Combat Robotics Weapon Impact Analysis

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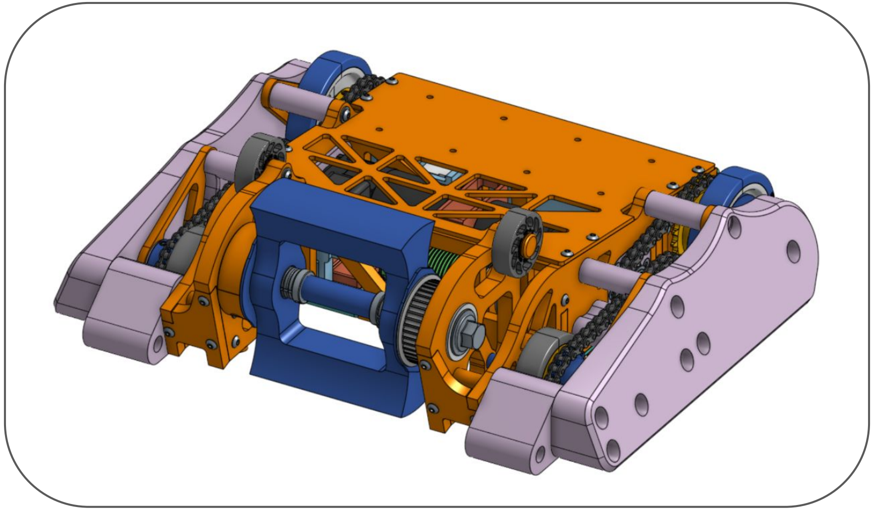
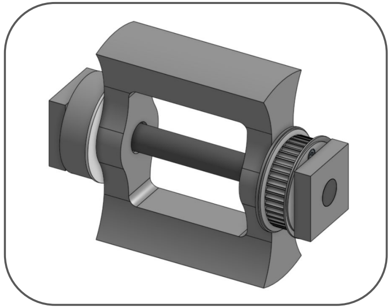
Background

- ➢ NDSU's "Boulderbot" Combat Robotics team is designing a 30 lb combat robot which will compete in UIUC's Robobrawl XI.
- ➢ The weapon is a ~7 lb modified drum spinning in the vertical direction.
- ➢ Structural group is interested in evaluating performance of the weapon through non-destructive testing (NDT) methods as the organization's material is scarce.
- ➢ Analysis done with static structural -> modal analysis -> explicit dynamics
- ➢ Weapon subsystem characteristics:
  - Material: S7 Tool Steel
  - Estimated Mass Moment of Inertia: 34.8 in^2lb
  - Power Transfer Mechanism: Timing Belt
  - Motor Max RPM: 56,000 (Castle 1717)
  - Theoretical Weapon Max RPM: 8,000 RPM (837.76 rad/s)

"Boulderbot" CAD (Onshape) model, color coded by material

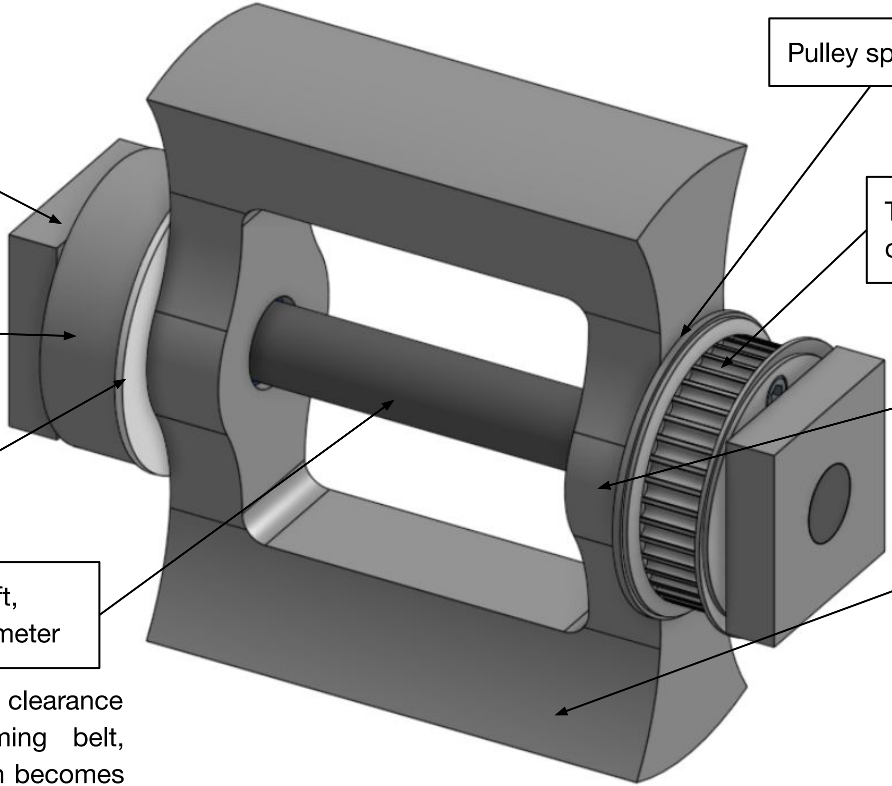


*Any complex geometries that need to be simplified downstream were changed through Ansys Discovery



