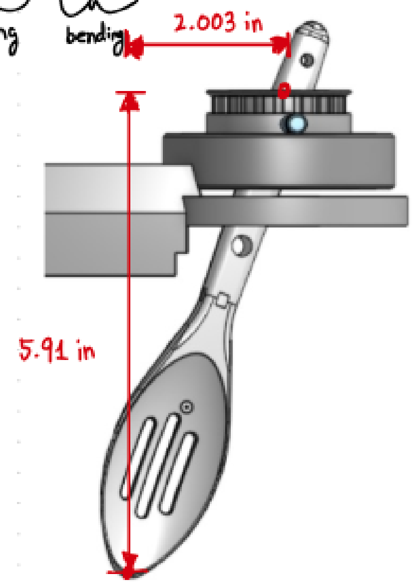
$$I_x = \frac{bh^3 - b'h'^3}{12} = \frac{(3.15)(.36)^3 - (1)(.36)^3}{12} = 0.00836 \text{ in}^4$$

$$I_y = \frac{hb^3 - h'b'^3}{12} = \frac{(.36)(3.15)^3 - (.36)(1)^3}{12} = 0.908 \text{ in}^4$$

$$J = I_x + I_y = 0.00836 + 0.908 = 0.916 \text{ in}^4$$

$$T = r \times F = (-2.003\hat{i} - 5.91\hat{k}) \times (-8.39\hat{j}) = \underbrace{-16.805\hat{k}}_{\text{bending}} \underbrace{-49.58\hat{i}}_{\text{torsion}}$$

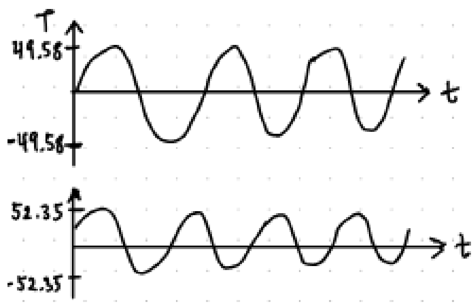
$$(2.003\hat{j} - 5.91\hat{k}) \times (8.39\hat{i}) = \underbrace{-16.805\hat{k}}_{\text{bending}} \underbrace{-49.58\hat{j}}_{\text{bending}}$$



Combination of Loading Modes

Torque: -49.58 lb-in to 49.58 lb-in

Bending: -52.35 lb-in to 52.35 lb-in



$$\tau = \frac{Tc}{J} = \frac{49.58(1.575)}{.916} = 85.2 \text{ psi}$$

$$\sigma_B = \frac{Mc}{I} = \frac{52.35(1.575)}{.908} = 90.805 \text{ psi}$$

Based on the spoon attachment’s position throughout the pot, there are different stresses applied to the frame. This results in the cyclic load described above. The maximum stress was calculated at the edge of the frame where the maximum bending and torsional stresses occur.

Using a maximum temperature of 212°F, the boiling temperature of water, there is negligible effects when considering k_d. To calculate k_a, the material was assumed to be machined and at the lowest region of the graph, leading to the value used in the calculations. Given a safety factor of 2, the ultimate stress needed in the frame is 1.3ksi. This justifies the use of PPS as described previously. While the ultimate stress of 52ksi of PPS may seem extreme, its lightweight properties and chemical inertness make it a perfect material for the W.H.I.S.K.

Fatigue Analysis

$k_a = 0.8785$

$k_b = 1$

$k_c = 1$

$k_d = 1$

$k_e = .868$

$$\bar{\sigma}_a = \bar{\sigma}_m = \left(\bar{\sigma}_b^2 + 3\tau^2\right)^{1/2} = \left(90.805^2 + 3(85.2)^2\right)^{1/2} = 173.27$$

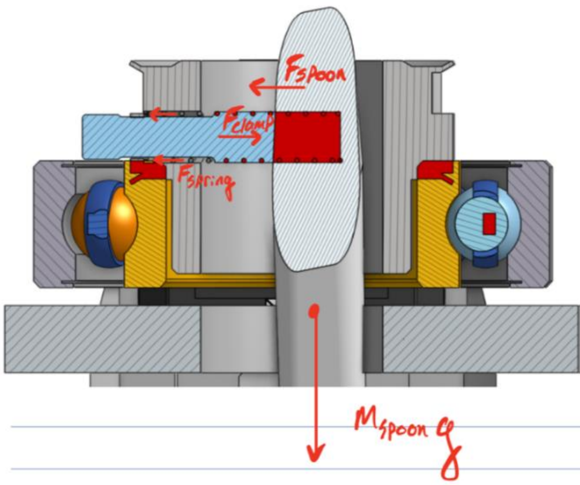
$$\frac{173.27}{.3682\,S_{ut}} + \frac{173.27}{S_{ut}} = \frac{1}{2}$$

