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Overall Objective

The primary goal of this analysis is to evaluate the structural integrity and fatigue life of a trailer hitch under various loading conditions. This report outlines the assumptions made, the geometry and material data of the hitch, boundary conditions, mesh convergence study, results, and recommendations for redesign to improve performance.

Assumptions

Thermal effects are not considered.

The analysis is based on linear elasticity.

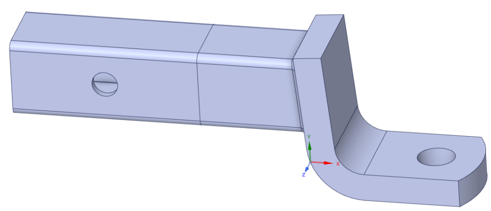
Stress risers near the frictionless support are ignored.

The hitch's constraints are assumed to be frictionless, allowing free sliding, which is an idealization and not reflective of real-world conditions where resistance due to attachment occurs.

Fatigue analysis is governed by the Goodman curve.

Geometry

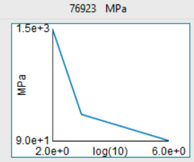
Below figure shows the geometry of the hitch:



Material Data

The trailer hitch is made from a custom material, the properties of which are detailed in the material data section.

CustomeMaterial

Density	7.85e-06 kg/mm³
Structural	
Isotropic Elasticity	
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	2e+05 MPa
Poisson's Ratio	0.3
Bulk Modulus	1.6667e+05 MPa
Shear Modulus	76923 MPa
S-N Curve	
Tensile Ultimate Strength	460 MPa

Boundary Conditions

The trailer hitch is made from a custom material, the properties of which are detailed in the material data section. Load Cases are as below:

LC1: Tongue Load of 2500 N, Drop, infinite design life (1e6 cycles), load applied then removed.

LC2: Tongue Load of 2500 N, Rise, infinite design life (1e6 cycles), load applied then removed.

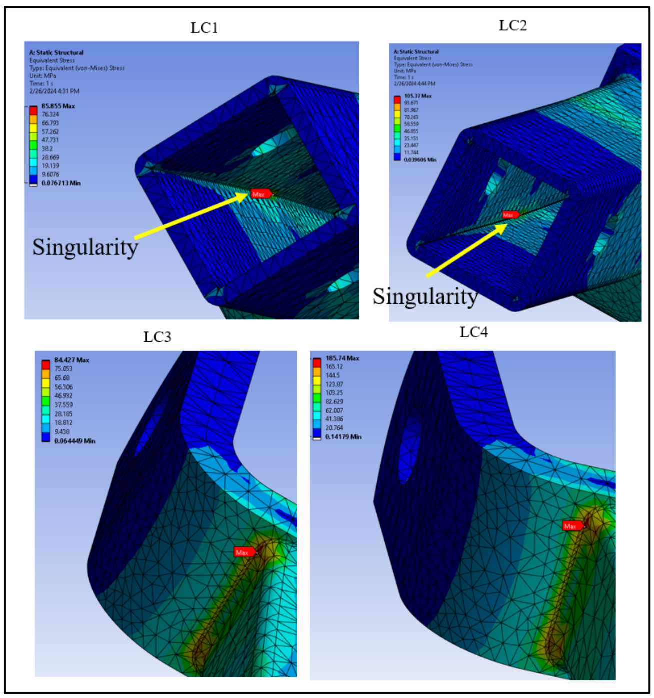
LC3: Trailer Load Small Hill: 5,000 N, design life = 1e5 cycles, load applied and reversed.

LC4: Trailer Load Steep Hill: 11,000 N, design life = 5000 cycles, load applied and reversed.

Details of implementing boundary conditions for each load case is shown in the appendix.

Mesh convergence study:

A mesh size of 5 mm is initially chosen, and below figure shows the Von Mises stress for each load case:



Singularities in LC1 and LC2 are indicated. Singularities can occur due to load points, sharp corners, or sudden changes in constraints and boundary conditions. In this case, we have identified sharp corners inside the receiver as potential sources of singularities. It's important to note that the maximum stress locations in LC3 and LC4 are not singularities because they occur at the smaller fillet, where we anticipate observing critical stress levels.