CAD Model to be used for subsequent analyses

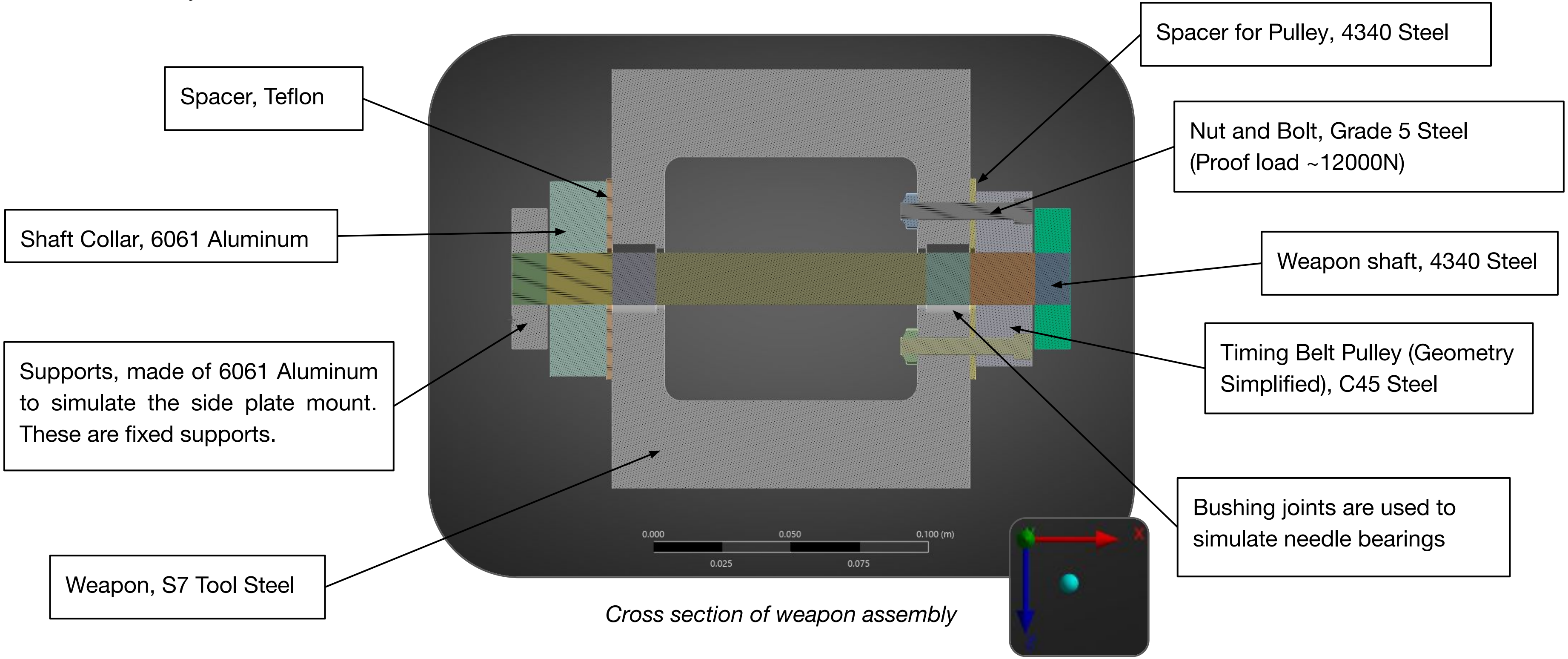
Background



*The pulley spacer is meant to provide clearance between the weapon and the timing belt, especially incase the tip of the weapon becomes deformed near the timing belt.

Finite Element Model Setup

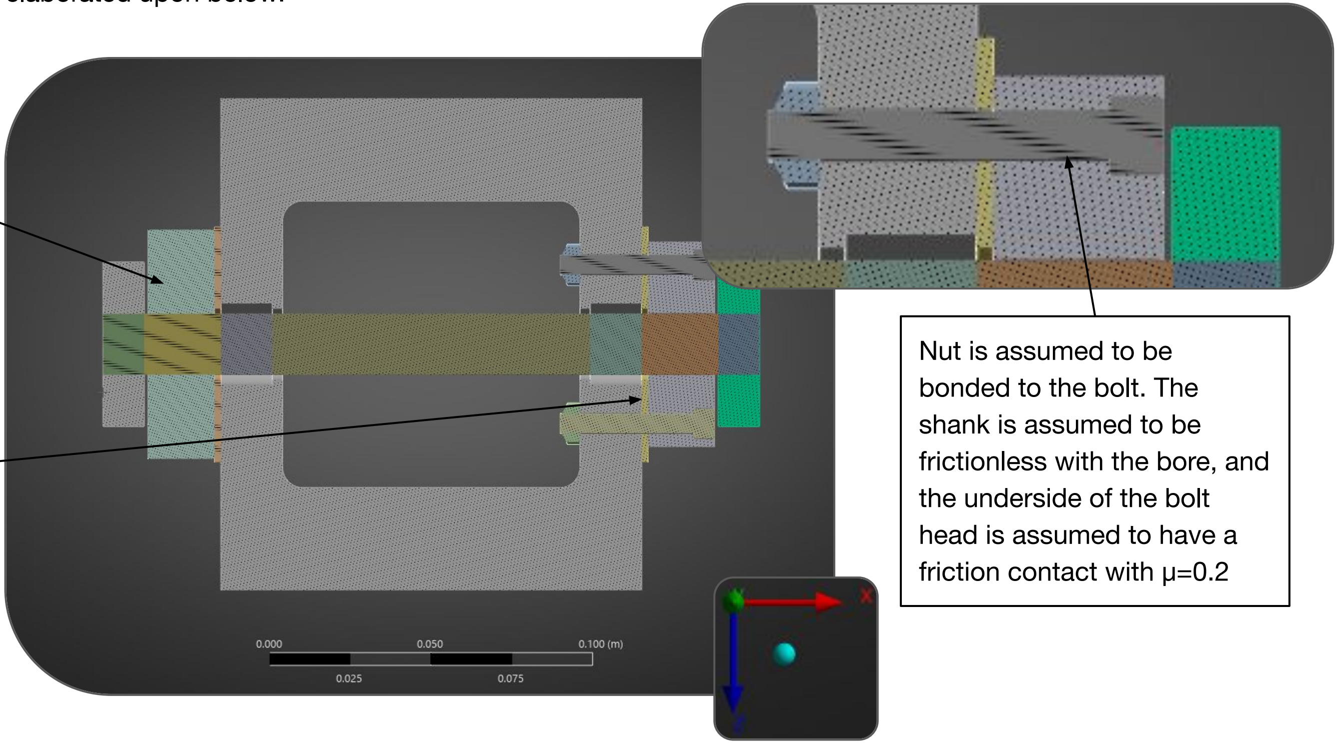
- Using Ansys Workbench 2025 R2 Student Version
- All body interactions are frictionless unless indicated otherwise.



