Final Project:
Boat’s Trailer Main Axis

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Design Course for B.S. in Mechanical Engineering Department
University of Puerto Rico at Mayagüez, Mayagüez, PR

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Introduction

Puerto Rico is completely surrounded by water, with more than a hundred beaches. This causes a big interest in diving, navigation, surfing in the Puerto Rican. This is one of the reason we see a big number of boats and boat’s trailers on the road. Moving a boat from ground to water seems like an easy task. But when you start to analyze all the different parameters that are taken into the design of a boat trailer, you see that this in is more complicated. Of all the different considerations that need to be taken into account the most important is the structure of the trailer. In this design project, we analyze the overall design of the boat’s trailer main axis, the forces and the stresses that act on it, the selection of the correct material, and the durability of it, among others.
Objective

To provide a complete analysis of the boat’s trailer main axis that will aid in the construction of future trailers and improve their reliability, by applying the knowledge obtained in the Machine Component Design I Course.
For our design project we are going to analyze the main axis of the boat’s trailer. We choose this part because this is the most important part of the trailer because if the main axis fails, the whole trailer will fail. The main axis is the part that is under most forces and stresses because it is the main structural element of the trailer. It must hold the weight of the boat, must hold the reactions from the wheels, must hold the damages caused by corrosion, and must hold the overall structure of the boat. We will first concentrate our analysis on the forces and stresses that act on the boat trailer statically. Then we will calculate the deflection of static load. Once we have the deflection we will make the selection of the most efficient material. We will finish with the analysis of the axis under dynamic loads. In this part we will estimate the stresses, the component life and the safety factor.

Our goal of our design is to produce a reliable and functional product for the customers, that will give them security and durability. We want to provide a boat’s trailer main axis that will last longer than the ones that are in the market. As engineers we must be very aware of safety, because of this we will give a safety factor to the material to enable its safety. The purpose for the design of this trailer is for it to be as usable as possible, with this we will help minimize the disposal of failed trailers, thus minimizing contamination. We have as our customer base, the owners of boats that are used in the sea, since Puerto Rico is surrounded by sea water. The material selection of the main axis will demonstrate if the product is manufacturability. We have to choose a material that can withstand the high forces acting on the axis and that is cost-effective.
Design Details

Our design is inspired in the boat’s trailer in Figure 1. This trailer is design to hold a boat weights 5,000 lbs and measures 22.5 fts. In the Figure 2 we can see the main axis. In this picture we see the effect of corrosion on this important part of the trailer. We know as engineers that if the main part of a structure fails, the whole structure fails. For this reason we must provide a material that is both anticorrosion and capable of holding the different loads and stresses.

Figure 1: The Boat

Figure 2: Effect of corrosion on boat’s main axis
• Preliminary Design

We have design our main axis as a rectangular beam with a hollow end. The axis is 90 inches long. The solid part of the front face of the axis is 4 inches in height and 4 inches in width. The hollow part of the axis is 2 inches in height and 2 inches in width, it is 1 inch from each sides of the solid part of the axis (see Figure 3 and Figure 4). This hollow part is made so the main axis will be lighter to the overall weight of the trailer because the lighter the trailer, the less force the car produces and the less gas the car consumes. It will also be cheaper to produce because less material is needed.

Here are some 3-D views made in SolidWorks of the main axis.

![Figure 3: Isometric View](image1)

![Figure 4: Trimetric View](image2)

![Figure 4: Front View](image3)

![Figure 5: Side View](image4)

Below are two drawings which depict the different dimensions of the main axis. There is also a close-up of the front view in which the dimensions. See the appendix for a clearer and more detailed drawing.
Figure 6: 2-D Drawing with dimensions

Figure 7: 2-D Drawing (detailed Front View) with dimensions
1. Calculations of stresses under static loads

In our force analysis we see what are the forces, stresses and strains that act on the axis. We will calculate Shear diagrams and Moment diagrams. We will present the calculations for the normal stresses $\sigma_x$, shear stresses $\tau_{xy}$, principal stresses $\sigma_{1,2}$ and maximum shear stresses $\tau_{\text{max}}$ for different points in the cross-sectional area. We will then demonstrate normal, shear, principal, and maximum shear stresses for different points in the cross-sectional area of our main axis. We will leave the dimensions of our preliminary design unaltered.

For our analysis we made the following assumptions:

- We will analyze the main axis as a rectangular beam.
- We will analyze the static loads. We will not take into consideration fluctuating and variable loads.
- We will analyze the weight of the boat on the main axis, when in reality this weight is distributed to support beam and this beam exerts a force on the axis.
- We will analyze the cross-sectional area as a square (called the analysis cross-section, see Figure 13). For purpose of analysis we will ignore the fillet that the main axis has (called the real design cross-section, see Figure 12).
- We will neglect the weight of the material, $W_m = 0$.