Furthermore, upon examining the stress plots for LC1 and LC2, we observe high stress concentrations at the small fillet connecting the receiver to the plate, a location similar to those identified in LC3 and LC4 as critical stress areas. To ascertain whether we are encountering a singularity at LC1/LC2 rather than critical stress, we proceed to refine the mesh at the identified singularity locations. This refinement allows us to evaluate how the stress peak changes, providing insight into whether the observed high stress is due to a singularity or is indeed indicative of a critical stress area.

Upon refining the mesh at the singularity location of LC1/LC2, we observe a notable increase in the maximum stress at this location: it escalates from 85 MPa (for a 5 mm mesh without any refinement) to 122 MPa (for the mesh with one level of refinement) and further to 140 MPa (for the mesh with two levels of refinement). This trend of increasing stress highlights the presence of a singularity. Conversely, at the critical stress locations, mesh refinement leads to a convergence of peak stress values (as will be detailed in the next paragraph). This convergence indicates that these areas are indeed critical stress locations, distinguishing them from singularities where stress values continue to escalate with mesh refinement.

For the mesh convergence study, given that the location of critical stress is similar between LC1 and LC2, as well as between LC3 and LC4, and considering that our analysis is based on linear stress principles, it is not necessary to perform mesh convergence studies for all cases. Consequently, we will focus our mesh convergence efforts on LC1 and LC3 only. The results of mesh refinement for LC1 and LC3 are presented in the table below:

Mesh Size	Peak Stress for LC1 (MPa)	Peak Stress for LC3 (MPa)
5 mm	85.855	84.427
5 mm + refinement #1	97.086	103.77
5 mm + refinement #2	98.493	109.12
5 mm + refinement #3	99.249	112.98

Upon examining the maximum stress values, it becomes evident that a 5 mm mesh with two levels of refinement is sufficiently detailed, leading to converged stress values for both LC1 and LC3. This indicates that further refinement does not significantly alter the stress results, suggesting that the mesh is adequately capturing the stress distribution for these load cases. The plots obtained during the mesh convergence study are documented in the appendix for detailed review.

Results

The table below presents the results of the fatigue analysis for all loading cases. Plots of damage, life, safety factor, and fatigue equivalent alternating stress for each load case are reported in the appendix.

Result Case	Peak Stress (MPa)	Life	Damage	Safety Factor	Fatigue Equivalent

					Alternating Stress (MPa)
Load Case 1	98.493	1e6	N/A	1.76	49.247
Load Case 2	98.493	1 e6	N/A	1.5285	55.151
Load Case 3	109.12	6.7 e6	0.149	1.834	109.12
Load Case 4	240.07	43226	0.116	1.429	240.07

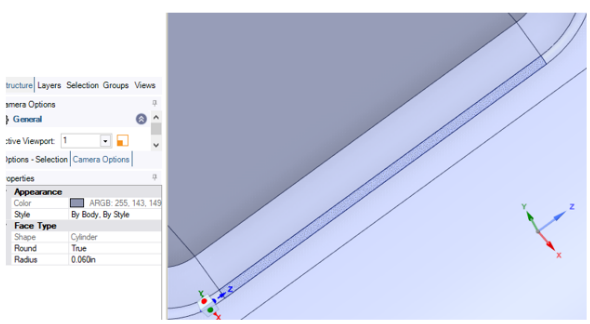
The fatigue analysis results show differences between LC1 and LC2 at the critical location, with LC1 being safer than LC2. This difference arises because, in LC1, more areas are subjected to compression compared to LC2. Fatigue is more susceptible to failure under tensile stress; thus, parts under compression are less likely to experience fatigue.

In conducting the fatigue analysis, the type of loading was selected as either Fully Reversed or Zero-based, depending on the loading conditions specified in the problem statement. The Goodman curve was utilized for the fatigue analysis. Furthermore, to accurately predict the design life for each loading case, the signed von Mises stress was used.