Cross section of weapon assembly

Finite Element Model Setup

- Rigid body motion is to be expected around the x axis because of the bushing joint. Weak springs are turned on for analyses (such as static structural) to attain a solution.
- Surface contacts are further elaborated upon below.

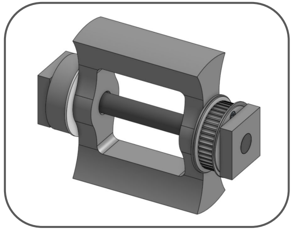


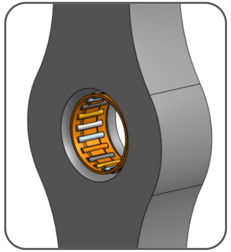
Collar (clamp-on) and teflon spacer are assumed to be bonded to each other and the weapon shaft to simplify analysis

The weapon - spacer - pulley stack is assumed to be in frictional contact with one another with μ=0.2

Nut is assumed to be bonded to the bolt. The shank is assumed to be frictionless with the bore, and the underside of the bolt head is assumed to have a friction contact with μ=0.2

Finite Element Model Setup





- ➢ The weapon shaft is a “dead shaft” - the weapon spins around the shaft rather than the shaft spinning at the supports.
- ➢ This allows for the weapon to continue to operate even if the supports at the end of the shaft are damaged.
- ➢ A timing belt turns the pulley, and the pulley is attached to the weapon through the bolts

- ➢ Most probable load path through the subsystem upon impact are the 2 bolts holding the pulley to the weapon, therefore modeling the needle bearings are not of particular interest.
- ➢ Bearing will be modeled as a simplified linear bushing / joint stiffness matrix (instead of explicit rollers + contact) to reduce model cost while preserving support stiffness behavior. The issue is stiffness depends on the force that the bearing experiences.

- ➢ 1e9 is a ballpark approximation from Molnár et al.
- ➢ This part of the model could be refined for a more accurate analysis. For now, this is a good enough approximation.
- ➢ A value of 1e8 (10% of Kzz and Kyy) for stiffness in the Z direction (Kzz, parallel the weapon shaft) to avoid a free axial mode.

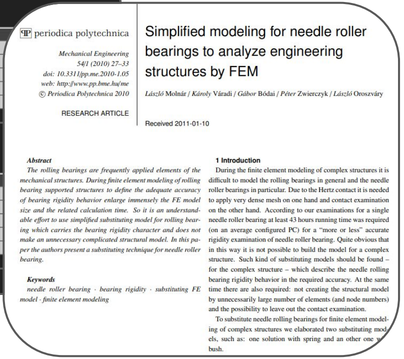
Stiffness Coefficients

Stiffness	Per Unit X (m)	Per Unit Y (m)	Per Unit Z (m)	Per Unit θx (°)	Per Unit θy (°)
Δ Force X (N)	1.e+009				
Δ Force Y (N)	0.	1.e+009			
Δ Force Z (N)	0.	0.	1.e+008		
Δ Moment X (N·m)	0.	0.	0.	0.	
Δ Moment Y (N·m)	0.	0.	0.	0.	0.
Δ Moment Z (N·m)	0.	0.	0.	0.	0.

Damping Coefficients

Viscous Damping	Per Unit X (m)	Per Unit Y (m)	Per Unit Z (m)	Per Unit θx (°)	Per Unit θy (°)
Δ Force * Time X (N·s)	0.				
Δ Force * Time Y (N·s)	0.	0.			
Δ Force * Time Z (N·s)	0.	0.	0.		
Δ Moment * Time X (N·m·s)	0.	0.	0.	0.	
Δ Moment * Time Y (N·m·s)	0.	0.	0.	0.	0.
Δ Moment * Time Z (N·m·s)	0.	0.	0.	0.	0.

Stiffness matrix values for bushing joint



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RESEARCH ARTICLE

Simplified modeling for needle roller bearings to analyze engineering structures by FEM

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Abstract

The rolling bearings are frequently applied elements of the mechanical structures. During finite element modeling of rolling bearing supported structures to define the adequate accuracy of bearing rigidity behavior enlarge immensely the FE model size and the related calculation time. So it is an understandable effort to use simplified substituting model for rolling bearing which carries the bearing rigidity character and does not make an unnecessary complicated structural model. In this paper the authors present a substituting technique for needle roller bearing.