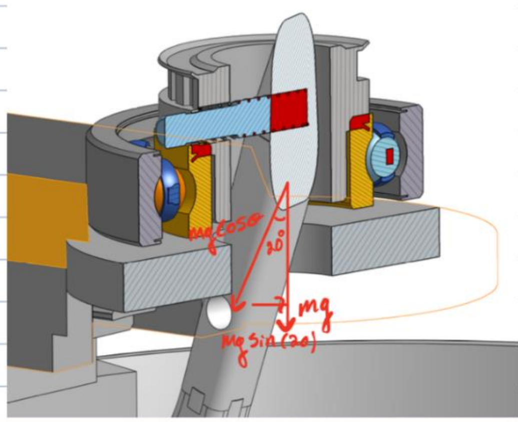
$$S_e = k_a k_b k_c k_d k_e (0.5 S_{ut}) = 0.3682 S_{ut}$$

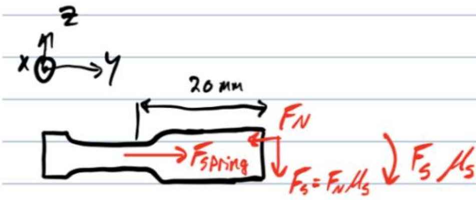
$S_{ut} = 1.3$ ksi → We can use PPS!

1.4.3 Spoon attachment holder

Results of Static Analysis







$F_{spring} > F_s$ to keep in place

$$F_{Spoon} = 0.07257478\ \text{Kg} \cdot 9.81 = 0.711959\ N \quad (0.16)$$

$$F_N = F_{Spoon}\sin(20°) = 0.2435\ N$$

$$F_s = F_N \mu_{PPS/wood} = 0.2435(0.4) = 0.097402\ N$$

$$M_x = F_s \cdot 20\text{mm} = 1.948\ N\text{-mm}$$

$$F_{spring} > 0.097402\ N > 0.021897\ \text{lbf}$$

Spring Chosen: 2.8 lb capacity

Results of Fatigue Analysis

Factor of safety based off fillet on right:

Spoon Attachment
weight: 0.05 lbs.

Made of PPS:

$$\text{Stress} = \frac{F_{SPRING}}{A} = \frac{2.8 \text{ lbs.}}{0.056 \text{ in.}^2} = \boxed{50 \text{ lbs.}/\text{in.}^2}$$

$$S_e = k_a k_b k_c k_d k_e S'_e$$

$$S_{ut} = 52 \text{ ksi} \leq 200 \text{ ksi}$$

$$k_a = a S_{ut}^{\,b} = (2.00)(52)^{-0.217}$$

$$k_a = 0.8485$$

$$S'_e = 0.5 S_{ut} = 0.5(52)$$

$$S'_e = 26 \text{ ksi}$$

$k_b = 1$ $\quad k_c = 1$ $\quad k_d = 1$ $\quad k_e = 0.868$

$$S_e = (0.8485)(1)(1)(1)(0.868)(26) \longrightarrow \boxed{S_e = 19.1489 \text{ ksi}}$$

$$\frac{r}{d} = \frac{0.6}{5.4} = 0.11 \qquad q \approx 0.8$$

$$\frac{D}{d} = \frac{6.8}{5.4} = 1.26 \qquad k_t \approx 1.6$$

$$k_f = 1 + q(k_t - 1) = 1 + 0.8(1.6 - 1) = 1.48$$

$$\sigma_{a\,\text{Bending}} = (0.5368)(0.4)(1.757) = 0.377 \text{ lbs./in.} \longrightarrow \frac{Mc}{I} = \frac{(0.377)(0.1065)}{\frac{\pi}{64}(0.213)^4} = 397.378 \text{ lbs./in.}^2$$

$$\sigma_{a\,\text{Axial}} = 50 \text{ lbs/in.}^2$$

$$\sigma_a = \sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \cancelto{0}{\tau_{xy}^2}} = \frac{397.378 + 50}{2} + \sqrt{\left(\frac{397.378 - 50}{2}\right)^2} = 397.378 \text{ psi}$$

$$\sigma_b = \frac{397.378 + 50}{2} - \sqrt{\left(\frac{397.378 - 50}{2}\right)^2} \longrightarrow \sigma_b = 50 \text{ psi}$$

$$\sigma_a - \sigma_b = \frac{S_y}{2n} \longrightarrow n = \frac{13.7 \times 10^3}{2(397.378 + 50)} = \boxed{15.31}$$

First, the endurance limit, Se, was calculated, yielding a value of 19.1489 ksi. k_a was determined by assuming the material would be machined. The value for K_e was calculated based off the reliability percentage being 95%, giving a K_e factor of 0.868. Bending stress as well as axial stress was calculated using the formulas Mc/I and F/A, respectively. These values, along with the yield stress for PPS, was used to give a factor of safety of 15.31, meaning it should not fail.

2. Connections Analysis

<u>2.1 Clamp & Rail interface</u>

Here, the connection is going to be of the clamp and rail interaction. The reason for using a bolted connections is because this the rail is a COTS (counter-off-the-shelf) item that already has tapped screws that allow for a bolted connection. The image below shows the cross section of the interaction which is 1.228" by 1.012". More dimensions regarding thickness and distances between the holes can be shown below.