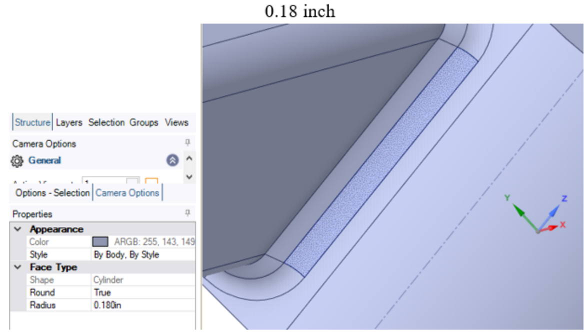
Redesign of the Hitch for Enhanced Geometry:

Considering the safety factors outlined in the table above, all safety factors exceed 1, indicating that none of the scenarios will fail under the life design specified in the problem statement. However, to further enhance the design, we focus on the lowest safety factor, identified in LC4. Our objective is to improve the design to elevate the safety factor. This is achieved by increasing the radius of the small fillet, as illustrated in the subsequent figure.

Original design with small fillet radius of 0.06 inch

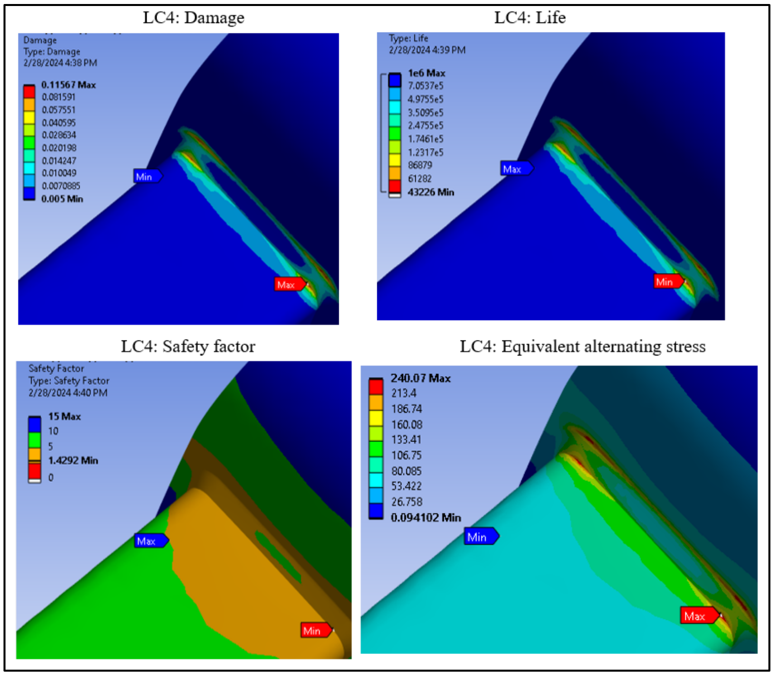


Proposed design with small fillet radius of 0.18 inch

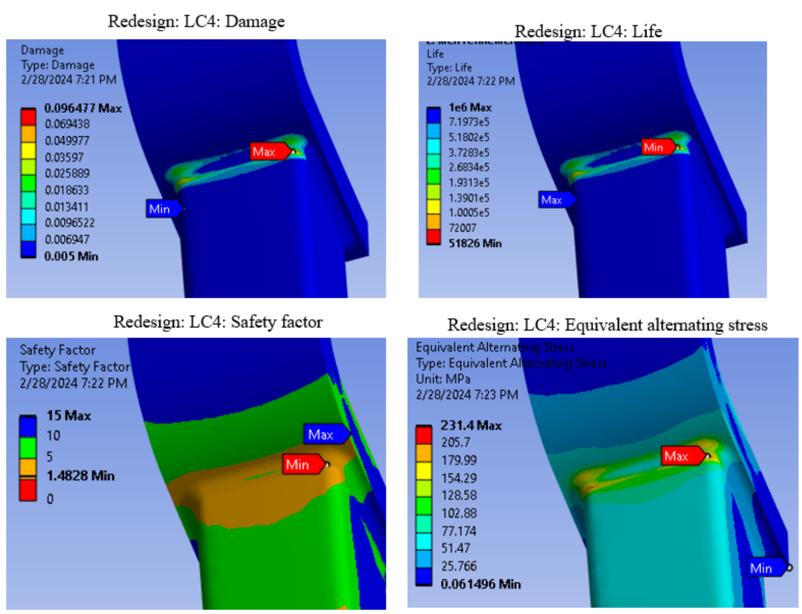


This modification results in a decrease in the peak stress for LC4, from 240.07 MPa to 231.4 MPa. The figures below demonstrates the comparison of fatigue analysis for LC4 between the original and the proposed designs.

Fatigue analysis for LC4 in the original design:



Fatigue analysis for LC4 in the proposed design:

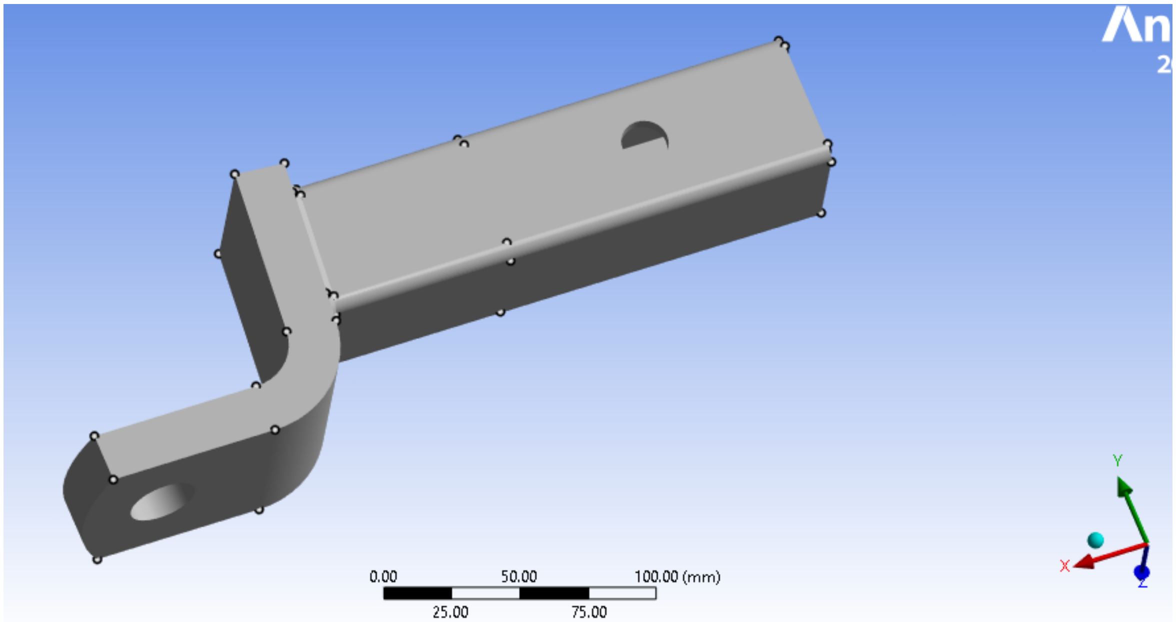


Conclusions

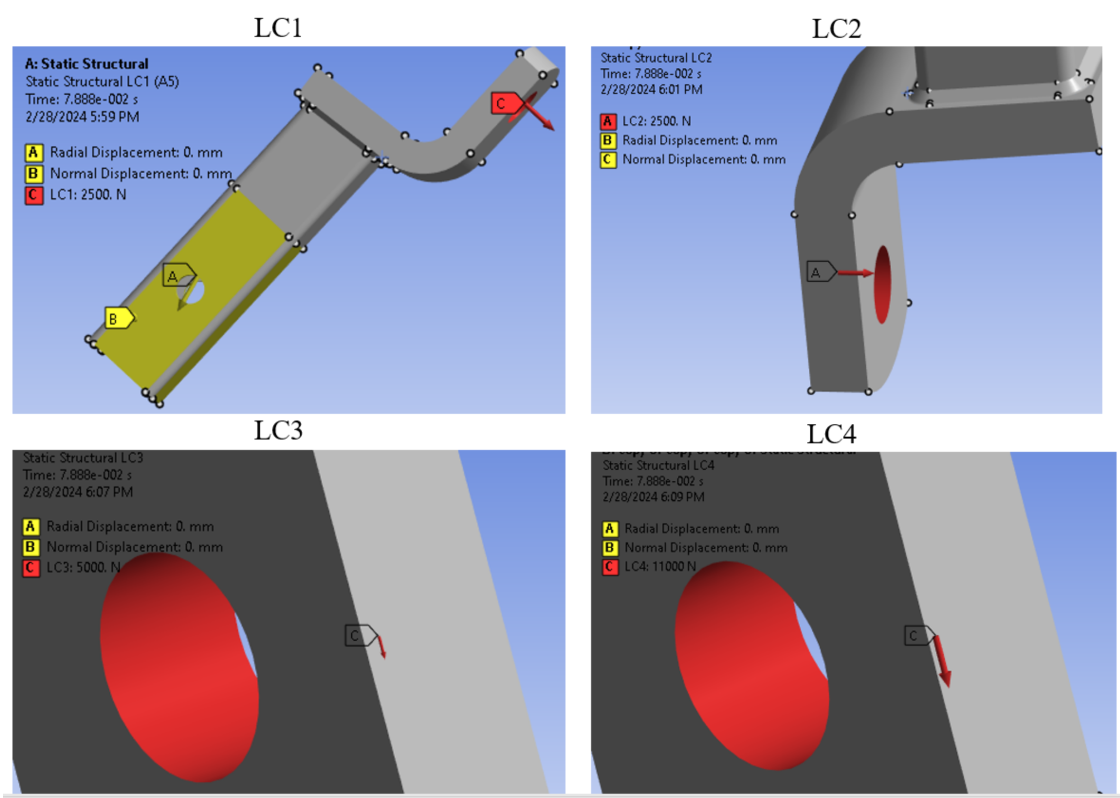
This analysis confirmed the structural integrity and fatigue life of the trailer hitch, highlighting the importance of mesh refinement and fatigue analysis in identifying critical stress areas. A focused redesign effort successfully improved the safety factor