Keywords

needle roller bearing · bearing rigidity · substituting FE model · finite element modeling

1 Introduction

During the finite element modeling of complex structures it is difficult to model the rolling bearings in general and the needle roller bearings in particular. Due to the Hertz contact it is needed to apply very dense mesh on one hand and contact examination on the other hand. According to our examinations for a single needle roller bearing at least 43 hours running time was required (on an average configured PC) for a “more or less” accurate rigidity examination of needle roller bearing. Quite obvious that in this way it is not possible to build the model for a complex structure. Such kind of substituting models should be found – for the complex structure – which describe the needle rolling bearing rigidity behavior in the required accuracy. At the same time there are also required: not creating the structural model by unnecessarily large number of elements (and node numbers) and the possibility to leave out the contact examination.

To substitute needle rolling bearings for finite element modeling of complex structures we elaborated two substituting models, such as: one solution with spring and an other one w[...] bush.

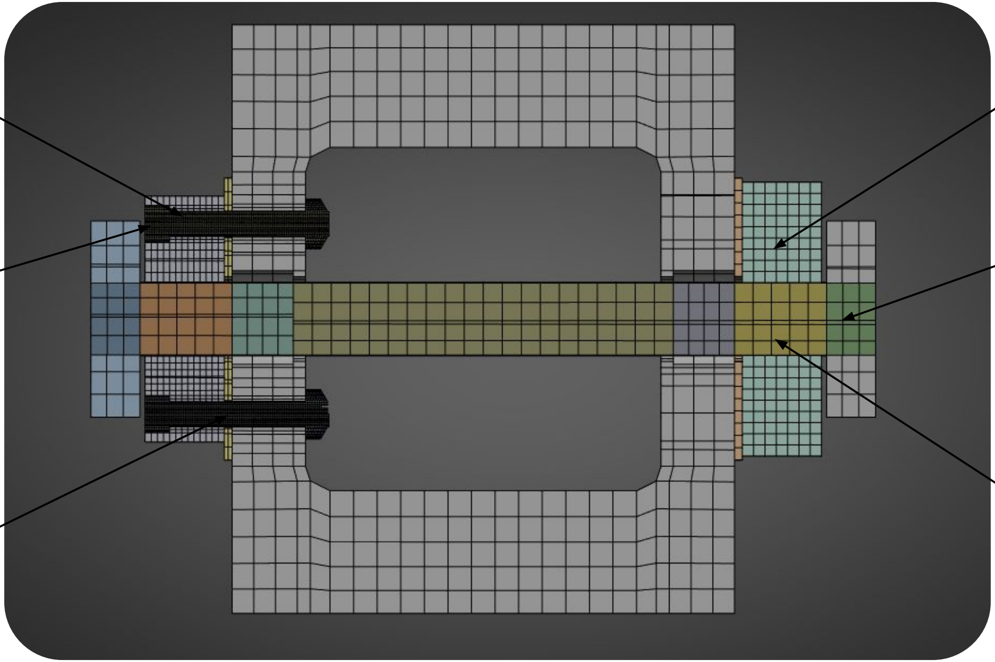
Finite Element Model Setup

- ➢ Adaptive mesh is turned on
- ➢ Convergence study will be performed for static structural analysis

Hole wall is given 1mm mesh

Bolt and nut are given multizone mesh method

Bolts have a more refined mesh since they will be preloaded. Additionally, they are the likely point of failure for the impact analysis



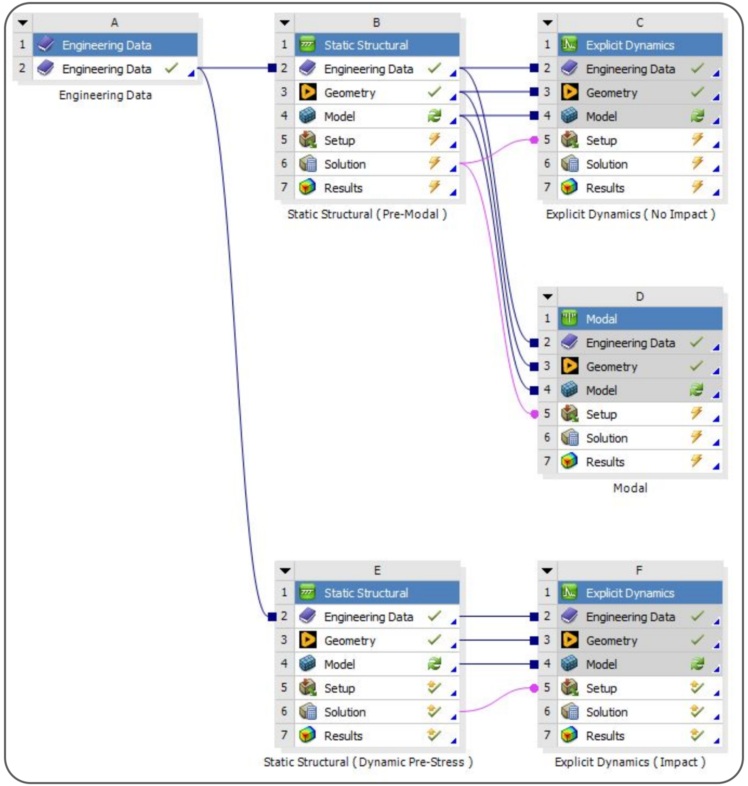
Cross section of weapon assembly - Mesh

All bodies except for the pulley spacer is given multizone mesh method

Support and shaft are given 5mm mesh

The weapon bar is divided into parts to allow the bearing contact node to center on the seat location rather than in the middle of the whole bar

