University of Texas at El Paso

Mechanical Engineering

Final Project: Baja Buckle

MECH 3334: Mechanical Design

CRN 10913

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Table of Material Properties

Table 1: Material Properties of Titanium Alloy Ti-6Al-4V (Grade 5), Annealed

Property	Value (Metric)	Notes
Density	4.43 g/cc	
Tensile Strength, Ultimate	950 MPa	
Tensile Strength, Yield	880 MPa	
Compressive Yield Strength	970 MPa	
Modulus of Elasticity (E)	113.8 GPa	
Poisson's Ratio	0.342	
Shear Modulus (G)	44 GPa	
Shear Strength (Ultimate)	550 MPa	
Hardness (Rockwell C)	36 HRC	
Ultimate Bearing Strength	1860 MPa	
Bearing Yield Strength	1480 MPa	
Elongation at Break	14%	
Reduction of Area	36%	
Charpy Impact (V-notch)	17 J	
Fatigue Strength (Notched)	240 MPa	at 1×10^7 cycles
Fatigue Strength (Unnotched)	510 MPa	10,000,000 cycles

Source: MatWeb Titanium Ti-6Al-4V Grade 5 Datasheet.

Handwork: Stress Analysis and Factor of Safety

Finite Element Analysis

This section presents the finite element analysis (FEA) conducted on the buckle design using Autodesk Fusion 360. The objective of the simulation was to evaluate how the titanium buckle responds to the applied loading conditions and to verify that the design meets the safety factor requirement of 3.0. The analysis included the definition of loads and constraints, mesh generation, and the evaluation of stress, displacement, and safety factor results.

Load Application

A static load representing the tension from the connected strap was applied directly to the inner face of the slot. The magnitude of the load was set to 5000 N, acting in the vertical direction. Figure 1 shows the boundary where the load was applied.

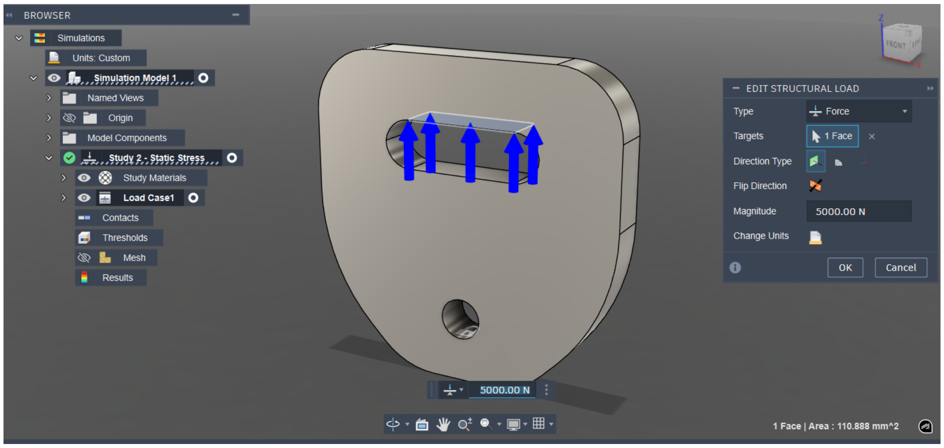


Figure 1: Applied load of 5000 N on the slot interface.

Boundary Conditions

The circular bottom hole was fully constrained to replicate the pin connection that anchors the buckle to the supporting structure. This fixed constraint prevents all translational and rotational degrees of freedom, ensuring that the model behaves realistically during loading.

Mesh Generation

A tetrahedral mesh was automatically generated using Fusion 360's adaptive meshing at an element size of 1%. The mesh achieved 96,937 nodes and 64,831 elements, providing high resolution around critical stress concentration regions such as the slot fillets and lower connection area. Figure 2 shows the final mesh used in the simulation.