for the most challenging load case (LC4), demonstrating the effectiveness of iterative design optimization in enhancing structural performance and safety within specified loading conditions.

Appendix:

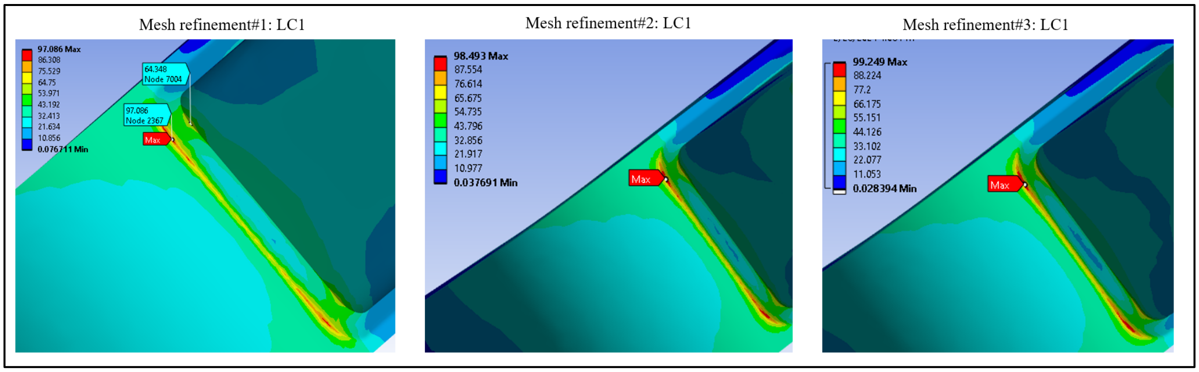
The figure below, accompanied by a scale bar, provides a visual representation of the hitch's dimensions.