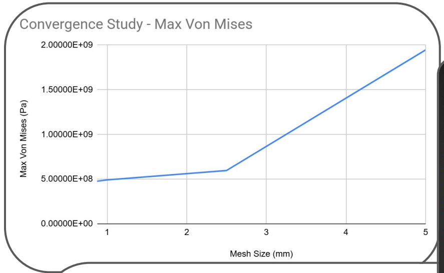
Analysis Setup

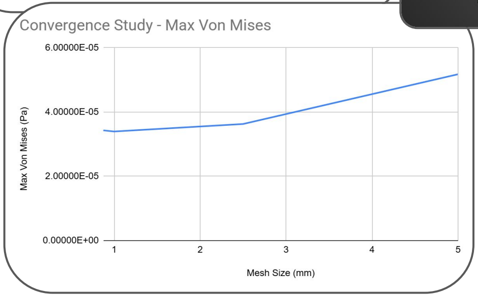


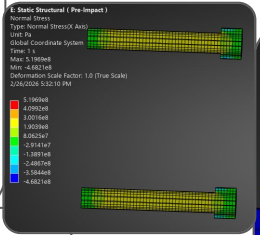
- 2 separate systems, one for modal analysis and verifying that the explicit dynamic model works (no impact target), and the other for explicit dynamics with impact to a rigid target.
- Modal analysis performed on the weapon assembly to determine if its natural frequencies line up with any operating excitation frequencies + identify any resonance risks.
  - Forcing frequency 8000 RPM / 60 = 133.3 Hz
- Explicit dynamics impact analysis performed to determine impact deformation (particularly on the weapon) and the risk of shearing the 2 bolts

Results (Static Structural)

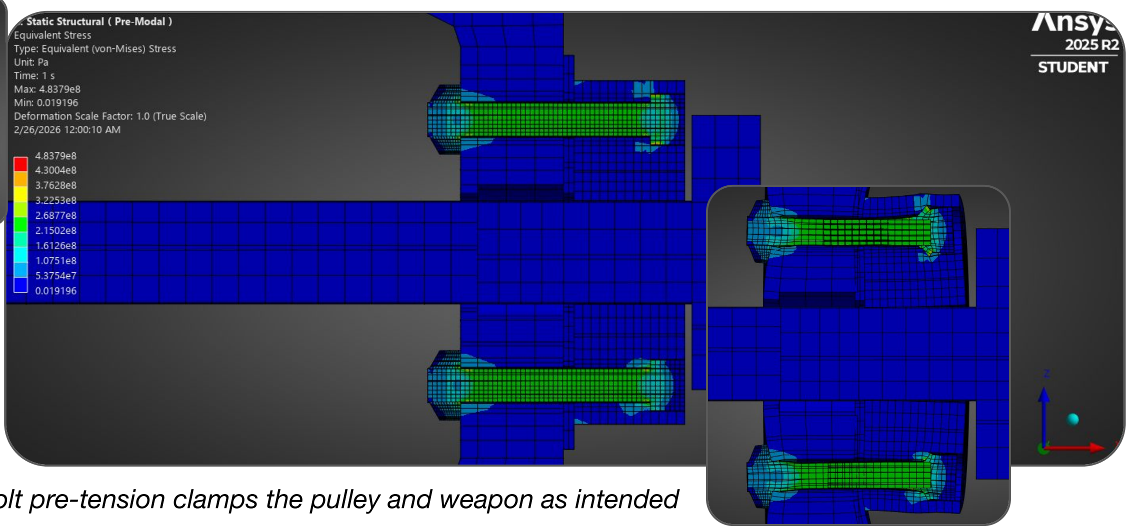
- ➢ Static structural (bolt pretension)
- ➢ Main expected load path: bolts -> 9000 N pretension per bolt (based on 75% proof load)
- ➢ Main goal here is to verify that the pre-tension clamps the pulley+weapon as intended, and that the loads make sense.
- ➢ Convergence study done with body mesh size on the bolt
- ➢ For a ¼” bolt, x-sectional area perpendicular to tension is 20.55 mm^2, and with 9000 N the expected normal stress is around 4.39E8 Pa (σ=F/A)







- ➢ Maximum von mises output seem to stabilize around 1 mm mesh size



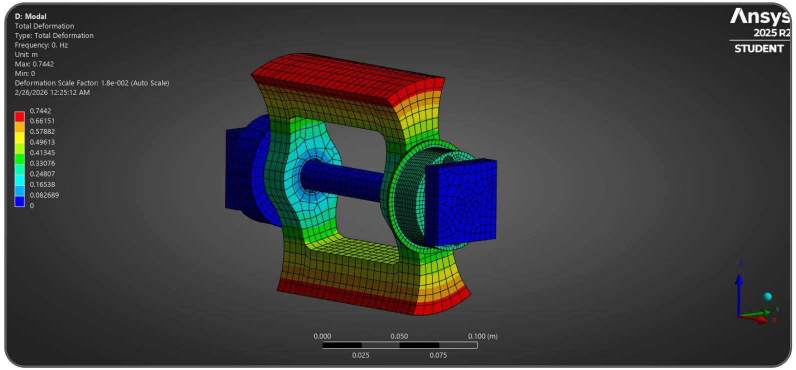
Bolt pre-tension clamps the pulley and weapon as intended

Results (Modal)

- ➢ The first 6 modes are gathered

Mode	Frequency (Hz)
1	0
2	110.43
3	136.4
4	653.5
5	748.95
6	1021.2

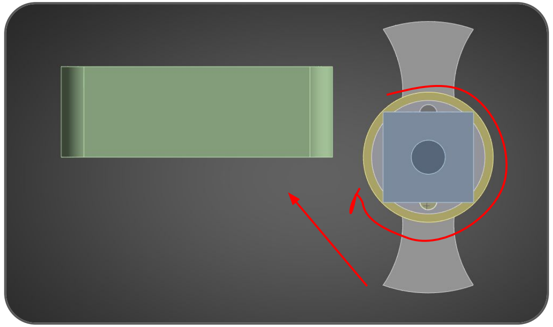
- ➢ The 1st mode being 0 is a result of the free spin DOF around the shaft. It is intentionally left unconstrained to represent the weapon's ability to freely spin.
- ➢ Subsequent modes represent the 1st elastic vibration modes of the assembly
- ➢ Of particular interest is the 3rd mode, since it's close to the weapon's operating frequency at 8000 rpm (133.3 Hz)
- ➢ Concern would be high vibration sensitivity, particularly if weapon is slightly unbalanced