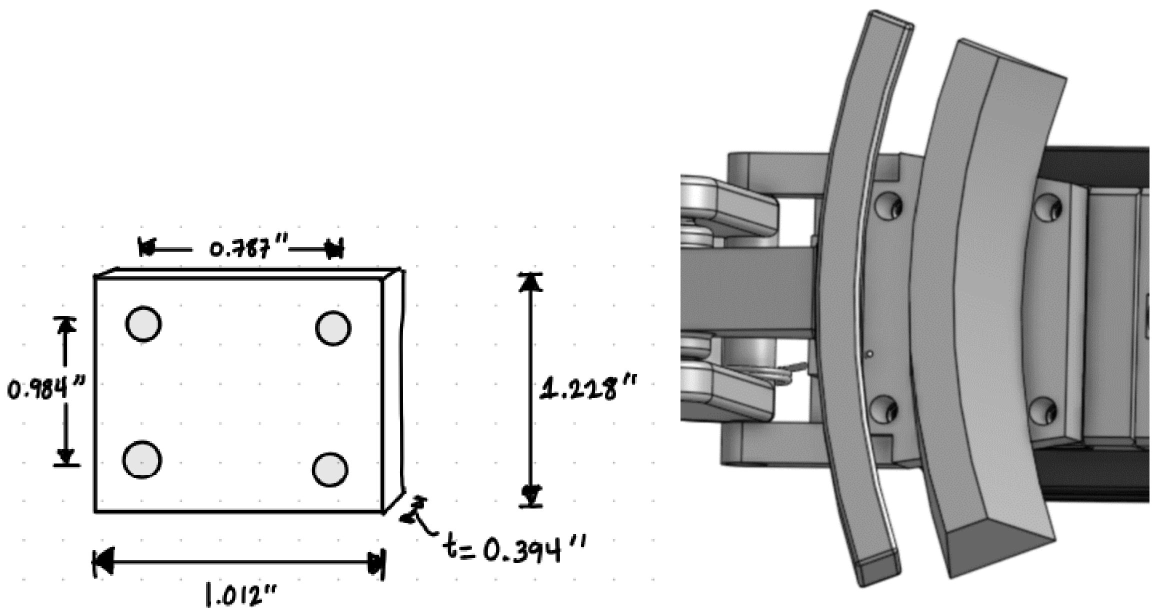


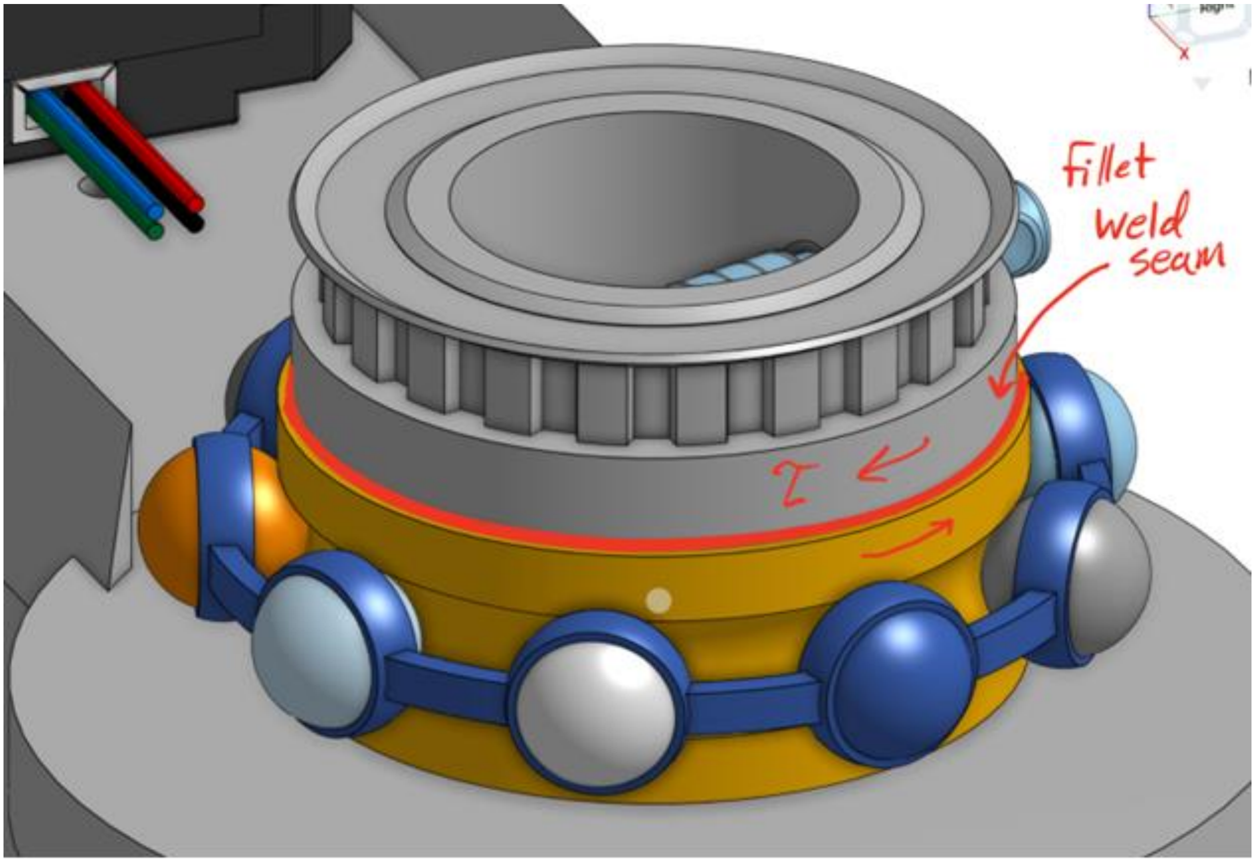
Figure 1 and 2: Bolt Connection for Clamp and Rail Interaction

Justification for analysis:

The clamp and rail interface is one of the most important areas of analysis since it holds the entire structure onto the pot. The clamp is responsible for holding the weight of the mechanism in place while it is stirring. If this connection fails at any point during the cooking process, the entire device will fail with it. This connection will be in shear as the force of the clamp creates a parallel force in the rail.

__2.2 Pulley to Bearing__

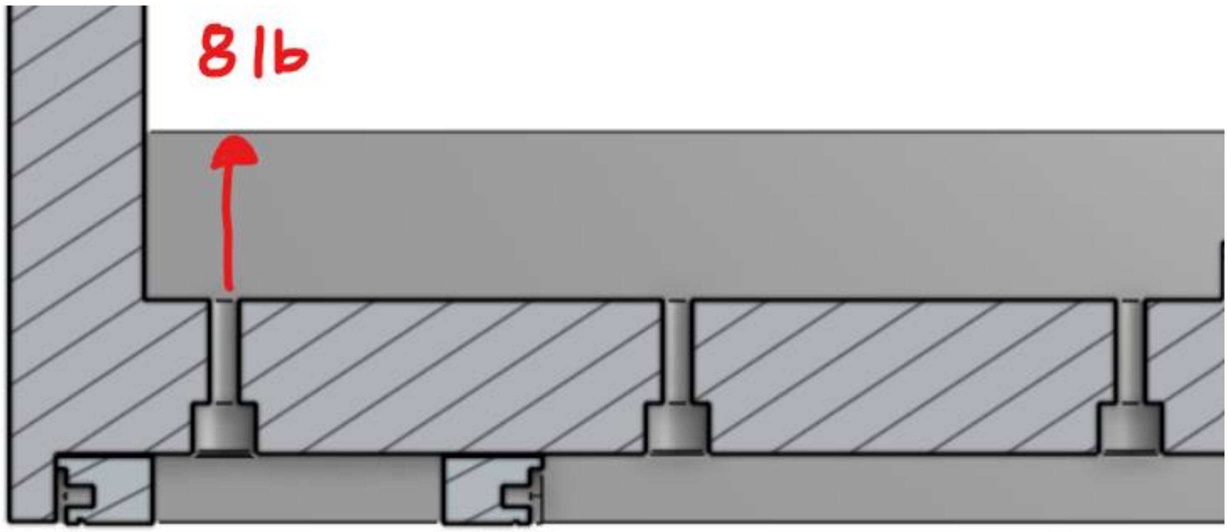
FBD:



Justification for analysis:

The type of connection here will be a fillet weld around the base of the pulley wheel and the inner ring of the ball bearing. We are using the maximum moment exerted on the spoon (and therefore the rotation mechanism) as our quantity for the shear experienced. We used a fillet weldment because the two adjoining surfaces are circular, so it seems like the easiest method for fixing them together in a secure manner.

__2.3 Linear Rail to Frame__



FBD:

Justification for analysis:
The linear rail to frame interface ensures the mixing motion does not remove the structure from the bottom half and the clamp mechanism. This is important for adjusting the depth of the attachment for various pot sizes. If this connection fails during use, the mechanism will detach from the clamp and fall into the pot. The bolts for this connection are in tension with no moment acting on the bolt pattern and a centroid on the middle bolt as seen in the free body diagram. A bolt of UNC #6-32 was chosen for the diameter specifications within the width of the rail and limited force in the member.

2.4 Summary of Results

2.4.1 Clamp & Rail interface

Fastener Stiffness Calculations

The drawing of the fulcrums shown below was used to eventually determine the proper safety factors for bolt failure, part loosening, and fatigue failure. The external load that will be distributed to the bolts was determined to be 8.39 lb which is the maximum force applied to the edge of the spoon. An important note is that the steel railing had tapped holes as shown from McMaster Carr and didn't require a nut connection. Also, the material selected for the railing was a 1/4 – 32 UNC Grade 5 Hex bolt.