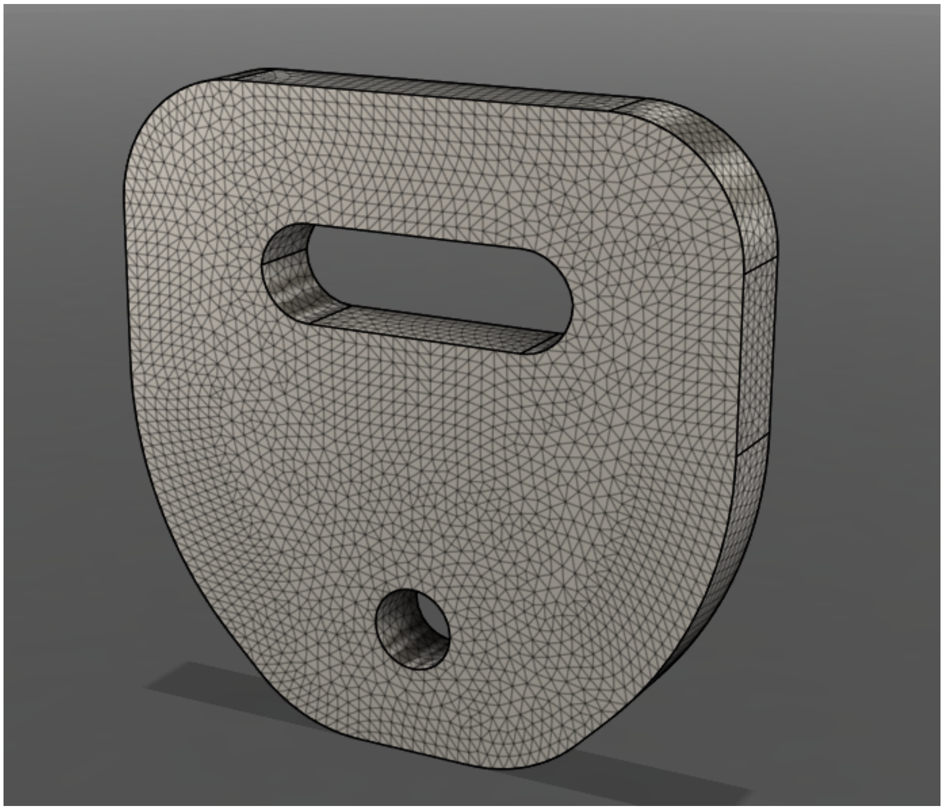


Figure 2: Final tetrahedral mesh used in the simulation.

Safety Factor Results

The safety factor was evaluated based on the yield strength of Ti–6Al–4V (Grade 5). The minimum safety factor observed was 3.51, which exceeds the required minimum safety factor of 3.0. This confirms that the buckle design is structurally sufficient for the applied 5000 N load. Figure 3 illustrates the safety factor distribution across the model.

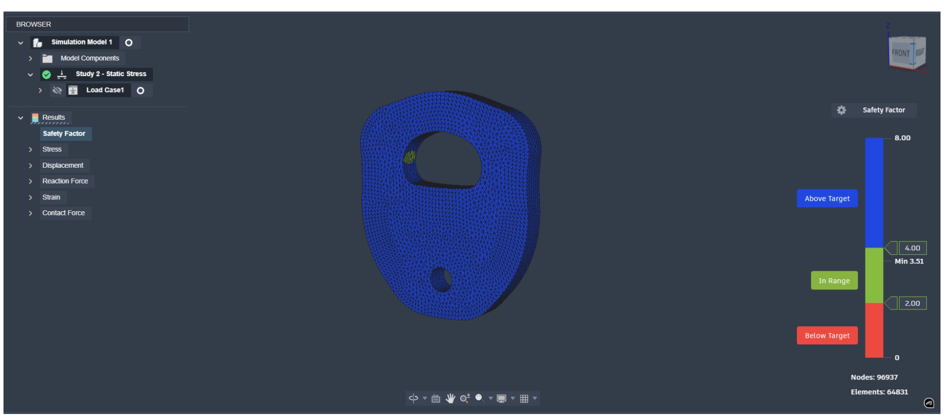


Figure 3: Safety factor plot showing a minimum safety factor of 3.51.

Stress Distribution

Von Mises stress was used as the failure criterion. The maximum stress occurred around the inside edges of the slot, where bending and tension effects combine. The peak stress recorded was approximately 251 MPa, which is below the Ti–6Al–4V yield strength of 880 MPa. The stress plot is shown in Figure 4.

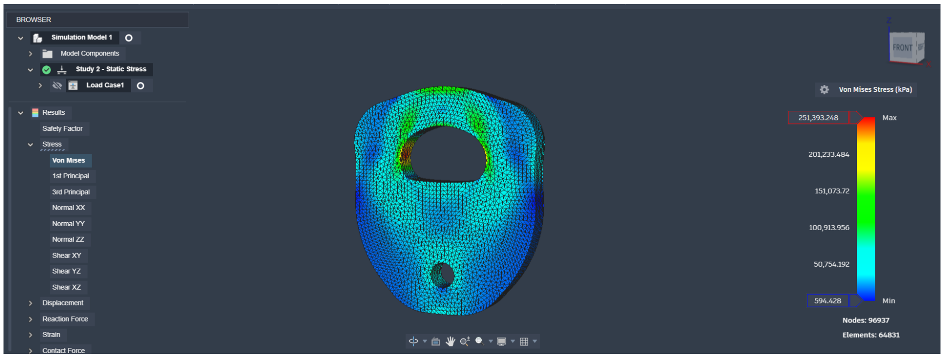


Figure 4: Von Mises stress distribution, showing peak stress concentrations near the slot.

Displacement Results

The total deformation under the 5000 N load was small, with a maximum displacement of approximately 0.043 mm. This amount of deflection is negligible relative to the overall part size and does not compromise performance. Figure 5 displays the displacement field.

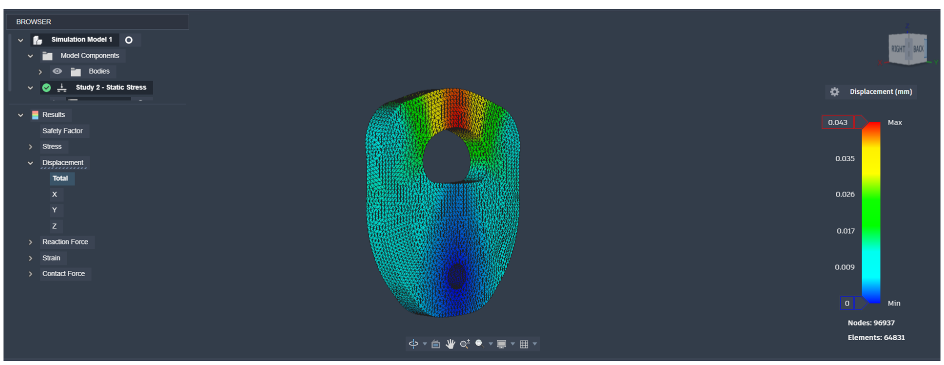


Figure 5: Total displacement results showing a maximum deformation of 0.043 mm.

Summary of FEA Findings

The results of the FEA confirm that the buckle design meets all safety and performance requirements.

- Minimum safety factor: 3.51 ($>$ 3.0 requirement)
- Maximum von Mises stress: 251 MPa (« 880 MPa yield)

- Maximum displacement: 0.043 mm (negligible)

These results validate that the Ti–6Al–4V buckle component is structurally adequate for the expected loading conditions.

Blueprint and 3D Print

The buckle was fully modeled in Autodesk Fusion 360 and exported as a set of engineering drawings. The blueprint includes all relevant dimensions required to manufacture the part, such as overall width and height, slot length, slot width, material thickness, fillet radii, and pin-hole diameter. These dimensions ensure that the geometry satisfies the project requirement of a one-inch strap slot and allows the part to be inspected or machined accurately.

Figure 6 shows the 2D drawing view generated from the CAD model.

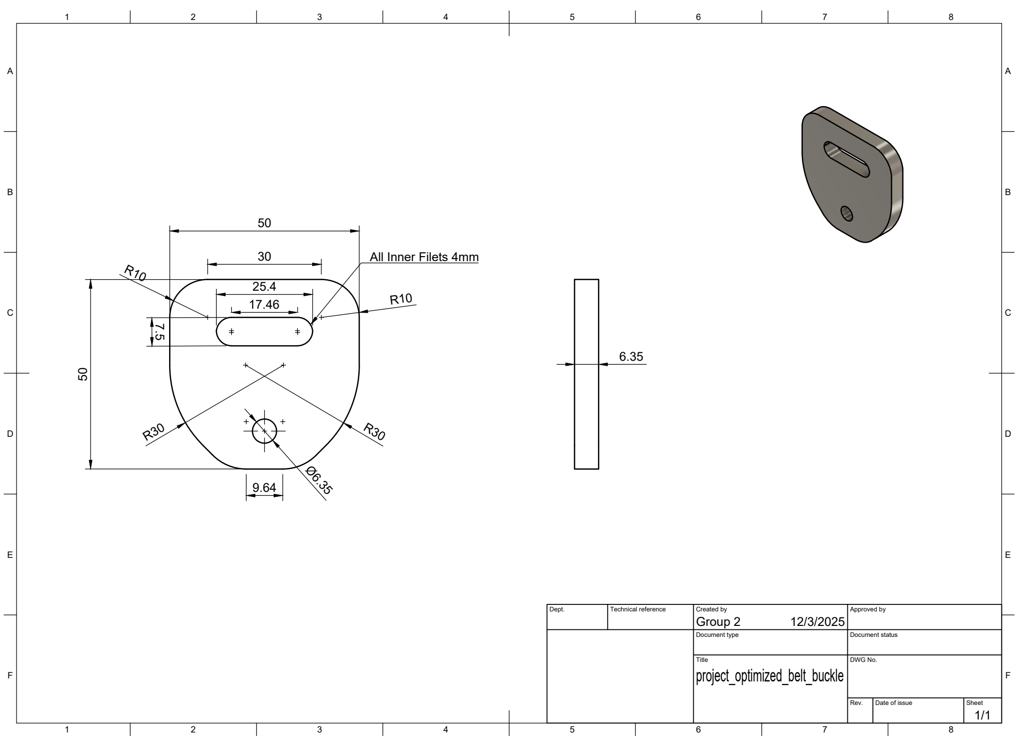


Figure 6: Front-view engineering drawing exported from Fusion 360.

A 3D printed prototype of the buckle was also produced to verify the geometry and ensure that the one-inch strap fits properly. The model was printed using FDM printing with a 0.2 mm layer height. Figure 7 shows the printed prototype.

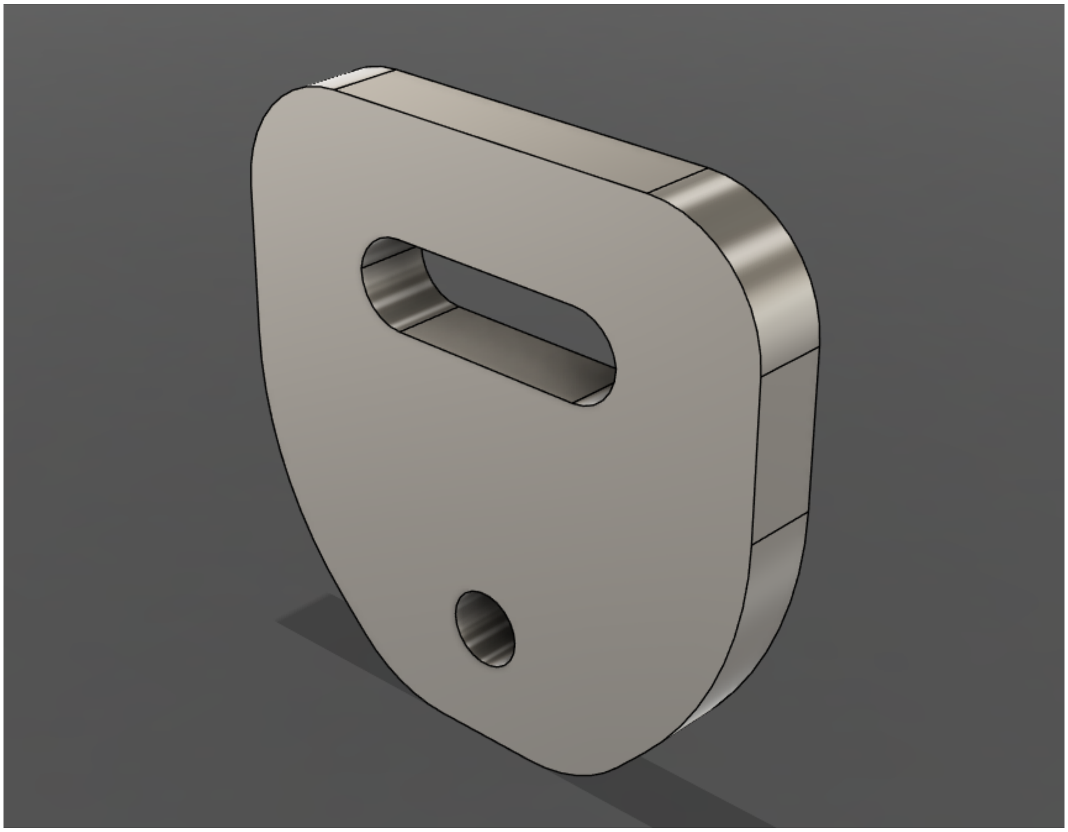
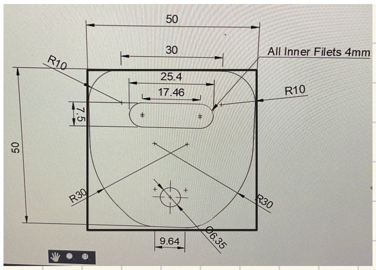


Figure 7: 3D printed prototype used for dimensional verification.

Handwork: Stress Analysis

This section contains the manual stress analysis performed for the buckle design using two critical locations: the slot fillet and the bottom pin hole. All calculations were completed by hand, and the derivations are shown in Figure 8.

The calculations use the material properties of Ti–6Al–4V (Grade 5), including a yield strength of 880 MPa. A load of 5000 N was applied at the strap slot to determine the axial, bending, and bearing stresses at the chosen locations.



Design 2

$A = 50 \cdot 50\ mm - A_{slot} - A_{circle} = 2500 - 175.13 - 31.67$

$A_{circle} = \pi\left(\frac{6.35}{2}\right)^2 = 31.67\ mm^2$ $\quad A_{box} = 50 \cdot 50 = 2500$ $\quad A_{slot} = \pi\left(\frac{7.5}{2}\right)^2 + 7.5 \cdot 17.46 = 175.13\ mm^2$

$\sigma_{allowed} = 92\ MPa$

$\sigma_{Design\ 2} = \frac{F}{A} = \frac{5000}{(2500 - 175.13 - 31.67)} = 2.18\ MPa$

① σ_y for Ti-6AI-4V $= 880\ MPa$

$t = 6.35\ mm$

$b = 25.4\ mm$

$A_{slot} = 175.13\ mm^2$

$\sigma_{slot} = \frac{F}{A} = \frac{5000}{175.13} = 28.55\ MPa$

② Bending Stress at Slot Fillet

$c = 17.46/2 = 8.73$

$M_{O/C} = F \cdot c = 5000 \cdot 8.73 = 43650\ N \cdot mm$

$I = \frac{bt^3}{12} = \frac{25.4(6.35^3)}{12} = 541.9\ mm^4$

③ Bending Stress $\rightarrow \sigma_B = Mc/I \rightarrow \sigma_B = 43{,}650(8.73)/541.9\ mm^4 = 703.2\ MPa$

④ Total stress

$\sigma_{total} = \sigma + \sigma_b = 28.55 + 703.2 = 731.75\ MPa$

⑤ $FOS_A = \sigma_y/\sigma_{total} = 880/703.2 = 1.2514$

⑥ Bottom Pin Calculations

$d = 6.35\ mm \quad A = \pi r^2 = \pi\left(\frac{d}{2}\right)^2 = 31.67\ mm^2$

$Stress = \sigma_{pin} = 5000/31.67 = 157.88\ MPa$

$FOS_{pin} = 880/157.88 = 5.574$

Figure 8: Hand calculations for stress analysis and factors of safety at the slot fillet and pin hole.

From the handwork:

- The slot fillet experiences a combined axial and bending stress.
- The bottom pin hole primarily experiences bearing stress.

- Each location was evaluated using the yield strength criterion.

A summary of the results is included in the next section.

Results

This section summarizes the results obtained from both the hand calculations and the finite element analysis (FEA).

Hand Calculation Results

Using the handwritten calculations shown previously:

- **Slot Fillet (Critical Location)** - Total stress (axial + bending): 731.75 MPa - Factor of Safety: 1.25
- **Pin Hole** - Stress at pin interface: 157.88 MPa - Factor of Safety: 5.57

The hand calculations indicate that the slot fillet is the critical region due to the high bending moment created by the applied load.

FEA Results

The Fusion 360 simulation (shown in Section 4) produced the following key results:

- Maximum von Mises stress: 251 MPa
- Minimum safety factor: 3.51
- Maximum displacement: 0.043 mm

The highest stresses appear around the inner edges of the slot, which matches the location identified in the hand calculations.

Comparison

- Hand calculations give a conservative FOS of 1.25.
- FEA shows a significantly higher minimum FOS of 3.51.
- Both methods confirm the same critical region (slot fillet).
- The difference is expected because FEA accounts for fillets, curvature, and 3D stress distribution.

Conclusion

The buckle design was evaluated through manual hand calculations and finite element analysis to verify its ability to withstand a 5000 N load. The hand calculations identified the slot fillet as the most critical location, giving a conservative factor of safety of 1.25. The pin hole was shown to have a much higher factor of safety of 5.57.

The FEA results demonstrated that the buckle performs significantly better when modeled in three dimensions, with a minimum safety factor of 3.51 and a maximum stress of 251 MPa, which is well below the yield strength of Ti–6Al–4V. The maximum displacement of 0.043 mm indicates that deformation is negligible.

Overall, both hand calculations and FEA confirm that the buckle design is structurally adequate, with the FEA showing a comfortable margin of safety. The design successfully meets the project requirements.