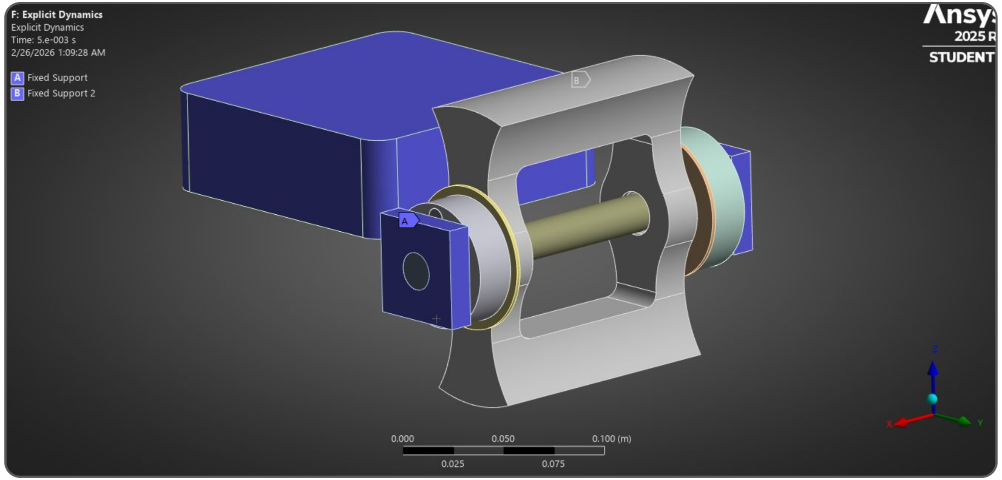


Deformation plot of the 1st mode.

Setup (Explicit Dynamics)

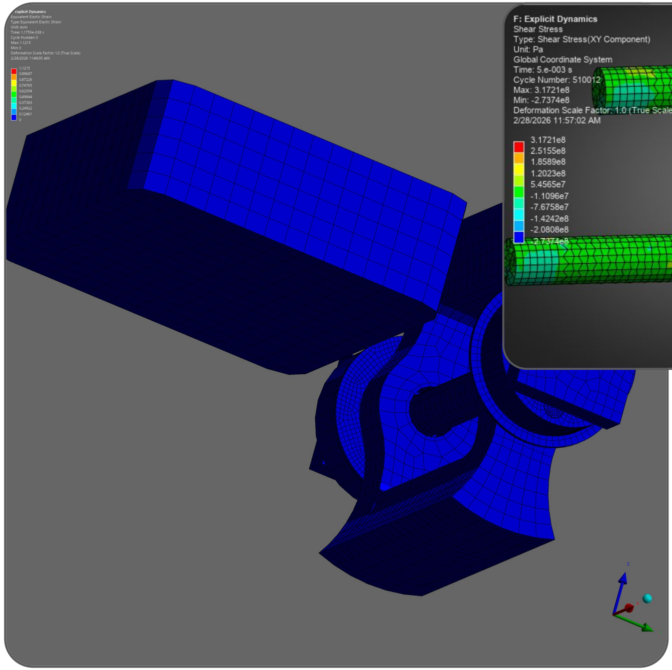
- ➢ 6x6x2" Aluminum 6061 block placed in front of the weapon, fixed supports as shown in the image below.
- ➢ The weapon's spin direction is such that the bottom edge of the weapon impacts the underside of the target.
- ➢ The preload results from the preceding static structural analysis was set as the pre-stress condition and an initial angular velocity about the x-axis was given to rotating components at 837.76 rad/s
- ➢ Probing outputs such as kinetic energy, maximum shear stress on the bolts, etc.

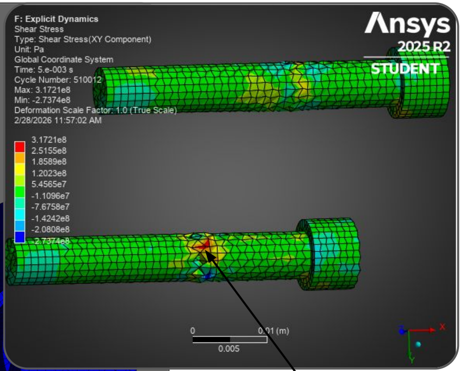




- ➢ It is important to note that the 837.76 figure is based off of the *theoretical* max RPM of the weapon at 8000 rad/s
- ➢ The weapon is also not likely to hit targets in actual combat with a full uppercut in a similar manner as this current setup.
- ➢ This means that the output extracted from this analysis is treated as the absolute worst-case loading scenario for the weapon system.
- ➢ A cyclic loading analysis may not be appropriate either as the service life is very short. Of interest is the maximum shear stress on the bolts or unexpected load paths to ensure there is minimal plastic deformation.

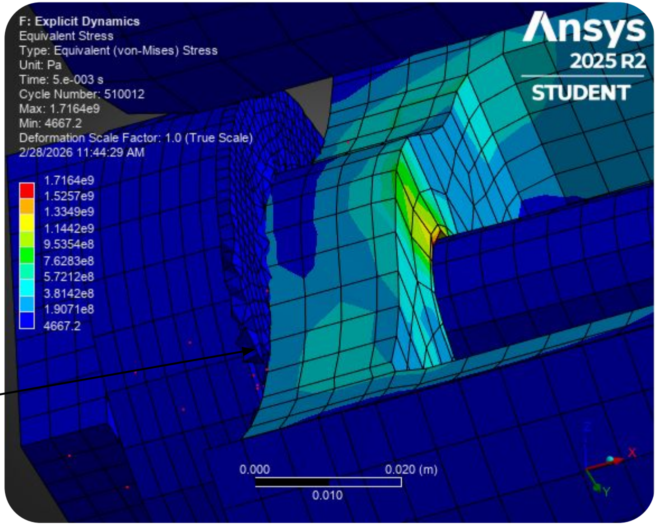
Results (Explicit Dynamics)





- Maximum shear stress experience by bolt -> 371 MPa
- Grade 5 steel has yield strength of 92,000 psi (634 MPa)
- Von mises yield criteria
  - σ_shear_yield = σ_yield / $\sqrt{3}$ = ~364 MPa
  - By that estimate, the bolt is likely to plastically deform under worst-case scenario

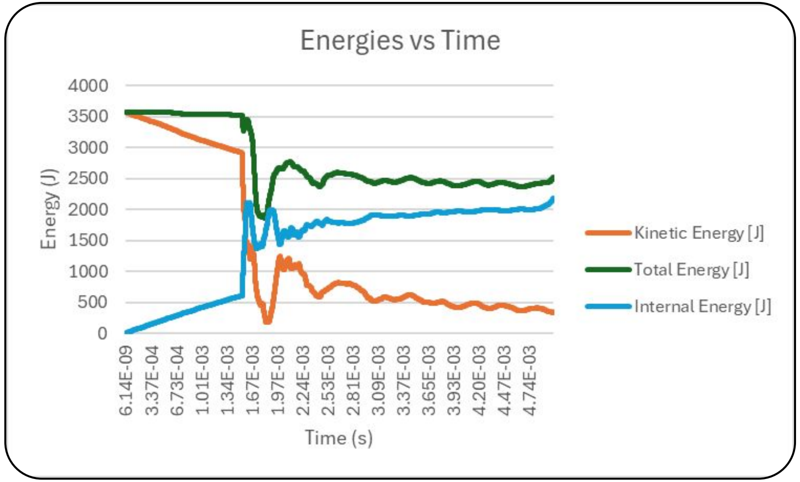
Shearing action experienced between the pulley spacer - weapon interface

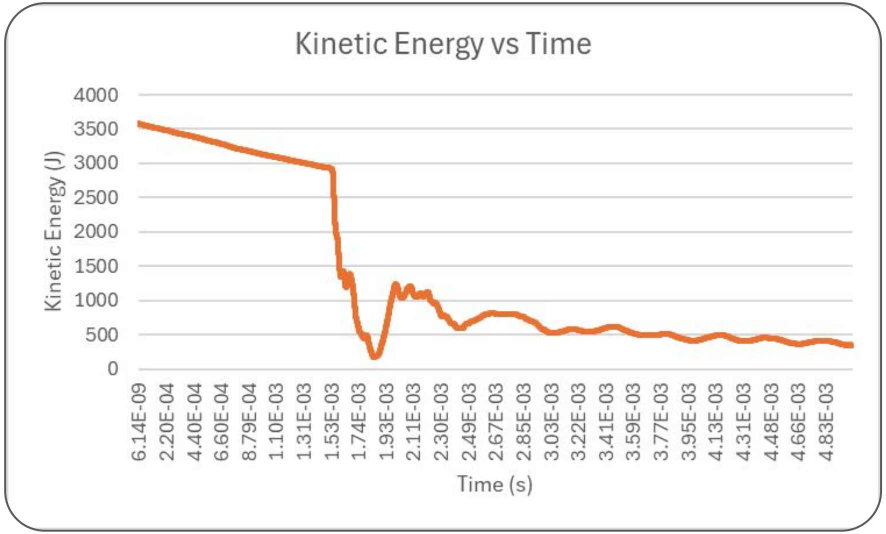


- Much deformation was experienced by the teflon spacer, and so was remeshed with 2mm body sizing before the analysis. Spacer experiences erosion as shown on the right.

Results (Explicit Dynamics)

- Energy of the weapon body
- ~3500 J at 8000 RPM with current geometry
- Sharp drop in KE at ~1.3 ms
- From 3000 J to almost 0, roughly 2800 J transferred
- Rebound peak of 1000 J, some elastic strain transferred into rotational motion.
- Not perfectly inelastic