We are designing for a boat that weight 5,000 lbs dry. We estimated that the weight of the rails that holds the boat is 1,000 lbs total, 500 lbs in each side. This will give the combined weight ($W_C$) of 6,000 lbs. This weight is distributed evenly on two support beams. These supports beams hold the boat in its position. They are placed 18 in from the wheel. We made the assumptions that the weight on the support beams will be the weight on the main axis. This gives two forces on the axis, as shown in Figure 8. Figure 8 shows the free body diagram of our main axis. For our static analysis we will neglect the weight of the beam.
Figure 8: Schematic of the forces that act on the main axis

Given data:

\[ W_c = 6,000 \text{ lb} \]
\[ W_c/2 = 3,000 \text{ lb} \]
\[ W_c/2 = P_1 = P_2 \]

\[ \sum M_{RG1} = 0 \]
\[ - \left( \frac{W_c}{2} \right)(18 \text{ in}) - \left( \frac{W_c}{2} \right)(72 \text{ in}) + (R_{G2})(90 \text{ in}) = 0 \]
\[ -(3,000 \text{ lb})(18 \text{ in}) -(3,000 \text{ lb})(72 \text{ in}) + (R_{G2})(90 \text{ in}) = 0 \]
\[ R_{G2} = 3,000 \text{ lb} \]

\[ \sum F_y = 0 \]
\[ R_{G1} - \left( \frac{W_c}{2} \right) - \left( \frac{W_c}{2} \right) + R_{G2} = 0 \]
\[ R_{G1} - 3,000 \text{ lb} - 3,000 \text{ lb} + 3,000 \text{ lb} = 0 \]
\[ R_{G1} = 3,000 \text{ lb} \]
This page presents the Load diagram (Figure 9), Shear diagram (Figure 10) and Momentum Diagram (Figure 11), for the forces action on our main axis. For Figure 10 and Figure 11, the x-axis is distance in inches (in).

**Figure 9:** Load Diagram

**Figure 10:** Shear Diagram; where the y-axis is force in pounds (lbs)

**Figure 11:** Momentum Diagram; where the y-axis is momentum in pounds-inch (lbs*in)
From Figure 10, we see that the maximum shear stress, $V_{\text{MAX}}$, is 3,000 lbs; and from Figure 11, we see that the maximum momentum, $M_{\text{MAX}}$, is 54,000 lbs*in. With this data we will calculate the normal stresses $\sigma_x$, shear stresses $\tau_{xy}$, principal stresses $\sigma_{1,2}$ and maximum shear stresses $\tau_{\text{max}}$ for different point in the cross-sectional area.

For our calculations we assumed a square cross-section (see Figure 13). We will present the formulas used for finding this different stresses and will do the calculation on the Point B. Note that Figure 14 contains all the values of the stresses for the different points calculated using Excel. In this section, we did not change the dimensions of our preliminary design.

- **Moment of Inertia, $I$**:

$$I = \frac{1}{12} (b_1^4 - b_2^4)$$

$$I = \frac{1}{12} (4^4 - 2^4) = 20 \text{ in}^4$$

- **Normal Stress, $\sigma_x$**:

$$\sigma_x = -\frac{(M_{\text{MAX}})y}{I}$$

$$\sigma_x = -\frac{(54,000 \text{ lbs*in})(1 \text{ in})}{(20 \text{ in}^4)} = -2,700 \text{ lbs/in}^2$$

- **Shear Stress, $\tau_{xy}$**:

$$\tau_{xy} = \frac{V_{\text{MAX}} Q}{Ib} = \frac{V_{\text{MAX}} (A \times \bar{y})}{Ib}$$

$$\tau_{xy} = \frac{(3,000 \text{ lbs})(1 \text{ in} \times 4 \text{ in})(1.5 \text{ in})}{(20 \text{ in}^4)(4 \text{ in})} = 225 \text{ lbs/in}^2$$
- Principal Stresses, $\sigma_{1,2}$:

$$\sigma_{1,2} = \left(\frac{\sigma_x}{2}\right) \pm \sqrt{\left(\frac{\sigma_y}{2}\right)^2 + (\tau_{xy})^2}$$

$$\sigma_{1,2} = \left(\frac{-2,700 \text{ lbs/in}^2}{2}\right) \pm \sqrt{\left(\frac{-2,700 \text{ lbs/in}^2}{2}\right)^2 + (225 \text{ lbs/in}^2)^2}$$

$\sigma_1 = 18.6216 \text{ lbs/in}^2$

$\sigma_2 = -2,718.6 \text{ lbs/in}^2$

- Maximum Shear Stress, $\tau_{\text{max}}$:

$$\tau_{\text{max}} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xy})^2}$$

$$\tau_{\text{max}} = \sqrt{\left(\frac{-2,700 \text{ lbs/in}^2}{2}\right)^2 + (225 \text{ lbs/in}^2)^2}$$

$\tau_{\text{max}} = 1368.62 \text{ lbs/in}^2$

Figure 12: Real Design Cross-Section

Figure 13: Analysis Cross-Section
<table>
<thead>
<tr>
<th>POINT</th>
<th>y (in)</th>
<th>$\sigma_x$ (psi)</th>
<th>$\tau_{xy}$ (psi)</th>
<th>$\sigma_1$ (psi)</th>
<th>$\sigma_2$ (psi)</th>
<th>$\tau_{\text{max}}$ (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>2</td>
<td>-5400</td>
<td>0</td>
<td>0</td>
<td>-5400</td>
<td>2700</td>
</tr>
<tr>
<td>B</td>
<td>1</td>
<td>-2700</td>
<td>225</td>
<td>18.6216</td>
<td>-2718.6</td>
<td>1368.62</td>
</tr>
<tr>
<td>C</td>
<td>0</td>
<td>0</td>
<td>525</td>
<td>525</td>
<td>-525</td>
<td>525</td>
</tr>
<tr>
<td>D</td>
<td>-1</td>
<td>2700</td>
<td>-225</td>
<td>2718.62</td>
<td>-18.622</td>
<td>1368.62</td>
</tr>
<tr>
<td>E</td>
<td>-2</td>
<td>5400</td>
<td>0</td>
<td>5400</td>
<td>0</td>
<td>2700</td>
</tr>
</tbody>
</table>

Figure 14: Table with all the values for the different stresses
2. Calculations of deflections under static loads

For our calculation of deflection under static loads we used the same assumptions as the stresses under static load. We calculated the deflection by the integration of the shear-force equation and the load equation. For our calculation we made three cuts. The first two cuts are from left to right, whereas the third cut is from right to left. The first cut is from \(0 < x < \frac{L}{5}\), the second cut is from \(\frac{L}{5} < x < 4\frac{L}{5}\) and the third and last cut is from \(0 < x < \frac{L}{5}\). The cuts are shown in Figure 15. A deflection diagram is provided by the Figure 16. The variables, \(W/2\), \(P_1\), and \(P_2\) are the same.

![Figure 15: Representation of the three cuts made](image)

![Figure 9: Load Diagram](image)
**Figure 16:** Deflection in the main axis

- **First Cut**

\[ 0 < x < \frac{L}{5} \]

\[ EIv'' = R_A x \]

\[ EIv' = \frac{R_A x^2}{2} + C_1 \]

\[ EIv = \frac{R_A x^3}{6} + C_1 x + C_4 \]

**Border Conditions:**

**B.C.1** \( v = 0 \) @ \( x = 0 \) \( \Rightarrow \) \( C_1 = 0 \)

**B.C.2** \( v = 0 \) @ \( x = 0 \) \( \Rightarrow \) \( C_4 = 0 \)

\[ R_A = \frac{W_b}{2} = P \]

\[ EIv = \frac{Px^3}{6} \]
• Second Cut

\[
\begin{align*}
\frac{L}{5} < x < \frac{4L}{5} & \\
EIv'' &= R_A x - Pl \left( x - \frac{L}{5} \right) \\
EIv' &= \frac{R_A x^2}{2} - Pl \left( \frac{x^2}{2} - \frac{L}{5} x \right) + C_2 \\
EIv &= \frac{R_A x^3}{6} - Pl \left( \frac{x^3}{6} - \frac{L}{10} x^2 \right) + C_2 x + C_5
\end{align*}
\]

Border Conditions:

B.C.3 \quad v' = 0 \quad @ \quad x = \left( \frac{L}{2} \right) \Rightarrow C_2 = -\frac{PL^2}{10}

\[R_A = Pl = P\]

\[EIv = \frac{PL}{10} x^2 - \frac{PL^2}{10} x + C_5\]

• Third Cut

\[0 < x < \frac{L}{5}\]

\[
\begin{align*}
EIv'' &= R_B x \\
EIv' &= \frac{R_B x^2}{2} + C_3 \\
EIv &= \frac{R_B x^3}{6} + C_3 x + C_6
\end{align*}
\]

Border Conditions:

B.C.4 \quad v' = 0 \quad @ \quad x = 0 \Rightarrow C_3 = 0

B.C.5 \quad v = 0 \quad @ \quad x = 0 \Rightarrow C_6 = 0
\[ R_B = \frac{W_B}{2} = P \]
\[ EIv = \frac{Px^3}{6} \]

**Border Conditions:**

**B.C.6** \[ \nu\left(\frac{L}{5}\right)^+ = \nu\left(\frac{L}{5}\right)^- \]

\[ \frac{Px^3}{6} = \frac{PL}{10} x^2 - \frac{PL}{10} x + C_5 \]

\[ \frac{P}{6} \left(\frac{L}{5}\right)^3 = \frac{PL}{10} \left(\frac{L}{5}\right)^2 - \frac{PL}{10} \left(\frac{L}{5}\right) + C_5 \]

\[ \frac{PL^3}{750} = \frac{PL^3}{250} - \frac{PL^3}{50} + C_5 \]

\[ C_5 = \frac{13PL^3}{750} \]

**Final Equation for Second Cut:**

\[ EIv = \frac{PL}{10} x^2 - \frac{PL^2}{10} x + \frac{13PL^3}{750} \]

- **Maximum Deflection**
  
  Maximum deflection occurs at \( x = L/2 \).
  
  \[ EI\delta_{\text{max}} = \frac{PL^3}{40} - \frac{PL^3}{20} + \frac{13PL^3}{750} \]

\[ EI\delta_{\text{max}} = -\frac{23}{3000} PL^3 \]
3. Calculations of material indices and material selection

In this part, we will transform our design into materials specifications. Here we need to see, by calculations, which materials work better for our design. We will rate the different materials in their ability to meet the given requirements and specification. We will rank the material indices to see which one of the given materials works best for our design. We will use the final equation obtained from the deflection under static loads to constraint our selection of materials. Our requirements are:

- **Function** Main axis for a boat’s trailer
- **Objective** Minimize Mass
  \[ m = \rho V = \rho L (\rho b_1^2 - \rho b_2^2) \]
- **Constraints** Stiffness Prescribed
  \[ EI \delta_{max} = -\frac{23}{3000} PL^3 \]

  \[ E \left( \frac{b_1^4}{12} - \frac{b_2^4}{12} \right) \delta_{max} = -\frac{23}{3000} PL^3 \]

- **Free variables** Thickness (Given by the height of \(b_2\))
  \[ b_2 = \left( -\frac{23}{250 E} \frac{PL^3}{\delta_{max}} + b_1 \right)^{\frac{1}{4}} \]

![Figure 17: Cross Section of axis](image-url)
Substituting the free variable in the mass equation we obtain:

\[
m = L \left[ \rho b_i^2 - \rho \left( -\frac{23}{250} \frac{PL^3}{\delta_{\text{max}}} \frac{1}{E} + b_i^4 \right) \right]^{1/2} \approx \frac{\rho}{E^2}
\]

This gives us the materials indices (MI’s) \( \frac{E^2}{\rho} \).

With this parameter we look in the Material Selection Chart. We see that we need to use the Young’s Modulus – Density Chart. This chart is shown in Figure 18.

![Young’s Modulus – Density](image)

**Figure 18:** Young’s modulus – Density
For this slope, Young’s Modulus-Density Chart gives us these possible candidates:

- Galvanized steel
- Aluminum alloy
- Titanium alloy

From these candidates we have to eliminate Galvanized steel because it is the one that most easily corrodes. The density of aluminum is approximately one third of the density of steel; as a result, structures made of aluminum have the potential to be lighter weight than the same structure made from steel; however, aluminum is not as strong or stiff as steel, and when these factors are considered, the potential weight savings of aluminum over steel is reduced.

Since the elastic modulus of aluminum is one third the modulus of steel, a structure built from aluminum will deflect more than the same structure built from steel. This can be significant when the structure is subjected to wind loads which may cause movement of the structure which will result in fatigue loading.

The life of your trailer is dependent on its ability to withstand fatigue stress as you drive down the road. Galvanized steel has fatigue strength about 30% higher than aluminum. When welded, heat treatable aluminum alloys (6000 series) lose a significant portion of their strength. For example, 6061-T6's tensile strength drops from 42000 to 24000 psi, there is about a 50% reduction in strength and is so significant that it is necessary to locate welds in low stress areas in the design made from aluminum. This often requires the use of thicker tube sections or the addition of extra bracing. Aluminum does not rust like untreated steel does; however, it does corrode, pit and develop a loose white powdery surface.

Considering our other material: Titanium alloys. Titanium alloys features excellent corrosion resistance, which stems from a thin oxide surface film which protects it from atmospheric and ocean conditions as well as a wide variety of chemicals. Such alloys have very high tensile strength and toughness, light weight and an ability to withstand extreme temperatures. However, the high cost of both raw materials and processing limit their use to military applications, aircraft, spacecraft, medical devices, and some premium sports equipment and Consumer electronics. The high cost of titanium alloy components may limit their use to applications for which lower-cost alloys, such as aluminum and stainless steels. The relatively
high cost is often the result of the intrinsic raw material cost of metal, fabricating costs and the metal removal costs incurred in obtaining the desired final shape. Also, Titanium is considered to be 40% lighter than steel and 60% heavier than aluminum.

Since Galvanized steel corrodes more easily than an aluminum alloy and a Titanium alloy is heavier than aluminum, we selected the Aluminum alloy. From the Aluminum alloy family we chose the Aluminum 6061-T6 as our material for the main axis. This Aluminum is perfect for our application because it combines high strength and high resistance to corrosion while being widely available. This material is used for a lot of applications including: camera lens, bike frames, brake pistons, aircraft fittings, etc. The Figure 19 demonstrates the composition of this alloy and the Figure 20 shows some of the mechanical properties. This information was taken from MatWeb of the Aluminum 6061-T6.

<table>
<thead>
<tr>
<th>Component</th>
<th>Min</th>
<th>Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum, Al</td>
<td>95.8</td>
<td>98.6</td>
</tr>
<tr>
<td>Chromium, Cr</td>
<td>0.04</td>
<td>0.35</td>
</tr>
<tr>
<td>Copper, Cu</td>
<td>0.15</td>
<td>0.4</td>
</tr>
<tr>
<td>Iron, Fe</td>
<td></td>
<td>0.7</td>
</tr>
<tr>
<td>Magnesium, Mg</td>
<td>0.8</td>
<td>1.2</td>
</tr>
<tr>
<td>Manganese, Mn</td>
<td></td>
<td>0.15</td>
</tr>
<tr>
<td>Silicon, Si</td>
<td>0.4</td>
<td>0.8</td>
</tr>
<tr>
<td>Titanium, Ti</td>
<td>0.15</td>
<td></td>
</tr>
<tr>
<td>Zinc, Zn</td>
<td></td>
<td>0.25</td>
</tr>
</tbody>
</table>

**Figure 19:** Composition of Aluminum alloy 6061-T6
<table>
<thead>
<tr>
<th>Properties Physical</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, lb/in³</td>
<td>0.0975</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mechanical</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardness, Brinell</td>
<td>95</td>
</tr>
<tr>
<td>Hardness, Knoop</td>
<td>120</td>
</tr>
<tr>
<td>Hardness, Rockwell A</td>
<td>40</td>
</tr>
<tr>
<td>Hardness, Rockwell B</td>
<td>60</td>
</tr>
<tr>
<td>Hardness, Vickers</td>
<td>107</td>
</tr>
<tr>
<td>Ultimate Tensile Strength, psi</td>
<td>45,000</td>
</tr>
<tr>
<td>Tensile Yield Strength, psi</td>
<td>40,000</td>
</tr>
<tr>
<td>Elongation at Break, %</td>
<td>12</td>
</tr>
<tr>
<td>Elongation at Break, %</td>
<td>17</td>
</tr>
<tr>
<td>Modulus of Elasticity, ksi</td>
<td>10,000</td>
</tr>
<tr>
<td>Notched Tensile Strength, psi</td>
<td>47,000</td>
</tr>
<tr>
<td>Ultimate Bearing Strength, psi</td>
<td>88,000</td>
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<tr>
<td>Bearing Yield Strength, psi</td>
<td>56,000</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.33</td>
</tr>
<tr>
<td>Fatigue Strength, psi</td>
<td>14,000</td>
</tr>
<tr>
<td>Fracture Toughness, ksi-in½</td>
<td>26.4</td>
</tr>
<tr>
<td>Machinability, %</td>
<td>50</td>
</tr>
<tr>
<td>Shear Modulus, ksi</td>
<td>3,770</td>
</tr>
<tr>
<td>Shear Strength, psi</td>
<td>30,000</td>
</tr>
</tbody>
</table>

**Figure 20:** Mechanical Properties of Aluminum alloy 6061-T6

Now that we have the material, we can calculate the Von Misses stress in order to calculate the static safety factor of our axis for our static analysis. We will calculate the safety factors for the point A and point B of Figure 15.

![Figure 21: Cross-sectional area of the main axis with the points](image)

Adapting the values calculated in Figure 14 (for the cross-sectional of Figure 14) and using Figure 21 as reference, we obtain the following results presented in Figure 22 for the stresses in the section.
Calculating the Von Mises stresses:

**Point A:**

\[
\sigma_{VM,A} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2}
\]

\[
\sigma_{VM,A} = \sqrt{(0)^2 + (-5400)^2 - (0)(-5400)} = 5.4 \text{ksi}
\]

**Point B:**

\[
\sigma_{VM,B} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2}
\]

\[
\sigma_{VM,B} = \sqrt{(-525)^2 + (525)^2 - (525)(-525)} = 0.9093 \text{ksi}
\]

We will now calculate the safety factor using the Distortion Energy Theory. For this theory we will use the Yield strength \( S_y \) of our selected material, Aluminum 6061 – T6, \( S_y = 40 \) ksi. (The Yield strength was retrieved from MatWeb.)

**Point A:**

\[
n_A = \frac{S_y}{\sigma_{VM,A}}
\]

\[
n_A = \frac{40 \text{ksi}}{5.4 \text{ksi}} = 7.074
\]

**Point B:**

\[
n_B = \frac{S_y}{\sigma_{VM,B}}
\]

\[
n_B = \frac{40 \text{ksi}}{0.9093 \text{ksi}} = 43.9899
\]

From both safety factors we can see that our preliminary design works under static loads.
4. Calculations of stress concentrations in critical sections

For this section we made the following assumptions:

- We are working with Aluminum 6061 – T6.
- It has a ground powder surface.
- The aluminum is annealed.
- The fillet radius is \( r = 0.25 \) in.
- A reliability of 99.9%
- It will be subjected at room temperature.
- For the material, the ultimate tensile strength is: \( S_{UTS} = 45 \) ksi
- The stress concentration factor for bending is \( k_t = 1.7 \)
- The stress concentration factor for shear is \( k_{t_s} = 1.5 \)
- The material will be design unlimited life.
- We will work with the rectangular cross-sectional area (we will not include the round corners of the original design)
- We will assume the maximum force when the trailer is loaded with the boat, thus taking into consideration the weight of the axis. The minimum force will be when the trailer is not loaded with the boat.
- The area of the cross-section is given by:
  \[
  A = b_1^2 - b_2^2 = (4in)^2 - (2in)^2 = 12in^2
  \]
- The inertia of the cross-sectional area is given by:
  \[
  I = \frac{1}{12} \left( b_1^4 - b_2^4 \right) = \frac{1}{12} \left( 4^4 - 2^4 \right) = 20in^4
  \]

Calculation the weight of the material

\[
 w = \rho \mathcal{A} = \rho L \left( b_1^2 - b_2^2 \right)
\]

Where the density of the Aluminum 6061 – T6 is \( \rho = 0.0975 \) lb/in\(^3\), the length of the axis is \( L = 90 \) in, the base of the cross-section is \( b_1 = 4 \) in, and the hole in the cross-section is \( b_2 = 2 \) in.

\[
 w = \left( 0.0975 \frac{lb}{in^3} \right) (90in) \left[ (4in)^2 - (2in)^2 \right] = 105.3 \text{lbs}
\]
The maximum force is the sum of the weight of the boat, weight of the main axis and the screws, washers, welds, etc. Whereas the minimum force will be the weight of the main axis and the screws, welds, etc. These values will be approximated to the following:

\[ F_{\text{max}} \approx 6,150 \text{ lbs} \]
\[ F_{\text{min}} \approx 150 \text{ lbs} \]

Now we proceed to calculate the mean force that acts on the main axis and the amplitude force that acts on the main axis. The following demonstrate the values and the calculations needed to obtain them.

\[ F_{\text{amplitude}} = \frac{6,150 \text{ lbs} - 150 \text{ lbs}}{2} = 3,000 \text{ lbs} \]
\[ F_{\text{mean}} = \frac{6,150 \text{ lbs} + 150 \text{ lbs}}{2} = 3,150 \text{ lbs} \]

We will now calculate the stresses in the points A and B of the Figure 21.

![Figure 21: Cross-sectional area of the main axis with the points](image)

- **Point A**

![Figure 23: Forces acting on Point A](image)
\[ \sigma_{\text{bending}} = \frac{Mc}{I} = \frac{F(45 \text{in})(2 \text{in})}{20 \text{in}^4} = \frac{9F}{2} \]

\[ \sigma_{\text{amp,b}} = \frac{9F_{\text{amp}}}{2} = \frac{(3000 \text{lb})(45 \text{in})(2 \text{in})}{20 \text{in}^4} = 13.5 \text{ksi} \]

\[ \sigma_{\text{mean,b}} = \frac{9F_{\text{mean}}}{2} = \frac{(3150 \text{lb})(45 \text{in})(2 \text{in})}{20 \text{in}^4} = 14.175 \text{ksi} \]

- **Point B**

![Figure 24: Forces acting on Point A](image)

\[ \tau_{\text{shear}} = \frac{4F}{3A} = \frac{F}{9} \]

\[ \tau_{\text{amp,s}} = \frac{F_{\text{amp}}}{9} = \frac{3000}{9} = 0.3333 \text{ksi} \]

\[ \tau_{\text{mean,s}} = \frac{F_{\text{mean}}}{9} = \frac{3150}{9} = 0.350 \text{ksi} \]

Since the material is assumed to be annealed aluminum we obtain the value of \( \sqrt{a} \), from the Table 6.7 of the slide 9 from the lecture of Chapter 6B by Dr. Pablo G. Cáceres-Valencia. Checking with the material properties of Aluminum 6061 - T6 from MatWeb, we see that \( S_{\text{UTS}} = 45 \text{ ksi} \). With this we obtained the value of \( \sqrt{a} = 0.111 (\text{in}^{0.5}) \)

We will now calculate the notch sensitivity, given by \( q \), of the main axis of our design. This is given by the Peterson’s equation, which is the following equation:

\[ q = \frac{1}{1 + \sqrt{\frac{a}{r}}} = \frac{1}{1 + \frac{0.111}{\sqrt{0.25}}} = 0.8183 \]
Since we have bending and shear, we have to calculate the stress concentration factor for each of these forces. We now to calculate the stress concentration factor for bending. This is given by the following equation:

\[ K_f = 1 + q(K_t - 1) = 1 + (0.8183)(1.7 - 1) = 1.573 \]

Now we have to calculate the stress concentration factor for the shear stresses. This values is given by the following equation:

\[ K_{f,s} = 1 + q(K_{t,s} - 1) = 1 + (0.8183)(1.5 - 1) = 1.409 \]

We can now calculate the stress concentration in the points A and B.

- **For Point A:**

  \[ \sigma_{amp,sc} = \sigma_{amp,b} \times K_f = (13.5 ksi)(1.573) = 21.24 ksi \]

  \[ \sigma_{mean,sc} = \sigma_{mean,b} \times K_f = (14.175 ksi)(1.573) = 22.30 ksi \]

- **For Point B:**

  \[ \tau_{amp,sc} = \tau_{amp,b} \times K_{f,s} = (0.3333 ksi)(1.409) = 0.4697 ksi \]

  \[ \tau_{mean,sc} = \tau_{mean,b} \times K_{f,s} = (0.350 ksi)(1.409) = 0.4931 ksi \]
5. Calculations of stresses under dynamic loads

In this section we now calculate the stresses under dynamic loads. We will continue with the assumptions made in the previous section. From the material we know that the ultimate tensile strength is $S_{UTS} = 45$ ksi. With this value we are able to calculate the fatigue strength ($S_f'$). We use the equation found in the slide 34 of the lecture Chapter 6A from Dr. Pablo G. Cáceres-Valencia. Since $S_{UTS} < 48$ ksi, we use the following:

$$S_f' = 0.4(S_{UTS}) = 0.4(45\text{ksi}) = 18\text{ksi}$$

This value must be modified to account for the physical and environmental differences between test experiments. We will use the following to obtain a corrected value:

$$S_f = k_{\text{size}} k_{\text{load}} k_{\text{surface}} k_{\text{temperature}} k_{\text{reliability}} S_f'$$

- $k_{\text{size}} = 0.7864$

$$A_{95} = 0.05(b_1^2 - b_2^2) = 0.6\text{in}^2$$

$$d_{\text{equivalent}} = \sqrt{\frac{A_{95}}{0.0766}} = \sqrt{\frac{0.6}{0.0766}} = 2.7987\text{in}$$

For $0.3\text{in} \leq d_{\text{equivalent}} \leq 10\text{in}$

$$k_{\text{size}} = 0.869 d_{\text{equivalent}}^{-0.097} = 0.869(2.7987)^{-0.097} = 0.7864$$

These equations are taken from the slide 39 and the slide 40 of the lecture Chapter 6A from Dr. Pablo G. Cáceres-Valencia.

- $k_{\text{load}} = 1$

This value is taken from slide 41 of the lecture Chapter 6A from Dr. Pablo G. Cáceres-Valencia, because it is bending.

- $k_{\text{surface}} = 0.9596$

This equation is taken from the ground surface finish from slide 37 of the lecture Chapter 6A from Dr. Pablo G. Cáceres-Valencia.

$$k_{\text{surface}} = a S_{UTS}^b = (1.34)(45)^{-0.085} = 0.9596$$

- $k_{\text{temperature}} = 1$

This equation is taken from the slide 43 41 of the lecture Chapter 6A from Dr. Pablo G. Cáceres-Valencia, because it is at room temperature (see assumptions).
• \( k_{\text{reliability}} = 0.753 \)

This value is taken from the slide 44 of the lecture Chapter 6A from by Dr. Pablo G. Cáceres-Valencia, for a reliability of 99.9% (see assumptions).

Therefore the corrected strength is given by:

\[
S_f = (0.7864)(0.9596)(1)(0.753) 18\text{ksi} = 10.228\text{ksi}
\]

Now we will calculate the Von Mises stresses for the amplitude and mean stresses.

\[
\sigma_{VM} = \sqrt{\sigma_{sc}^2 + 3\tau_{sc}^2}
\]

• For Point A:

\[
\sigma_{VM,\text{amp}} = \sqrt{\sigma_{\text{amp,sc}}^2} = \sqrt{(21.24)^2} = 21.24\text{ksi}
\]

\[
\sigma_{VM,\text{mean}} = \sqrt{\sigma_{\text{mean,sc}}^2} = \sqrt{(22.30)^2} = 22.30\text{ksi}
\]

• For Point B:

\[
\sigma_{VM,\text{amp}} = \sqrt{3\tau_{\text{amp,sc}}^2} = \sqrt{3(0.4697)^2} = 0.8135\text{ksi}
\]

\[
\sigma_{VM,\text{mean}} = \sqrt{3\tau_{\text{mean,sc}}^2} = \sqrt{3(0.4931)^2} = 0.8541\text{ksi}
\]

Now we will calculate the safety factor, \( n \), for the main axis. We will use the Modified-Goodman equation. This is given by:

\[
n = \frac{1}{\frac{\sigma_{VM,\text{amp}}}{S_f} + \frac{\sigma_{VM,\text{mean}}}{S_{UTS}}}
\]

• For Point A:

\[
n = \frac{1}{\frac{\sigma_{VM,\text{amp}}}{S_f} + \frac{\sigma_{VM,\text{mean}}}{S_{UTS}}} = \frac{1}{\frac{21.24}{10.228} + \frac{22.30}{45}} = 0.389
\]
• For Point B:

\[
n = \frac{1}{\frac{\sigma_{VM,amp}}{S_f} + \frac{\sigma_{VM,mean}}{S_{UTS}}} = \frac{1}{\frac{0.8135}{10.228} + \frac{0.8541}{45}} = 10.15
\]

From the safety factors, we see that the critical point is point A. We also see that our preliminary design fails.

• **Main Axis Redesign**

From the previous section we saw that our main axis failed under static loads. In order to correct this we will change the dimensions of the axis. We could have change the material, but since our ideal material is an Aluminum alloy, we decided that we will change the dimensions. We used the software program Excel to calculate the best combination. We created a spreadsheet in Excel where we can change the dimensions of the axis and find a safety factor (see the appendix for a picture of the spreadsheet). We are going to design for a safety factor of \( n = 2 \). In the Figure 25 we see a combination of possible dimensions, with the safety factor for Point A and the weight of the material. The dimensions of \( b_1 \) and \( b_2 \) are given in inches, whereas the weight of the material is given in pounds (lbs).

<table>
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*Figure 25: Possible combinations of new dimensions*

We chose the combination of \( b_1 = 8 \) and \( b_2 = 6 \) because it gave us a safety factor of \( n = 2.02 \) and it was the most light combination. With these changes we have design a main axis that does not fail. In the next page we present the corrected main axis.
**Figure 26:** 2-D Drawing with dimensions

**Figure 27:** 2-D Drawing (detailed Front View) with dimensions
In Figure 28, we see the S-N diagram for our design.

![S-N Diagram](image)

**Figure 28: S-N Diagram**

The equation for this line is $S_N = aN^b$. We can obtain the values of $a$ and $b$ with the following border line conditions:

- $S_N = S_M @ N_1 = 10^3$ cycles
- $S_N = S_f @ N_2 = 5 \times 10^8$ cycles

$$b \log\left(\frac{10^3}{5 \times 10^8}\right) = \log\left(\frac{40.5}{10.23}\right)$$

$$b = -0.104856$$

$$\log(10.23) = \log a + (-0.104856) \log(5 \times 10^8)$$

$$a = 83.564$$

Using these values in the line equation, we obtain the final value for the line. This is given by:

$$S_M = 83.564N^{-0.104856}$$
In our design we completed every requirement. Even though we did, we are unsure if our assumption were correctly made. We design a main axis to hold a 5,000 lb boat, and we only used a weight of 6,000 lbs for our calculations. Once we finished the project we saw that we neglected the motors of the boat, the appliances inside the boat, and the weight due to sea water, among others. We could not make the required corrections due to time constraints.

Another important part are the static and dynamic safety factors. The safety factors show that our original design fails due to dynamic loads. We are impressed that it does not fails in dynamic loads, when we have a safety factor \( n = 7 \) in static loads. Even though the static safety factors are commonly higher than the dynamic, we did not expect such a sudden change (from operational to failure). We think that this big discrepancy is due to the assumptions that we made.

The first assumption is that in the static load, the forces are in two points whereas for our dynamic load analysis is concentrated in one point. Since we only applied the weight of the boat on the main axis, we could have missed important forces acting on the main axis. If we miss any force in our analysis it will cause a big change in our stress calculations.

Another assumption that might cause an error are the values of the notch radius, and the stress concentration factors. For the stress concentration factors, we did not find any chart that gives the section or form of our design. Because of this we assumed a value, very similar to the ones used in the class examples. The assumption of values became the main problem of our design because we have assumed values with our limited knowledge. We hope that our assumptions are close to the real value.

One of the biggest advantages of our design is the material we selected. We selected Aluminum 6061 T6, which provides high strength and high resistance. This alloy has a wide range of uses from turbines blades to camera lenses. We needed a material that was light and resistive to corrosion; Aluminum 6061 T6 gave us those qualities. The rectangular cross-section of our design enables the force distribution along the axis. Also the hole made in the axis, gives us a lighter axis with less material and high strength.

One thing that we wanted to do with our design was to expand it more. We only made one part of the boat trailer. This gives an incomplete design of the boat trailer. Designing completely the boat trailer would have been a very fulfilling and challenging task, because we would have completed a mechanism. This was not possible due to time constraints and our limited knowledge in the field. We could have made a
better design if we have taken all of our design courses, maybe in the future we will go back to this project and finish it.

Overall we are very pleased with our design. We design the best main axis that we could. We do not think there is any problem, excluding our assumption, with our design. We created a part that could be completely functional with an existing boat trailer.
Conclusions

In this design project we were able to create and analyze a part. We choose the main axis of a boat’s trailer. This caught our attention, because we have to analyze the forces that act on the axis (the boat weight, the material, etc.), we have to choose a material that is strong against corrosion. Also we had to choose a material that will not fail under fatigue, which will enable less waste to the environment. We had to create a design that will benefit society.

We started by first analyzing which part we were going to analyze. Once we chose the part, we were able to apply what we have learned from the design courses. This design project is very complete because we have to select the materials, calculate the forces and stresses that act upon it. This project helps us familiarize ourselves with the equations that were given in the classroom. It also helps us become more familiar with computer programs, such as Excel, that enable us easier calculations. We saw that the original design that you have might not work for the purposes intended and this causes a redesigning of the part till it completely satisfies the requirements.

This project helped us apply what we have learned in the Machine Component Design I. This design project helps us see that everything we have done in the course is connected. It also shows us that these are the methods and equations used in real life to solve design problems. This project gave us an idea of what a design engineer does. It is very fulfilling to see that a part that you have worked on can be produced easily with all the specifications that this design project has. This project makes us feel like we already are engineers solving problems that will benefit all of us.
References

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<th>D</th>
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<td>24</td>
<td></td>
<td></td>
<td>( \tau_{amplitude, s, r} ) (ksi)</td>
<td>0.17015</td>
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<td>25</td>
<td></td>
<td></td>
<td>( \tau_{mean, s, r} ) (ksi)</td>
<td>0.19263</td>
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<td>26</td>
<td></td>
<td></td>
<td>( \sigma_{amplitude, shear} ) (psi)</td>
<td>125</td>
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<td>27</td>
<td></td>
<td></td>
<td>( \sigma_{mean, shear} ) (psi)</td>
<td>0.30560</td>
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<td>28</td>
<td></td>
<td></td>
<td>( \tau_{amplitude, shear} ) (ksi)</td>
<td>0.33365</td>
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</table>

N 2.6477
N 23.9186
In this spreadsheet we see that all of the stresses calculated earlier are calculated automatically in here. The values that we change are the ones in pink. These values are the dimensions of the main axis. For the selection of our new dimension, we took into consideration the values in yellow. These values were the weight of the boat and the dynamic safety factor for Point A (our failure zone).