**Topology Optimization of Bracket with Fatigue**

**Overall Objective:**

The principal objective of this engineering report is to design a bracket with a defined fatigue life and maximum allowable deformation.

**Assumptions**

1. The bracket will be manufactured using conventional processes such as forging, casting, or machining, rather than 3D printing.

2. Stresses where the support plate is fixed will be disregarded in the analysis.

3. The size and shape of the cylindrical surface will remain unchanged.

4. The center of mass of the support plate and its thickness of 25 mm will not be altered. Therefore, adjustments to the length (L) and width (W) will be made while preserving the plate's center of mass.

5. Semi-Log interpolation will be utilized for the S-N data.

6. The applied load is entirely reversible, and mean stress effects will not be factored into the damage calculation.

**Geometry**

We design a bracket that satisfies the requirements, with a support plate of length L, width W, and a fixed thickness of 25 mm. Also, cylinder is kept constant.

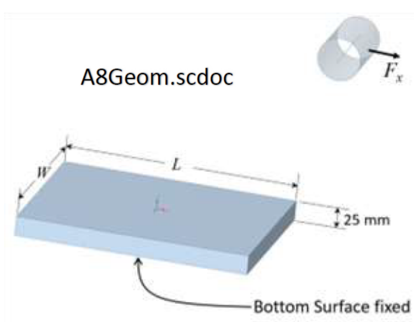



Figure 1

**Material Data**

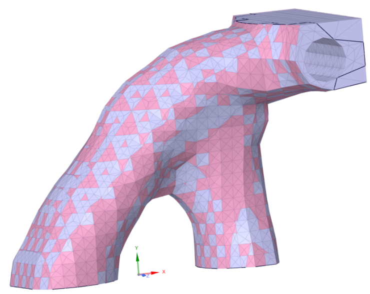
For the A6 aluminum alloy, the following material properties will be utilized: Young's modulus (E): 71 GPa, Poisson's ratio (ν): 0.31, Density (ρ): 2700 kg/m³. The provided S-N fatigue strength versus life cycles data is as follows: 105 MPa at 1000 cycles, 68.2 MPa at 1600 cycles, 42.2 MPa at 6500 cycles, 21.4 MPa at 500,000 cycles, 17.2 MPa at 1,800,000 cycles, and 14.1 MPa at 10,700,000 cycles. The material properties have been incorporated into the Engineering Data of Workbench, and relevant information is presented in the appendix.

**Boundary Conditions**

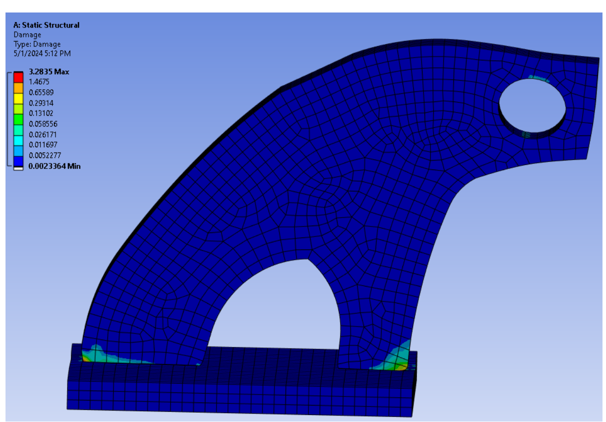
The bottom face of the plate support is fixed, and a load of 26,000 N is applied to the bracket via the cylinder.

**Results**

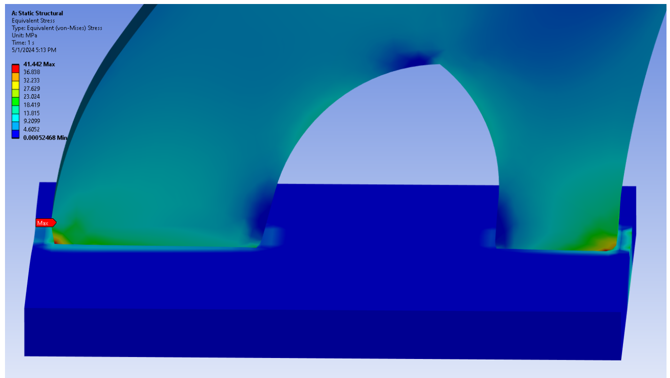
After defining the loads and encapsulating the plate and cylinder within a general cubic structure, we proceed to optimize the cubic, tasking it with reducing the weight by 25%. The obtained result is as follows:



Subsequently, we replicate a part resembling this shape, prioritizing manufacturability by avoiding intricate geometries. Hence, we acquire our initial design (as shown below) and subject it to fatigue analysis.

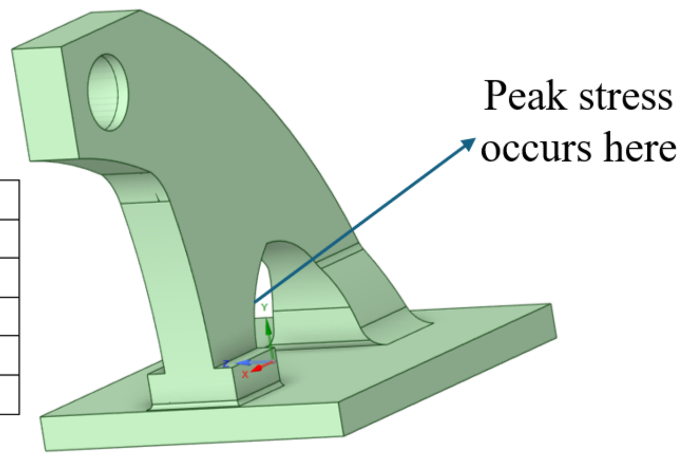



Upon detecting damage exceeding unity, we proceed to pinpoint areas of elevated stress. Through mesh refinement, we observe a rise in stress levels, potentially indicating the presence of singularities, as shown below:



Next, let's modify the geometry slightly by adding additional material to the region experiencing maximum stress. By doing so, we get our finalized geometry as below:

| | |
|---|---|
| Mass (kg) | 5.71 |
| L (mm) | 300 |
| W (mm) | 250 |
| Peak equivalent stress (MPa) | 35.48 |
| Fatigue Damage | 0.94 |
| Max Deformation (mm) | 0.23 |

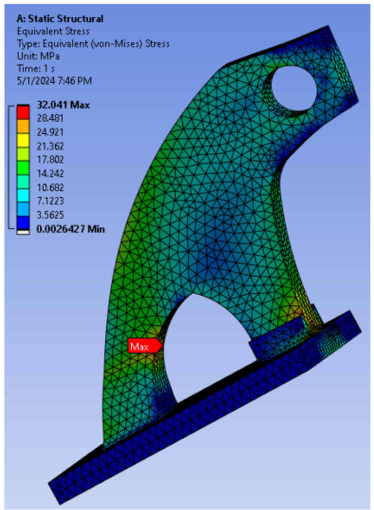


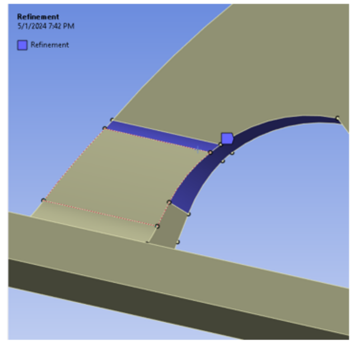

My final design

Please note that the mass of plate has been excluded in the mass reported in the above table. To make sure that our analysis is correct, we conduct the mesh convergence study.

**Mesh convergence study**

Given the focus on fatigue analysis, ensuring the accuracy of peak stress magnitude is important. Consequently, a mesh convergence study has been conducted to verify compliance with the design specifications. The figure below illustrates the Von Mises stress distribution across the designed bracket using a 10 mm mesh (left), accompanied by the regions where refinement occurs (right).






The results of the Von Mises stress after mesh refinement #1 and #2 are shown in appendix. Below table shows the mesh convergence study table.

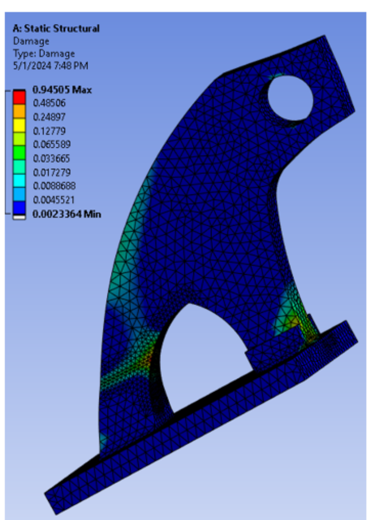
| Mesh size | 10 mm | 10 mm + refinement#1 | 10 mm + refinement#2 |
|---|---|---|---|
| Peak Von-Mises Stress (MPa) | 32.04 | 35.48 | 35.95 |

The analysis reveals that the variation in maximum stress between refinement #1 and refinement #2 is less than 1.5%. Hence, it is valid to infer that convergence has been achieved with the 10mm + refinement #1 mesh, and this mesh will be used for the rest of analysis.

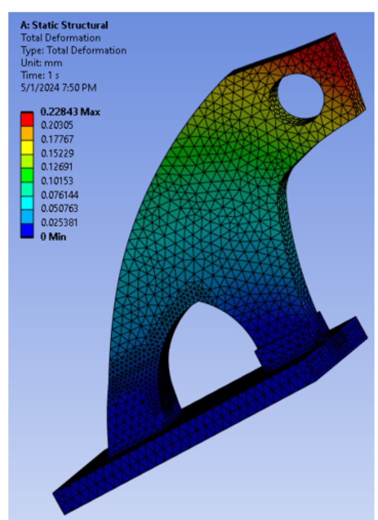
**Checking for maximum deformation and damage under fatigue:**

The fatigue specifications have been inputted into the "Fatigue Tool," as outlined in the appendix. The figure below illustrates the values of fatigue damage under 25,000 cycles of loading (left) and total deformation (right).

Fatigue Damage

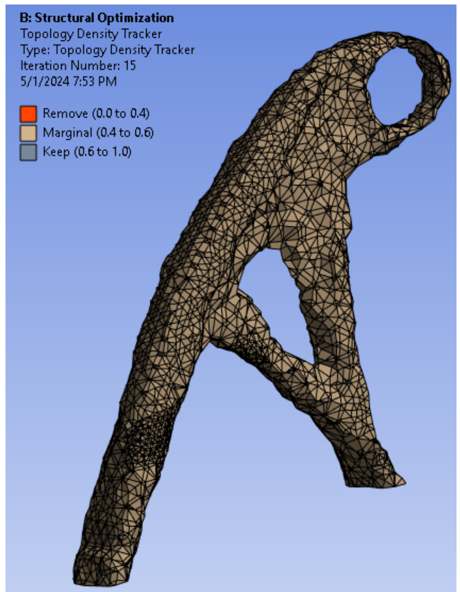



Total Deformation



Examining the figure above reveals a maximum damage of 0.94, indicating that fatigue damage is unlikely to occur. Additionally, the maximum deformation is 0.23 mm, well below the 1 mm threshold for maximum deformation.

Upon re-running the optimization process, the resulting shape is as follows:




Given that we are currently meeting the design specifications and the mass is sufficiently low, further optimization is deemed unnecessary at this stage.

**Conclusion:**

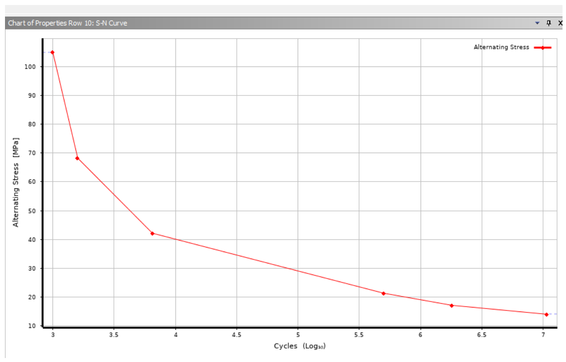
In summary, the iterative design process aimed at minimizing mass and fatigue damage resulted in an optimal bracket design. Utilizing A6 aluminum alloy, strategic geometric modifications, and mesh refinements, the design met requirements without compromising performance or safety. Further optimization was deemed unnecessary, affirming the effectiveness of engineering principles in achieving optimal solutions.

## Appendix:

Material data in workbench:

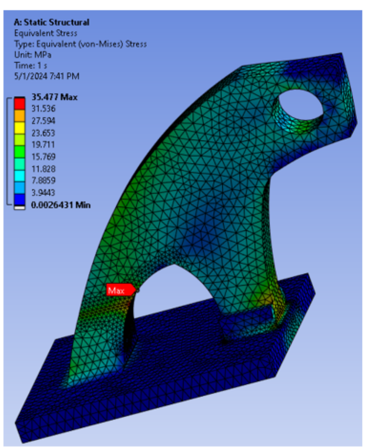
Properties of Outline Row 3: A6 Alumnium Alloy

| | A | B | C | D | E |
|---|---|---|---|---|---|
| 1 | Property | Value | Unit | | |
| 2 | Material Field Variables | Table | | | |
| 3 | Density | 2700 | kg m^-3 | | |
| 4 | Isotropic Elasticity | | | | |
| 5 | Derive from | Young's Modulus and Poisson's Ratio | | | |
| 6 | Young's Modulus | 71 | GPa | | |
| 7 | Poisson's Ratio | 0.31 | | | |
| 8 | Bulk Modulus | 6.2281E+10 | Pa | | |
| 9 | Shear Modulus | 2.7099E+10 | Pa | | |
| 10 | S-N Curve | Tabular | | | |
| 11 | Interpolation | Semi-Log | | | |
| 12 | Scale | 1 | | | |
| 13 | Offset | 0 | MPa | | |

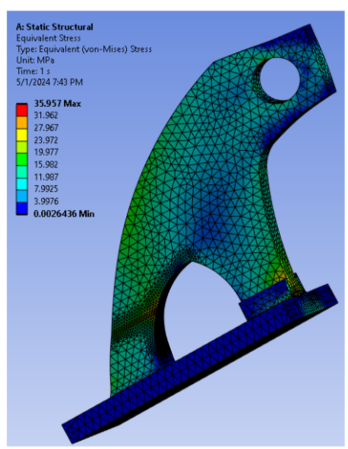



The results of the Von Mises stress after mesh refinement #1 and #2 are:

Mesh Refinement #1




Mesh Refinement #2




Analysis Settings in Fatigue Tool:

Details of "Fatigue Tool"

| | |
|---|---|
| **Domain** | |
| Domain Type | Time |
| **Materials** | |
| Fatigue Strength Factor (Kf) | 1. |
| **Loading** | |
| Type | Fully Reversed |
| Scale Factor | 1. |
| **Definition** | |
| Display Time | End Time |
| **Options** | |
| Analysis Type | Stress Life |
| Mean Stress Theory | None |
| Stress Component | Equivalent (von-Mises) |
| **Life Units** | |
| Units Name | cycles |
| 1 cycle is equal to | 1. cycles |