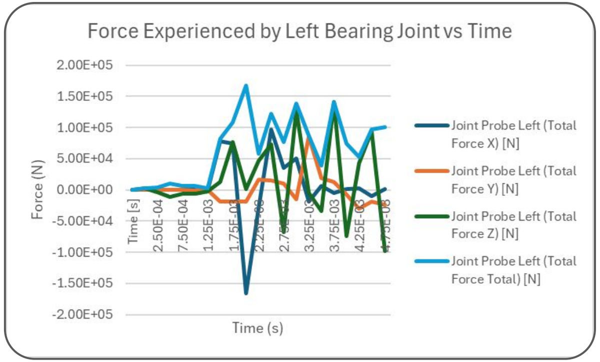


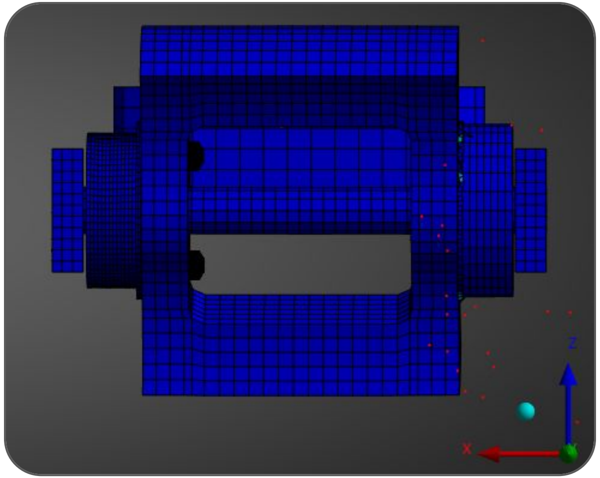


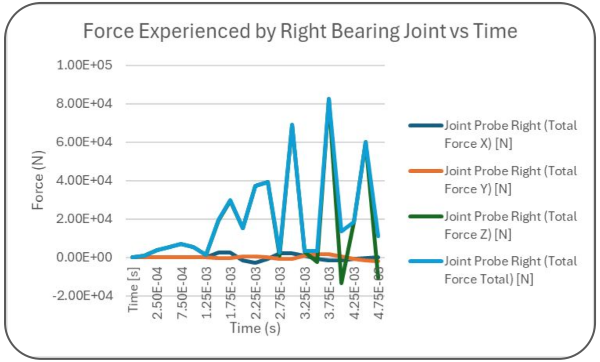
- Weapon assembly model is soft -> 3-6 cycles of oscillation
- This is because no motor torque is assumed, in actuality a timing belt will be connected to the pulley which would restrict some movement

Results (Explicit Dynamics)

- ➢ Total forces experience by each “needle bearing” bushing joint
- ➢ Left bearing takes more peak load - geometric center of mass is offset to the left - bearings are loaded asymmetrically
- ➢ Shaft is likely experiencing torsional / lateral vibration as evident from oscillation

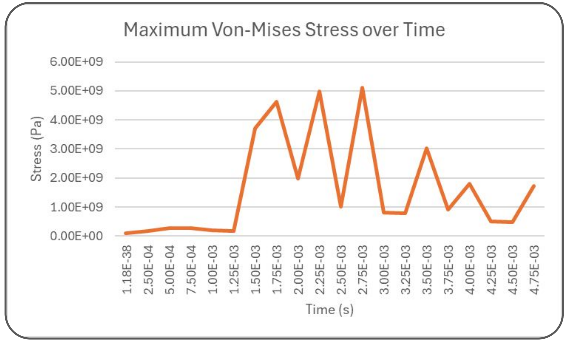


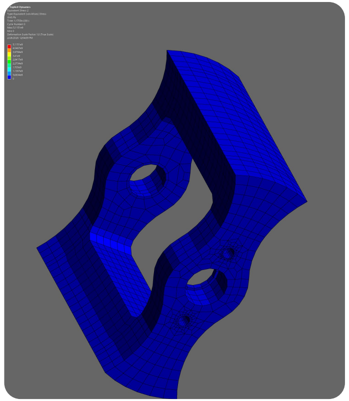


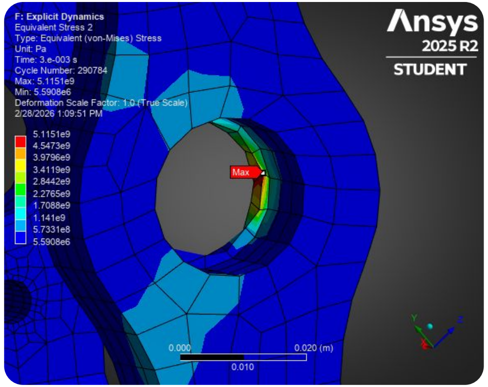


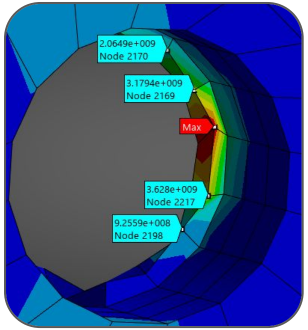
Results (Explicit Dynamics)

- ➢ 5.11 GPa max von mises stress experienced in bearing seat over time according to model
- ➢ Likely not accurate as the mesh is not refined at that maximum
- ➢ Cannot further refine mesh due to limitations of student license
- ➢ Maximum von-mises over time jumps erratically, sign of non-converged contact stresses due to coarse mesh or high distortions









- ➢ Stress hotspot appears on the side of the bearing bore
- ➢ Impact is pushing the weapon sideways
- ➢ Further proof that bearings are loaded asymmetrically
- ➢ Though numbers are not dependable, qualitative weak spot (concentration on the bearing shoulder) is correct
- ➢ May point to reinforcing this shoulder

Conclusions

- For model:
  - make bolt diameter thicker to ~⅜”
  - Needle thrust bearings instead to support axial loads.
  - Harden S7 tool steel for weapon (anneal), may help in rigidity of bearing seat.
  - Weapon shaft could be thicker to reduce dynamic loads on the weapon + bearings.
  - Reinforce bearing shoulder with more material to reduce stress hotspot.
  - Ensure the selected bearings can withstand the simulated forces
  - At max RPM, weapon operates dangerous close to the 3rd mode. Future design iterations should shift this mode away from operating speed.
- For analysis
  - Make target closer to weapon tip to preserve initial angular velocity at impact.
  - Finer mesh around weapon bearing seat stress hotspot - yield more accurate stress predictions.