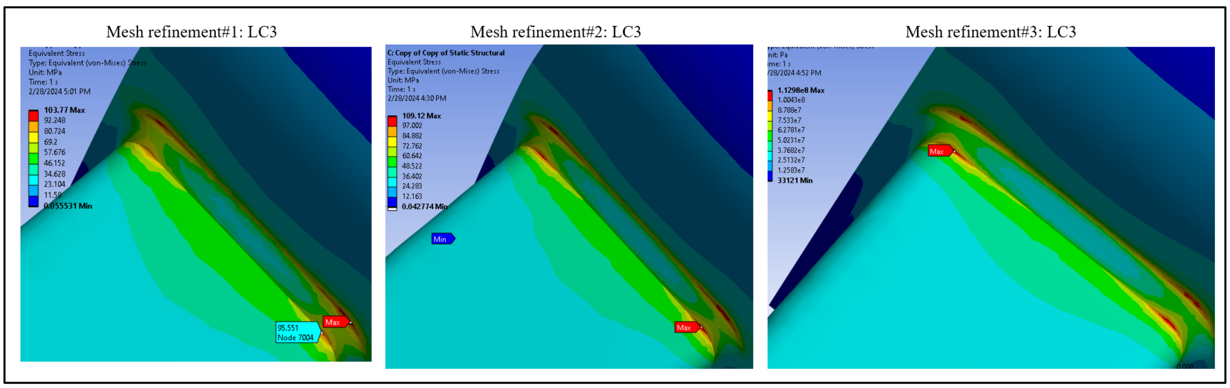
Boundary Conditions for Each Load Case:



Mesh Refinement for LC1:



Also, mesh refinement result for LC3 has shown below:



Upon examining the maximum stress, it is observed that a 5 mm mesh with two levels of refinement is adequate, leading to converged values. It is noted that the location of maximum stress has slightly shifted from the smaller fillet to the larger fillet, potentially indicating a singularity. However, given that the difference in peak stress is less than 5%, we consider the mesh with two refinements to be sufficiently detailed for the convergence study.

The table below shows the toolbox used for the fatigue analysis:

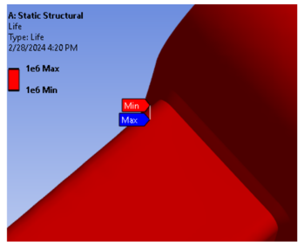
- Fatigue Tool
  - Damage
  - Life
  - Safety Factor
  - Equivalent Alternating Stress

Details of "Fatigue Tool"

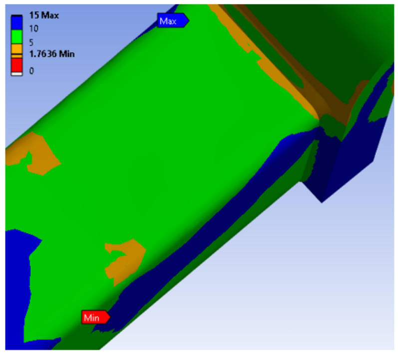
Domain	
Domain Type	Time
Materials	
Fatigue Strength Factor (Kf)	1.
Loading	
Type	Fully Reversed
Scale Factor	1.
Definition	
Display Time	End Time
Options	
Analysis Type	Stress Life
Mean Stress Theory	Goodman
Stress Component	Signed von-Mises
Life Units	
Units Name	cycles
1 cycle is equal to	1. cycles

Fatigue analysis for LC1:

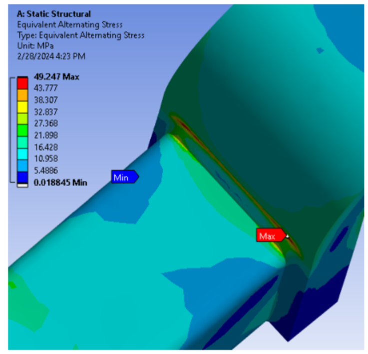
LC1: Life



LC1: Safety factor

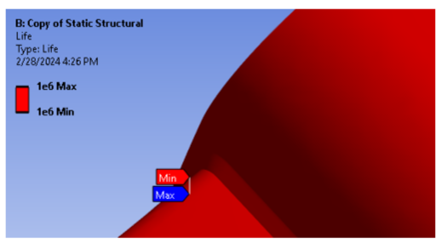


LC1: Equivalent alternating stress

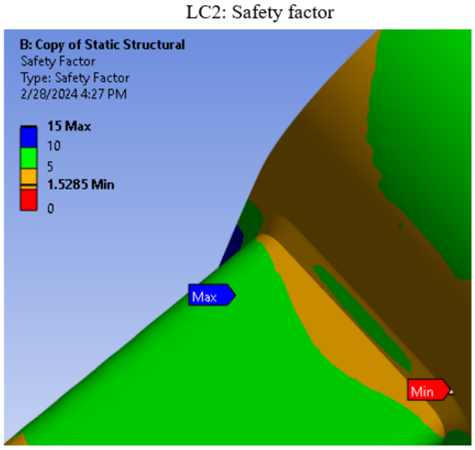


Fatigue analysis for LC2:

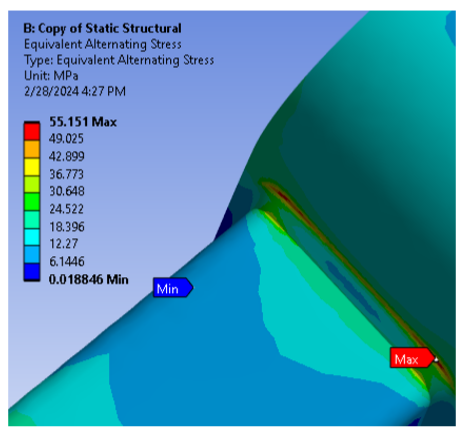
LC2: Life



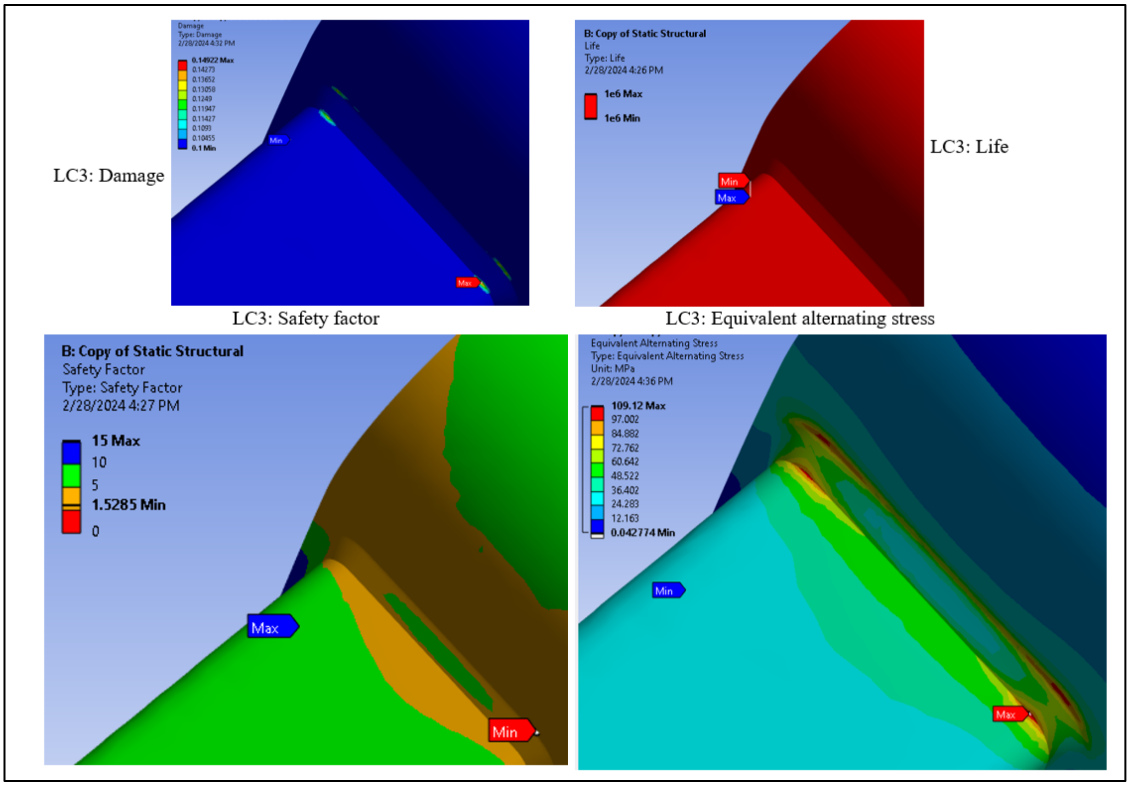
LC2: Safety factor



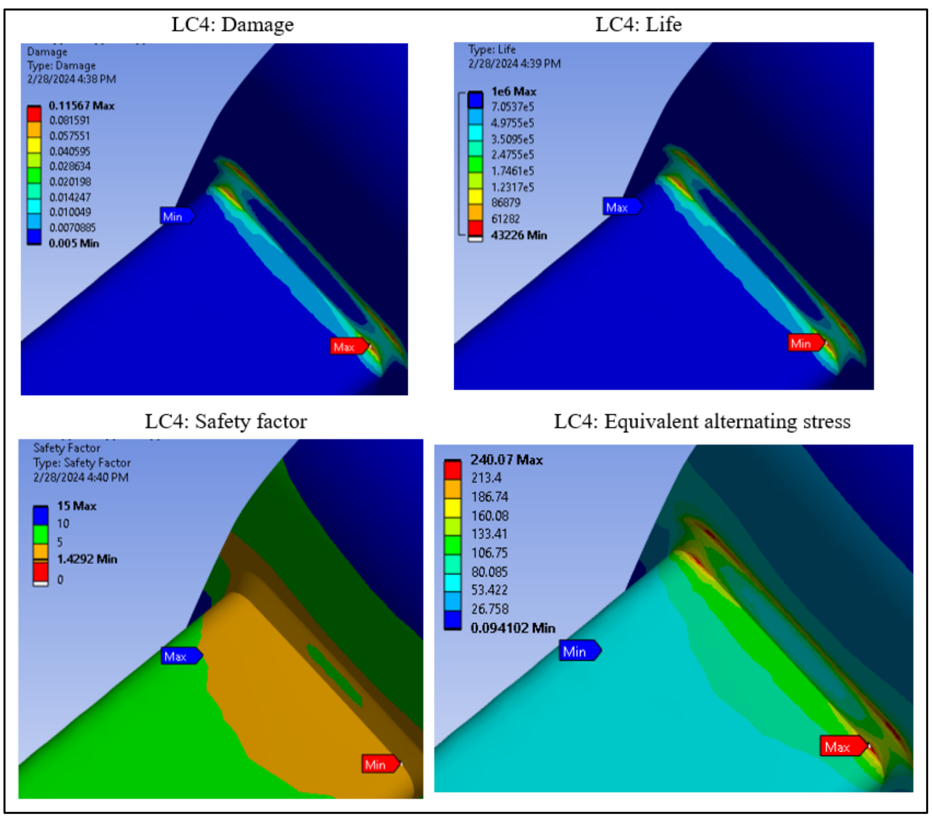
LC2: Equivalent alternating stress



Fatigue analysis for LC3:



Fatigue analysis for LC4:



Equivalent Stress Plot for LC4 in the Proposed Design: