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Section 201
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MAE 3250: Project 2 Report

Part 1:

Boundary Conditions

The boundary conditions applied to the crank are the displacements on the surface of hole A are zero.

Mesh Refinement Study

Refinement	Number of Elements	Deflection at C	Maximum Effective Stress
1	734	1.6982 mm	160.78 MPa
2	1263	1.6982 mm	160.9 MPa
3	2220	1.6983 mm	160.35 MPa

While using the default element size, the mesh refinement generated more precise results for stresses along curved edges, where the stress concentrations occur. I refined the meshes along every curved surface on the crank.

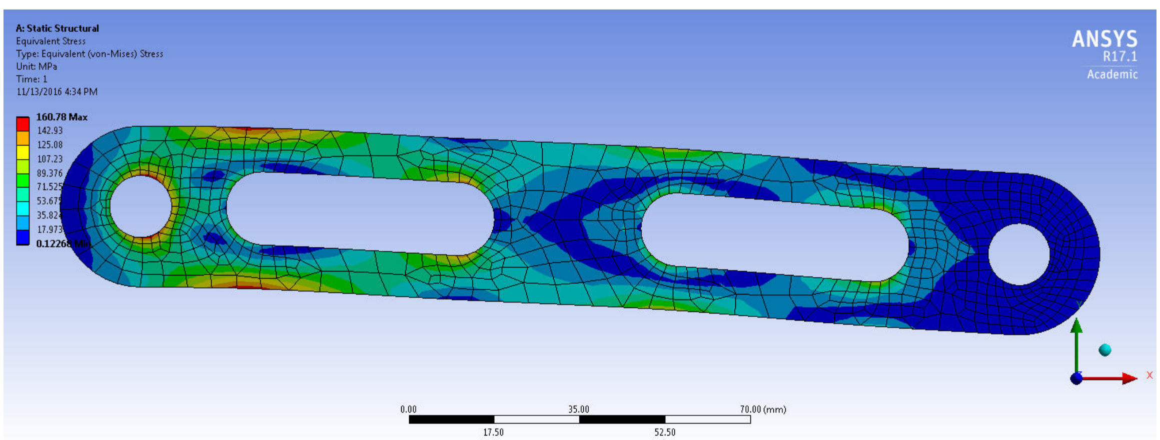

Figure 1: Von Mises Effective Stress for Mesh Refinement of 1

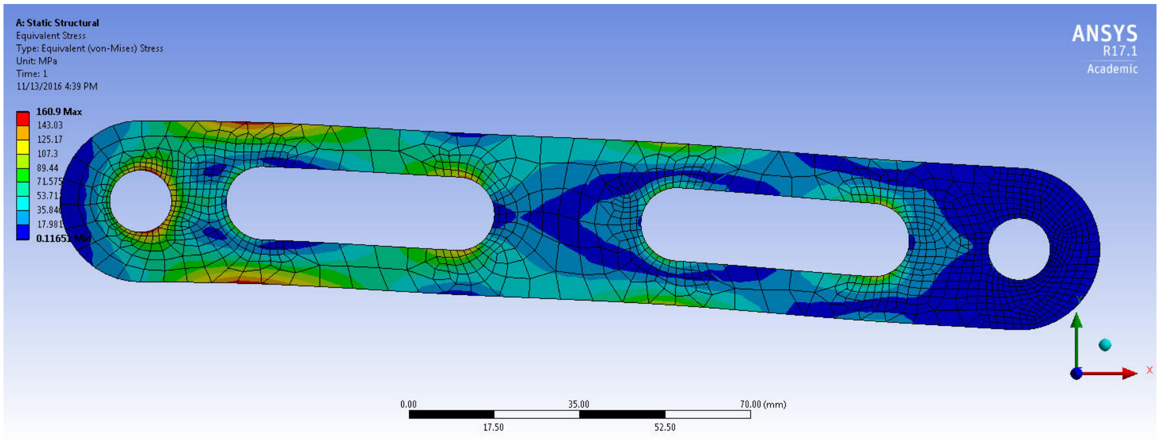

Figure 2: Von Mises Effective Stress for Mesh Refinement of 2

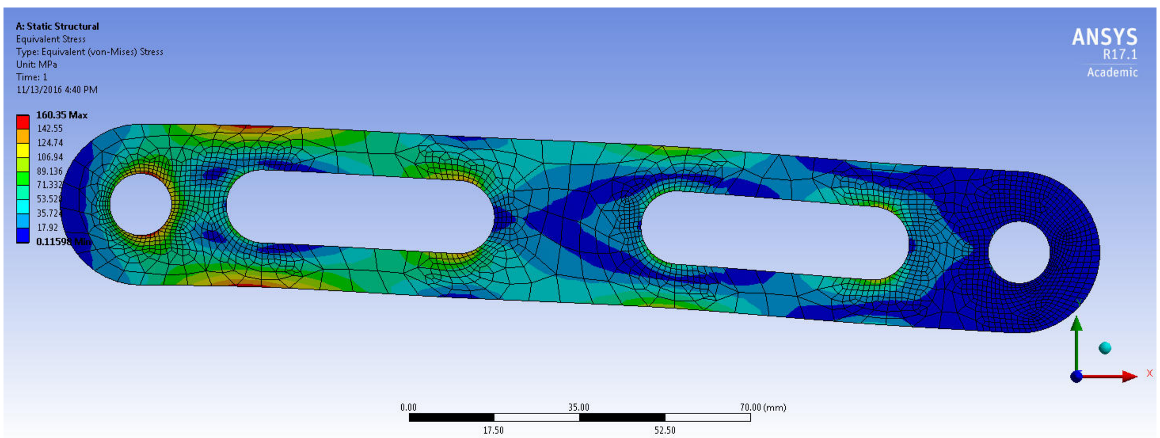

Figure 3: Von Mises Effective Stress for Mesh Refinement of 3

Analysis

I used a mesh refinement of 3 for the analysis because it satisfied the essential boundary condition and made the results more precise. My deflections were off by a factor of 10^3 because I accidentally set the Young's Modulus in MPa instead of GPa.

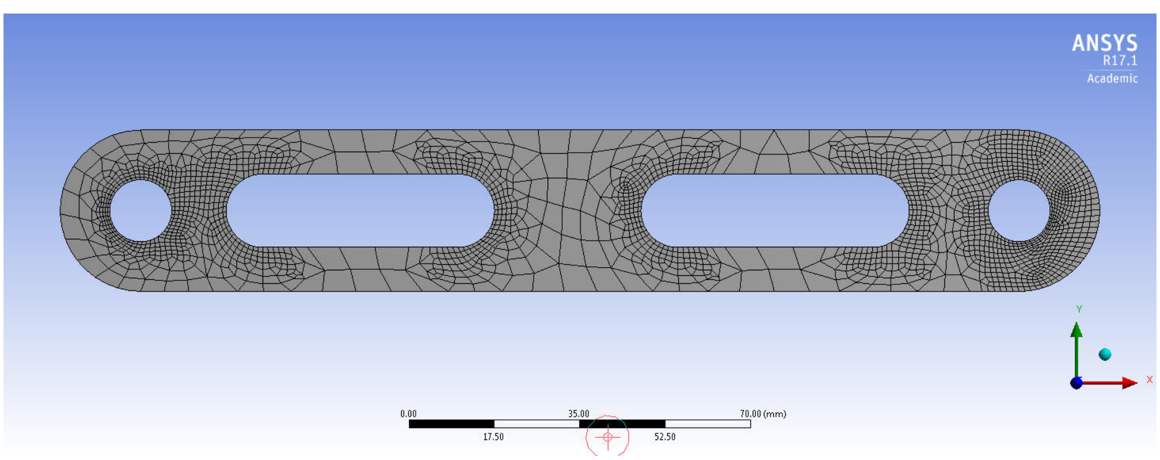

Figure 4: Mesh with Refinement of 3 on Curved Surfaces

Using the mesh show above in Figure 4, I was able to plot the normal stresses in the horizontal and vertical directions, the planar shear stress, all three principal stresses, and the von Mises effective stress already shown in Figure 3.

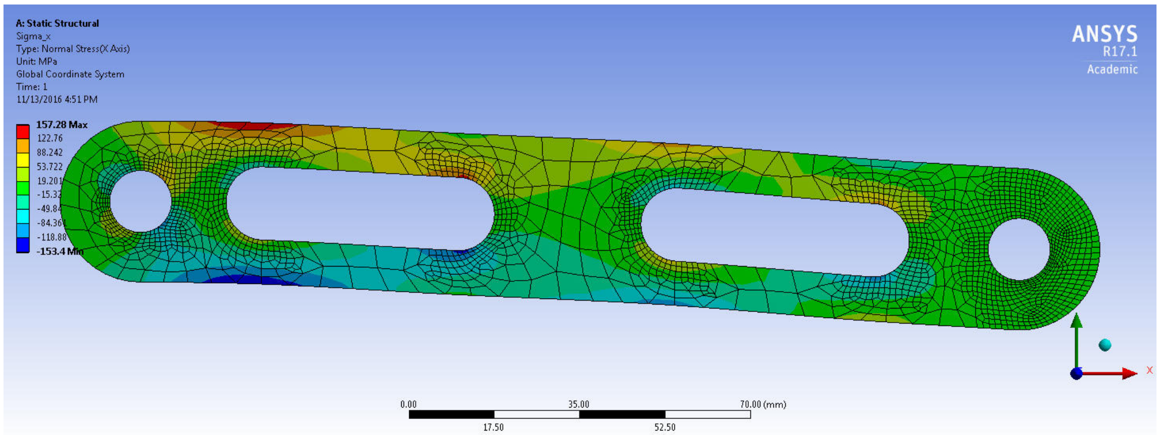

Figure 5: Plot of σ_x along the crank

The horizontal stress is greatest along the left part of the upper edge of the crank and least at the same horizontal position of the lower edge. The maximum value is 157.4 MPa, while the minimum is -153.3 MPa. This makes sense because the crank would be in tension along its upper edge and compression along its lower edge. The horizontal positions of the stress concentrations are just to the right of the fixed hole and far from the applied downward load. There are also stress concentrations along some of the curved surfaces of the extruded ovals.

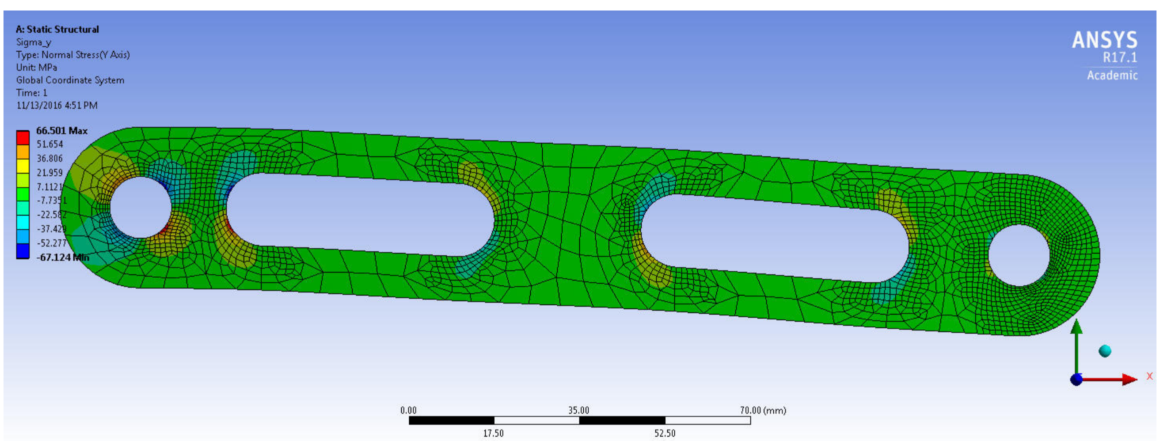

Figure 6: Plot of σ_y along the crank

The vertical stress component of the crank is uniform throughout except for at locations along the inner surfaces of the holes and extruded ovals. This makes sense because we would expect there to be stress concentrations along these surfaces. The stress concentrations occur in the corners of the circular surfaces approximately 45 degrees from the horizontal and vertical. The maximum is 66.523 MPa, and the minimum is -67.122 MPa.

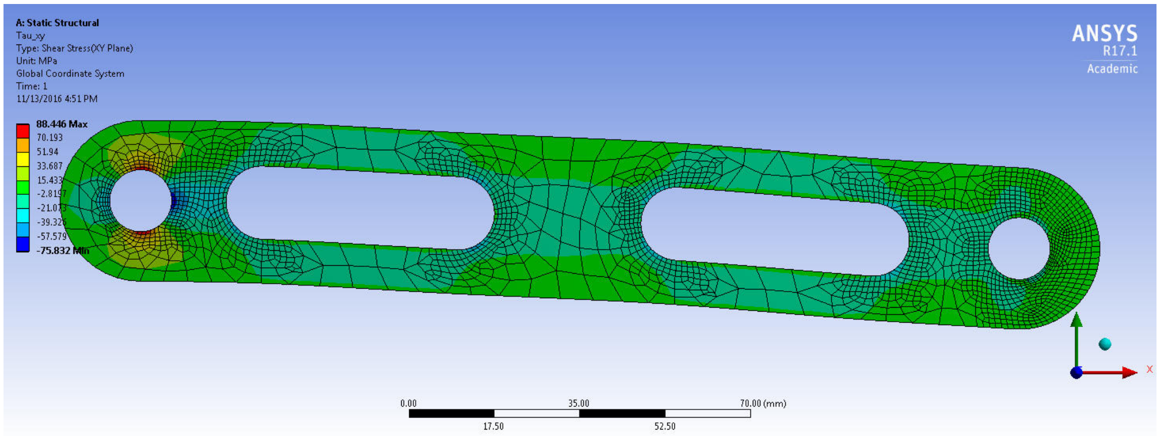

Figure 7: Plot of τ_{xy} along the crank

The shear stress along the crank is uniform except for stress concentrations along the surface of the fixed support. The stress concentrations occur correspond horizontal and vertical axes, and the stress is positive on the vertical surface and negative on the horizontal surface. The maximum is 88.455 MPa, and the minimum is -75.83 MPa.

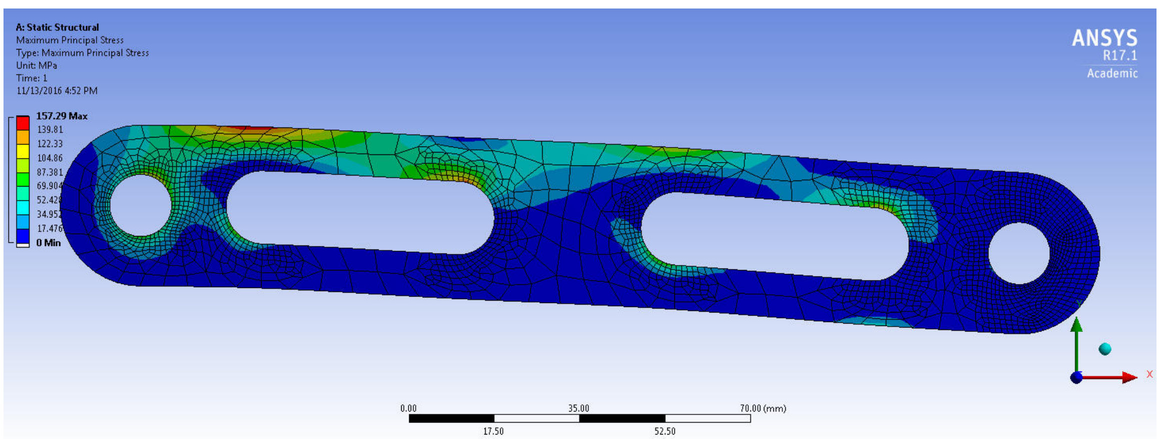

Figure 8: Plot of σ_1 along the Crank

The maximum principal stress along the crank is at a minimum throughout the lower part and right end of the crank. The location of the maximum coincides with the location of the maximum stress in the horizontal direction, where the bar is in tension. The maximum of the maximum principal stress is 157.7 MPa.

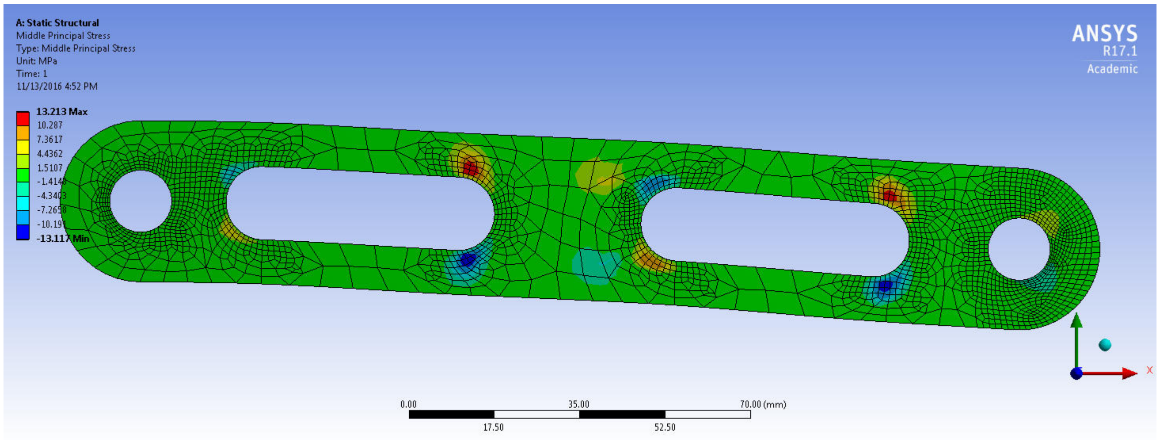

Figure 9: Plot of σ_2 along the crank

The middle principal stress is uniform throughout the bar except for at the stress concentrations along the curved surfaces of the extruded ovals.

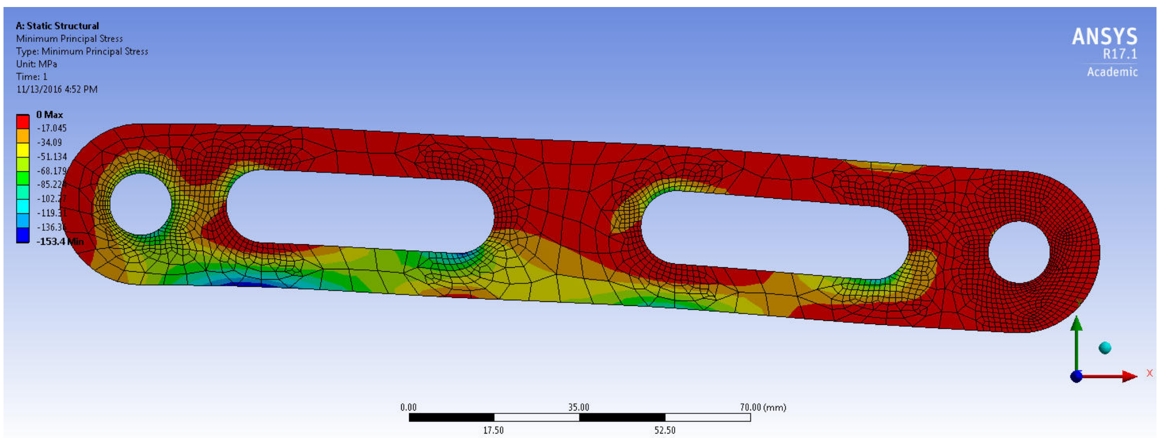

Figure 10: Plot of σ_3 along the crank

The minimum principal stress is zero along the upper half and right end of the crank. The value fluctuates along the bottom half of the crank and at the stress concentrations on the curved surfaces. The minimum stress occurs on the bottom left edge that is in compression at a stress of -153.3 MPa.

Significant Values

ANSYS reported the mass of the crank to be 0.13469 kg. As stated above, the deflection at point C is 1.6983 mm, and the maximum effective stress is 160.35 MPa.

If the tensile yield strength of Al 6061-T6 is 276 MPa, the safety factor in yielding based on the von Mises stress criterion is 1.72, which is less than the design constraint of 2.

Fatigue Life

In order to get the fatigue life of the crank, I needed to get the number of cycles first. The Soderberg relationship allows one to solve for the value of σ_{ar}.

$$\frac{\sigma_a}{\sigma_{ar}} + \frac{\sigma_m}{\sigma_o} = 1$$

$$where \quad \sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min}), \sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{\min})$$

Since the crank alternates experiencing loads of 1 kN downward and 0.5 kN upward, I found the values of σ_{max} and σ_{min} by finding the maximum of the absolute values of the maximum and minimum stresses for each applied load. Using the stress values for the von Mises equivalent stress analysis, the 1 kN downward static loading yielded absolute values of 160.35 MPa for the maximum and 0.10875 MPa for the minimum. For the 0.5 kN upward applied load, the absolute values were 80.174 MPa for the maximum and 0.054377 MPa for the minimum.

Therefore, using σ_{max} = 160.35 MPa and σ_{min} = 0.054377, σ_m = 80.20 MPa and σ_{ar} = 80.15 MPa.

Using the Soderberg relationship, σ_{ar} = 112.98 MPa.

ANSYS gives the maximum equivalent alternating stress as 120.26 MPa and the minimum life as 59,263 cycles, which doe not meet the design constraint of a minimum of 10^7 cycles.

<u>Parts 2-3:</u>

The process of material selection is contingent upon minimizing mass while satisfying the design constraints. The material selection should maximize the safety factor in yielding, maximize the fatigue life in cycles, and minimize the deflection at point C. After deriving these respective material indices, I used CES charts to maximize three expressions: $\frac{\sqrt{\sigma_o}}{\rho}, \frac{\sqrt{Fatigue\ Strength}}{\rho}, and\ \frac{E^{1/3}}{\rho}$.

Without using these criteria for my selection process, I selected commercial purity Magnesium ASTM 9980A. After analyzing the safety factor in life for different designs with this material, I realized I did not select an optimal material. I then decided to select a different Magnesium alloy that maximized all the above criteria using the CES software, namely Magnesium EA55RS.

After experimenting with the adjustable dimensions, I found a design that reduced weight and met the design constraints. With H1 set to 17.5 mm, H2 set to 20 mm, and L2 kept at 40 mm, the crank has a safety factor in yielding of 2.03, a safety factor in life of 1.215 which means the fatigue life is greater than 1e7 cycles, and a maximum deflection of 3.2575 mm which is less than twice the base case of 1.69 mm. For this material, the safety factor in yielding which is contingent on the maximum von Mises effective stress is the limiting factor for increasing the adjustable dimensions. The weight of the crank is 87.1 grams with these dimensions.

For the automated design process, I set H1, H2, and L2 as input parameters and surface body mass, maximum effective stress, and maximum total deflection as output parameters. After setting the range for the input parameters, I generated the design of experiments values and then set my goals for the optimization. My optimization minimized mass, maximized effective stress to an upper bound of 205 MPa, and maximized the deflection to an upper bound of twice the base case.

Optimization Method	H1	H2	L2
Manual	17.5 mm	20 mm	40 mm
Automated	18.06 mm	17.504 mm	53.673 mm

Optimization Method	Crank Mass	Deflection at Point C	Maximum Effective Stress	Safety Factor Against Yielding	Fatigue Life
Manual	87.1 g	3.2575 mm	201.89 MPa	2.03	1.215e7 cycles
Automated	83.124 g	3.1243 mm	204 MPa	2.01	1.438e7 cycles