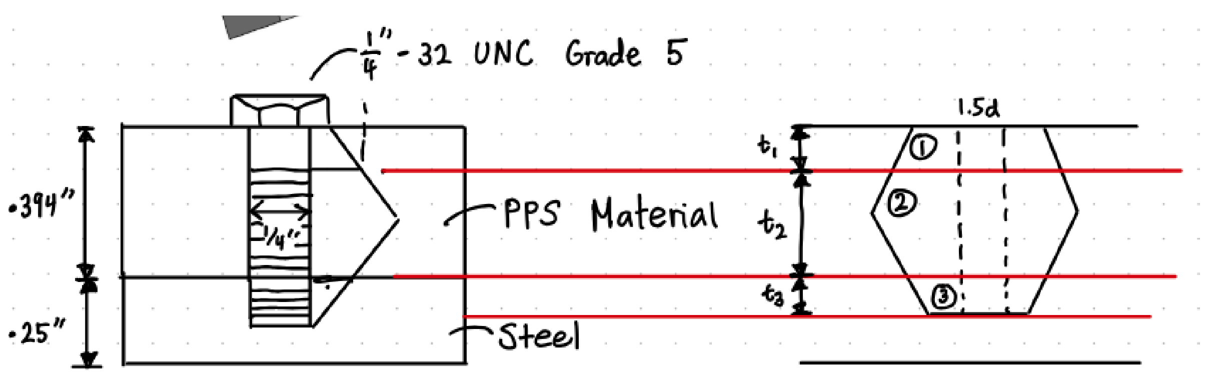


Figure 3: Fulcrum and Drawing of Bolt Connection

Using Table 8-7, the fastener stiffness (k_b) was determined to be $8.65 * 10^5$ lb/in. Calculations are shown below.

Table 8-7 Calculations

$$L > h + 1.5d$$

$$L > 0.394 + 1.5(0.25) \approx 0.769'' \rightarrow 1''$$

$$l = h + d/2 = 0.394 + 0.125 = 0.519''$$

$$L_T = 2d + 1/4 = 2(1/4) + 1/4 = 0.75''$$

$$l_d = L - L_T = 1 - 0.75 = 0.25''$$

$$l_t = l - l_d = 0.519 - 0.25 = 0.269''$$

$$A_d = \frac{\pi d^2}{4} = \frac{\pi (1/4)^2}{4} \approx 0.0491 \text{ in}^2$$

$$A_t = 0.00909 \text{ in}^2$$

$$t_1 + t_2 = .394'' \rightarrow t_1 = 0.1'', \quad t_2 = .294''$$

$$t_3 = 0.25''$$

Fastener Stiffness

$$K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(.0491)(.00909)(30 \times 10^6)}{(.0491)(.269) + (.00909)(.25)}$$

$$= 8.65 \times 10^5 \text{ lb/in}$$

Material Stiffness Calculations

The material stiffness was determined for all three fulcrums and eventually used to determine the material stiffness of the entire interaction of the bolt connection. The material stiffness (k_m) was determined to be $1.94 * 10^6$ lb/in.

$$K_{m_1} = \frac{0.5774\pi(3\times10^6)(0.25)}{\ln\frac{(1.155(0.1)+0.375-0.25)(.375+.25)}{(1.155(0.1)+0.375+0.25)(.375-.25)}} = 2.8\times10^6 \text{ lb/in}$$

$$K_{m_2} = \frac{0.5774\pi(30\times10^6)(0.25)}{\ln\frac{(1.155(0.294)+1.53-0.25)(.375+.25)}{(1.155(0.294)+1.53+0.25)(.375-.25)}} = 10.15\times10^6 \text{ lb/in}$$

$$K_{m_3} = \frac{0.5774\pi(30\times10^6)(0.25)}{\ln\frac{(1.155(0.25)+.375-0.25)(.375+.25)}{(1.155(0.25)+.375+0.25)(.375-.25)}} = 16.65\times10^6 \text{ lb/in}$$

$$\frac{1}{K_m} = \frac{1}{K_{m_1}} + \frac{1}{K_{m_2}} + \frac{1}{K_{m_3}} \rightarrow K_m \approx 1.94\times10^6 \text{ lb/in}$$

$$D_1 = 1.5d = 1.5(0.25) = 0.375''$$

$$D_2 = D_1 + 2\tan 30^\circ = 1.53''$$

$$D_3 = D_1 = 0.375''$$

$$E = 435 \text{ ksi}$$

$$K_m = \frac{0.5774\pi Ed}{\ln\frac{(1.155t+D-d)(D+d)}{(1.155t+D+d)(D-d)}}$$

After determining the fastener stiffness and material stiffness, the value of C was determined to be 0.308.

$$C = \frac{K_b}{K_b + K_m} = \frac{0.865}{0.865+1.94} \approx 0.308$$

Before conducting any factor of safety calculations, the pre-tension force was calculated for non-permanent connections (serviceable) using the formula $F_i = 0.75A_tS_p$ where A_t represents the Tensile-Stress Area for the bolt and S_p represents the minimum proof strength. The pre-tension force (F_i) was found to be 579.49 lb.

Factor of Safety Against Bolt Failure Calculations

The FOS for Bolt Failure, n_L was found to be 74.05.

<u>FOS Against Bolt Failure</u>

$$n_L = \frac{A_t S_p - F_i}{CP} = \frac{(.00909)(85000) - 579.49}{0.308\,(8.39)} \approx 74.75$$

Factor Of Safety Against Part Loosening

The FOS against Part Loosening, n_0 was determined to be 99.81.

<u>FOS Against Part Loosening</u>

$$n_0 = \frac{F_i}{(1-C)P} = \frac{579.49}{(1-0.308)(8.39)} \approx 99.81$$

Fatigue Safety Factor

If the external load fluctuates between 0 and 8.39 lb, $P_a = P_m = \frac{P_{max}}{2} = 4.195\ lb$. The S_e was determined from Table 8-17 showing the endurance strength of a 1/4th inch SAE grade 5 bolt to be 18.6 ksi. Using the formula for fatigue safety factor (n_f), the value was found to be 53.1.

<u>Fatigue Safety Factor</u>

$$\left.\begin{aligned} S_e &= 18.6\ \text{ksi} \rightarrow \text{Table 8-17} \\ P_a = P_m &= \frac{8.39}{2} \approx 4.195\ lb \\ S_{ut} &= 120\ \text{ksi} \end{aligned}\right\} \quad n_f = \frac{S_{ut}A_t - F_i}{CP\left[\frac{S_{ut}}{S_e} + 1\right]} = \frac{(120000)(.00909) - 579.49}{(0.308)(4.195)\left[\frac{120000}{18600} + 1\right]} \approx 53.1$$

To conclude, the values for all of the FOS were considerably high, ensuring that the connections in the design are safe when under an external load of 8.39 lb from the possible maximum forces applied to the edge of the spoon from Appendix A.

2.4.2 Pulley to Bearing

Centroid: $d = 44.98\,mm = 1.77\,in$
centroid is in the center of the pulley assembly

We know τ_{base} is equal to our max moment from the spoon which is 43.3763 lb·in.

Solve for P to find other τ values

$\tau_{base} = \frac{P}{2\pi r h}$, $43.38 = \frac{P}{2\pi \cdot \frac{1.77}{2} \cdot .098}$, $\boxed{P = 23.64\,lb}$

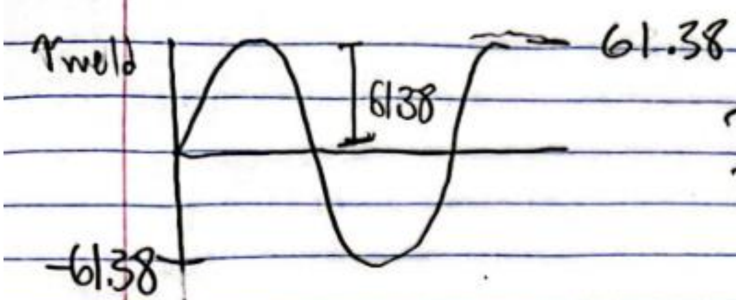
assuming 2.5 mm throat height

$\tau_{weld} = \frac{P}{2\pi r \cdot .707 h} = \frac{23.64}{2\pi \cdot \frac{1.77}{2} \cdot .707 \cdot .098} = \boxed{61.36\ lb \cdot in}$

To determine material we chose a saftey factor of 3

$\sqrt{3}\,\tau_{weld} \doteq \frac{S_y}{n}$, $\sqrt{3} \cdot 61.36 = \frac{S_y}{3}$, $\boxed{S_y = 318.83\,Psi}$

$\tau_{base} \doteq \frac{.45 S_y}{n}$, $43.38 = \frac{.45 S_y}{3}$, $\boxed{S_y = 325.35\,Psi}$



$\tau_a = 61.38$ lbin
$\tau_m = 0$

$K_{fs} = 2.7$ based on table 9.5

$$\frac{K_{fs}\sqrt{3}\tau_a}{S_e} + \frac{K_{fs}\sqrt{3}\tau_m}{S_{ut}} = \frac{1}{n_f}$$

$$\frac{2.7\cdot\sqrt{3}\cdot 61.38}{S_e} = \frac{1}{3}, \quad \boxed{S_e = 861.138\ Psi}$$

Based on these values of S_y and S_e we will use E60xx as a weld metal. Based off this material our actual factor of saftey is calculated below:

$S_{ut} = 62$ Ksi $\quad S_y = 50$ Ks $\quad S_e = K_a \cdot .5 S_{ut}$

$$K_a = a\,(S_{ut})^b = 12.7(62)^{-.758} = .556$$

$$S_e = .556 \cdot .5 \cdot 62 = 17.239\ Ksi$$

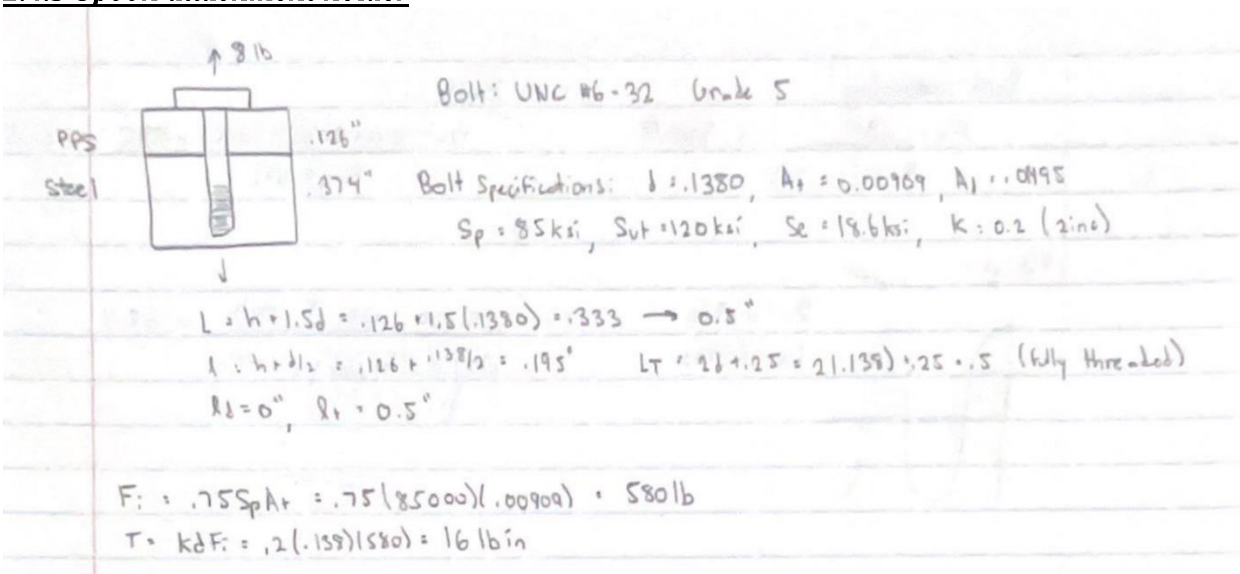
$$\sqrt{3}(61.36) = \frac{50\cdot 10^3}{n}, \quad n = 470.461 \text{ (for weld)}$$

$$43.38 = \frac{.4\cdot 50\cdot 10^3}{n}, \quad n = 461.042 \text{ (for base)}$$

$$\frac{2.7\cdot\sqrt{3}\cdot 61.38}{17.239\cdot 10^3} = \frac{1}{n_f}, \quad n_f = 60.076 \text{ (fatigue)}$$

Initial calculations were used to determine the material properties of the weld with a chosen safety factor of 3. This was done by calculating the P value through the shear at the base which was calculated previously as the moment the spoon experiences which was 43.38. Through this P was calculated and tau weld was found. Using these tau values and a safety factor of 3 the yield strengths of the base and weld were calculated to be both around 300 psi. The final value to be calculated is the endurance strength of the weld which is shown below along with new safety factors calculated based on an E60XX weld. Shown in the above calculations the final safety factors are all well above 1 which means that the selected material of E60XX is more than strong enough to support the loads experienced by the weld.

2.4.3 Spoon attachment holder



The drawing on the top left of the image shows the bolt in tension connecting the rail to the frame. The external load applied to the bolt is 8.39lb based on the component analysis done previously. A UNC #6-32 Grade 5 bolt was chosen due to its stiffness and small size for the required rail. This image also displays the bolt specifications, required length calculation of 0.5", and tightening torque of 16lb-in.

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(.01495)(.00909)(30\times10^6)}{(.01495)(.5)} = 545400$$

$$k_{m_1} = \frac{.5774\pi(435\times10^3)(0.138)}{\ln\left(\frac{(1.155(.126)+.207-.138)(.207+.138)}{(1.155(.126)+.207+.138)(.207-.138)}\right)} = 139176$$

$$k_{m_2} = \frac{.5774\pi(30\times10^6)(.138)}{\ln\left(\frac{(1.155(.124)+.145-.138)(.145+.138)}{(1.155(.124)+.145+.138)(.145-.138)}\right)} = 2826750$$

$$k_{m_3} = \frac{.5774\pi(30\times10^6)(.138)}{\ln\left(\frac{(1.155(.25)+.207-.138)(.207+.138)}{(1.155(.25)+.207+.138)(.207-.138)}\right)} = 7165175$$

$$k_m = \left(\frac{1}{139176} + \frac{1}{2826750} + \frac{1}{7165175}\right)^{-1} = 130234$$

$$C = \frac{545400}{545400 + 130234} = 0.81$$

This section displays the required stiffness of the bolt and member calculations needed to calculate the C value for failure analysis. A C value of 0.81 was found.

Part Loosening

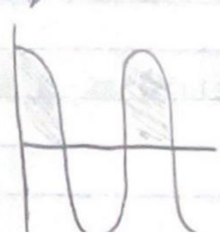
$$n_L = \frac{580}{8.39(1-.81)} = 363.8$$

Bolt

$$n_0 = \frac{0.00909(85000) - 580}{.81(8.39)} = 28.35$$

Using the initial force of 580lbs and C value of 0.81, the member loosening factor of safety was calculated to be 364. The bolt factor of safety was found to be 28.4.

Fatigue



$$P_a = 8.39/2$$

$$P_m = 8.39/2$$

$$n_f = \frac{120000(.00909) - 580}{.81\left(4.195\left(\frac{120}{168}\right) + 4.195\right)} = 18.5$$

For the fatigue analysis, the bolt will experience a force from -8.39 to 8.39lb. This creates an applied force on the bolt of 0lb to 8.39lb. The fatigue factor of safety was calculated to be 18.5 using the tabulated values for Sut and Se based on a Grade 5 bolt.

4. Powertrain

4.1 Main Pulley and Belt

The stepper motor is the main driving force for the stirring motion of our device. The stepper motor can submit a torque on the spoon by using a pulley system connecting the motor directly to the gear on the spoon assembly. This main pulley is essential to analyze because it will tell us how powerful of a motor we need to easily mix even the most viscous fluids. **We want a motor that is powerful enough to stir any fluid while not being overpowering or too pricey.** Analyzing the pulley also gives us useful information such as the pretension and the length of the pulley.

The length of the pulley was found by determining dimensions for the center distances of the driven and driver pulley and the diameters of both pulleys. The following three formulas were used for open belt to calculate this:

$$\emptyset_D = \pi - 2sin^{-1}\frac{D-d}{2C} \text{ (1)}$$

$$\emptyset_d = \pi + 2sin^{-1}\frac{D-d}{2C} \text{ (2)}$$

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\emptyset_d + d\emptyset_d) \text{ (3)}$$

where the center distances of both of the pulleys, C, was found to be 114.11 mm, the length of the belt found using this formula was 434.1 mm, the diameter of the small belt, d, was found to be 9.8 mm, and the diameter of the big belt, D, was found to be 33.338 mm. Using these variables, the length of the belt was calculated to be 434.1 mm.

Results of Analysis

Based on an expected input torque of 16.78lb-in and a required speed of 204rpm, the NEMA 23 stepper motor is an appropriately sized choice. It has a maximum holding torque of 29.2lb-in and a current of 2A at the expected torque. This motor is rated for high temperatures, critical for the cooking heat the unit will experience during use.

These values were calculated by using the maximum possible force against the spoon which is listed in Appendix A below. The rpm was assumed to be 60 rpm as it was the lowest angular speed of a competitor Kitchen Aid Utensil stirrer. The gear ratio of the pulleys can be found by dividing the diameter of the driven pulley to the diameter of the driver pulley and this value was found to be 3.4. To find the recommended power rating in horsepower, the following formula was used:

$$HP = \frac{T_{input}*n}{63000} \text{ (4)}$$

where T_{input} is the input torque and found to be 16.77 lb-in, n is the speed of the pulley which was found by multiplying the gear ratio (3.4) by the rpm of the spoon (60) which led to a value of 204 for n, and 63000 is the unit conversion factor.

Below shows the calculations for the forces F1, F2, and Fmax. These forces were calculated using values found after deciding what belt to use. Using values previously calculated such as the rpm and torque of the chosen motor, the force Fc was able to be calculated to be 361N. Using this value and the two equations for F1 and F2, a system of equations was set up and F1 and F2 were solved for to be 789N and 402N. Finally the maximum force the belt experiences is calculated using a Ks value of 1.2. This maximum value was calculated to be 947N.

$w = .09\ g/mm \quad d = 9.8mm$

$n = 204rpm$

$F_c = \frac{w}{g}(rw)^2 = \frac{.09}{9810}\left(\frac{9.8}{2} \cdot 2\pi \cdot 204\right)^2 = 361.9N$

$T = (F_1 - F_2)^{d}/_{2}, \quad 1896 = (F_1 - F_2)^{9.8}/_{2}$

$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\phi}, \quad$ Assuming $f = .8$

$\phi = \pi - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$

$\phi = \pi - 2\sin^{-1}\left(\frac{33.38 - 9.8}{2 \cdot 114.11}\right) = 2.935\ rad$

$\frac{F_1 - 361.9}{F_2 - 361.9} = e^{.8 \cdot 2.935}$

After substituting:

$F_1 = 789.72\ N$

$F_2 = 402.78\ N$

$F_{max} = K_s F_1 = 1.2 \cdot 789.72 = \underline{947.66N}$

Conclusion

The development of the W.H.I.S.K. (Wireless Hands-free Intelligent Stirring Kitchen-aid) represents a significant advancement in kitchen automation technology. Through rigorous analysis and careful material selection, our engineering team has ensured the device meets the stringent demands of modern cooking environments. The comprehensive examination of critical components such as the clamping handle, mechanism frame, and spoon attachment holder demonstrates the robustness and reliability of the W.H.I.S.K. in maintaining consistent stirring, even under high-stress conditions. The connections analysis further validates the stability and durability of the device, with factors of safety well above acceptable thresholds, guaranteeing secure operation throughout its lifecycle. By integrating high-performance materials like Polyphenylenesulphide (PPS) and employing meticulous engineering practices, the W.H.I.S.K. promises to enhance cooking efficiency, prevent food from burning, and free cooks to focus on other tasks. This innovative kitchen-aid tool not only simplifies meal preparation but also ensures a higher quality of cooking, ultimately leading to more enjoyable culinary experiences.

Appendix A

To determine the maximum forces applied to the edge of the spoon, one of the most viscous food items, molasses, was used. A simplified version of the Navier-Stokes equation was used, seen in Equation 1, with the assumption that the mixing attachment is a sphere and is completely submerged in the fluid.

$$F = 6\pi r v \eta \tag{1}$$

Where r is the radius of the mixing attachment, v is the velocity of the attachment, and η is the viscosity of molasses.

$$F = 6\pi\left(\frac{.355}{2}\right)(0.1778\pi)(20) = 37.377N = 8.39lbf$$

With a spoon diameter of 35.5mm and a speed of 60rpm, which is the standard slowest mixing speed in commercial mixers, the maximum drag force was found to be 8.39lbf.

PPS properties:

Sy = 13.7ksi

Su = 52ksi