NAU SAE Toolbox

Engineering Calculations Summary

Derek Griffith, Logistics Manager, Prototype Engineer

Hailey Hein, Project Manager, CAD Engineer, Financial Manager

Haoran Li, Test Engineer

Yanbo Wang, Manufacturing Engineer, Website Manager

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Project Sponsor: NAU SAE

Faculty Advisor: David Willy

Instructor: Armin Eilaghi

DISCLAIMER

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EXECUTIVE SUMMARY

The SAE Toolbox Capstone Project, undertaken by a team of Northern Arizona University engineering students, addresses the needs of the NAU SAE Formula and Baja Teams by designing and manufacturing a robust, multifunctional toolbox cart for use in competition pits and shop environments. Initiated in Summer 2025 and continuing through Fall 2025, the project responds to client requirements for a mobile, durable, and efficient cart capable of supporting race-day operations and year-round shop use. Key customer requirements include maneuverability on uneven terrain, organized storage for tools and equipment, a secure fire extinguisher mount, onboard power for charging tools, and single-person operability. The project, sponsored by NAU SAE and advised by Dr. David Willy, has a budget of $2,000, with a projected $2,129.32 spent to date, bolstered by a fully sponsored $1,150 base frame and a $501 sponsorship from Findlay Toyota Flagstaff.

This report presents the engineering models, calculations, and supporting evidence used to validate the design of the NAU SAE Formula Team's custom toolbox cart. The cart integrates multiple subsystems including the frame, steering, brakes, storage, casters, and power supply into a single cohesive platform designed to safely transport up to 500 pounds of tools and equipment across varied competition terrains. Each subsystem was analyzed under worst-case load scenarios, with material selection, geometry, and connection methods optimized to achieve minimum factors of safety above the acceptable threshold. Codes and standards from AWS, ASTM, OSHA, NFPA, and ANSI informed the structural design, weld specifications, and safety features. The project team used CAD models, flow diagrams, and system-level charts to visually document the layout, load paths, and functionality of each subsystem, ensuring that the design intent is clear for fabrication and integration.

The analyses performed during this project highlighted the critical areas requiring design improvements and verified the components with significant safety margins. Frame analysis identified the need to upgrade bracing on the 1×1-inch steel tubing to meet bending and deflection targets. Steering and braking systems were evaluated under peak handling and stopping loads to ensure reliable operation in tight pit spaces. The power supply subsystem was verified structurally but flagged for reinforcement of hold-down straps to improve safety margins. Storage and caster systems were examined for deflection and stress under shock loading to ensure usability and durability over time. Future tests will include hill-stopping, mock tech inspections, and maneuverability assessments under a 500-pound load to ensure single-user operability and reliability in real-world conditions. The project remains on track to deliver a fully functional, field-tested pit cart by the end of Fall 2025, accompanied by a comprehensive SolidWorks CAD model, technical drawings, and sponsor recognition materials, ensuring alignment with both client expectations and course requirements.

Overall, this engineering model summary demonstrates that the proposed toolbox cart is structurally feasible, compliant with relevant codes, and meets or exceeds the stated design objectives. The work completed thus far provides a solid foundation for final verification testing, fabrication of upgraded components, and the full integration of electrical and mechanical subsystems into the assembled cart. This report also serves as a reference for future improvements, maintenance, and training related to the system, ensuring that the team's investment supports its performance for multiple competition seasons.

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1 TOP LEVEL DESIGN SUMMARY

The SAE Toolbox Capstone Project addresses the need of the NAU Baja and Formula SAE teams for a rugged, multifunctional pit cart that provides reliable tool storage, mobility, and safety in both shop and competition environments. The solution is a custom-designed toolbox cart optimized for maneuverability over rough terrain, efficient organization of tools and spare parts, integrated charging capability, and a secure fire extinguisher mount. This system reduces downtime during competition and improves overall team efficiency.

A SolidWorks CAD model of the current design (Figure 1) illustrates the top-level system layout. The design is divided into several integrated subsystems, each serving a critical role in meeting customer requirements. The frame and structure form the foundation of the cart, consisting of welded steel tubing designed to support over 500 pounds of tools and equipment while maintaining stability on uneven ground. Attached to this structure is the mobility subsystem, which incorporates oversized rubber wheels between 8 and 10 inches in diameter to ensure smooth transport over gravel, grass, and other rough terrains commonly encountered at competition sites. The steering and braking mechanisms are designed so that a single operator can safely maneuver and control the cart, even when fully loaded.

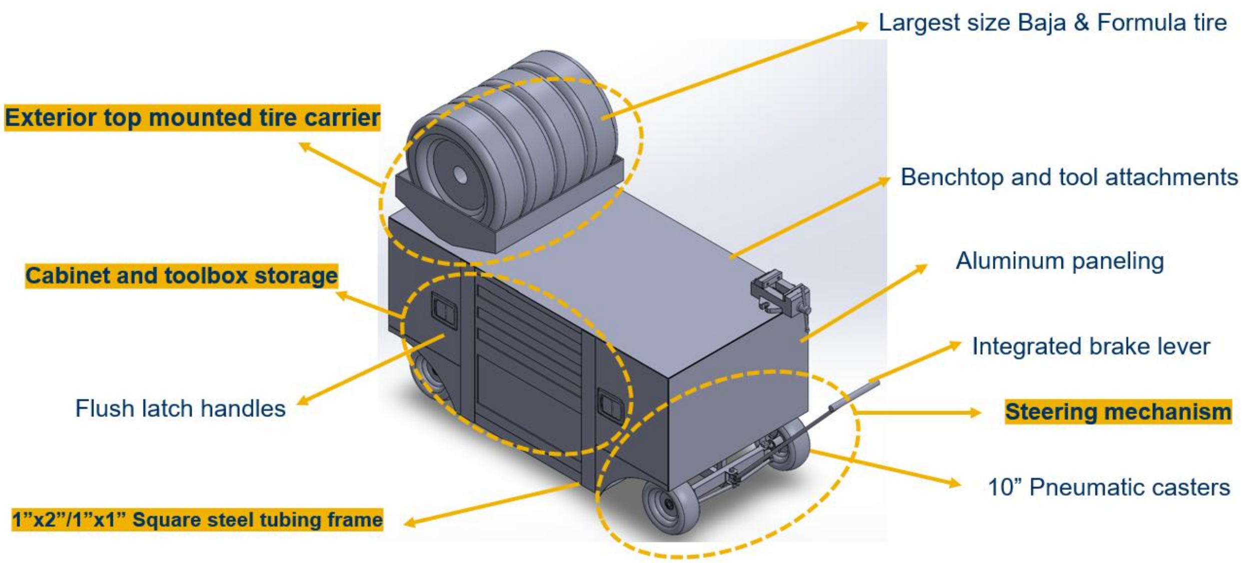

Figure 1: Top Level Cad Model

The storage subsystem is central to the design and includes both a large 3′×2′×1′ gear compartment for bulky items and multiple drawers for smaller SAE-specific tools. To protect high-value equipment, the storage space features a lockable compartment with a latch projected to withstand 100 pounds of pull force. Safety is further addressed through the fire safety subsystem, which integrates a dedicated mount for a standard 10-pound fire extinguisher in an easily accessible location. The design also includes a power and charging subsystem, allowing the teams to charge onboard batteries and operate electric tools directly from the cart, eliminating downtime and improving workflow efficiency. Finally, the sponsor display subsystem provides designated surfaces for mounting engraved nameplates or branded stickers, ensuring visibility and recognition for industry and community sponsors who contribute to the project.

Together, these subsystems integrate into a single, unified system that meets the customer requirements for functionality, safety, and ease of use. The cart remains within a 30 inch by 60-inch footprint to fit into trailers and pit spaces, passes tilt and balance requirements to ensure stability on 10° inclines, and can be operated safely by one person with less than 50 pounds of force applied on a 5° slope. The CAD model demonstrates how these elements come together to create a robust, transportable, and competition-ready toolbox solution. An updated QFD matrix (Figure 2) further illustrates the systematic translation of customer requirements into engineering requirements, showing how mobility, storage capacity, fire safety, power supply, and sponsor visibility are addressed through specific subsystems in the design.

	Rubber Casters 6" to 10"	Integrated manual steering	Hand Braking/ Locking Wheels	Acceptable Volumetric Footprint in Trailer	Locking Drawers	3'x2'x1' Volume for Driver bag	Fire Extinguisher Mount	Integrated Power System for Tools	Charging Capability	Powered Tools Storage	Sound System (optional)	Mounting locations for vice, arbor press, etc	Shade	Welded Aluminum frame with bolted accessories
Rubber Casters 6" to 10"														
Integrated manual steering	9													
Hand Braking/ Locking Wheels	9	9												
Acceptable Volumetric Footprint in Trailer														
Locking Drawers				6										
3'x2'x1' Volume for Driver bag				6										
Fire Extinguisher Mount														
Integrated Power System for Tools														
Charging Capability								9						
Powered Tools Storage				6	3			9						
Sound System (optional)														
Mounting locations for vice, arbor press, etc				6										
Shade				6										
Welded Aluminum frame with bolted accessories				3	3					3				
Storage for saftey equipment, brake bleed kit, saftey wire puller.				6	9									

Customer Needs	Customer Weights	Rubber Casters 6" to 10"	Integrated manual steering	Hand Braking/ Locking Wheels	Acceptable Volumetric Footprint in Trailer	Locking Drawers	3'x2'x1' Volume for Driver bag	Fire Extinguisher Mount	Integrated Power System for Tools	Charging Capability	Powered Tools Storage	Sound System (optional)	Mounting locations for vice, arbor press, etc	Shade	Welded Aluminum frame with bolted accessories	Storage for saftey equipment, brake bleed kit, saftey wire puller.	1 Poor	2	3 Acceptable	4	5 Excellent
Large Toolbox	4	6	3	6	9	9					6		3		6	3			B		AC
Hold all of driver equipment/ spare wheels + rims	4	6			9		9												AC		B
Mounted items (arbor press, vice, grinding wheel,etc)	4	3			9				3	6	9		9					AC	B		
Price	5	3	3							3		6		6		3	ABC				
Brakes/ steering/ 6-10 inch casters	4	9	9	9															B		AC
Technical Requirement Units		inch	N/A	lbf/ N	ft^3	N/A	ft^3	N/A	Watts	Watts	ft^3	N/A	N/A	ft^3	ft^3	ft^3					
Technical Requirement Targets		10 in	Y	2+	72 ft^3	Y	6 ft^3	Y			2ft^3	Y	3 locations	72 ft^3	Y	2 ft^3					
Absolute Technical Importance		50	50	50	50	30	50	50	30	30	50	10	50	30	50	30	Total Score				
Relative Technical Importance		8%	8%	8%	8%	5%	8%	8%	5%	5%	8%	2%	8%	5%	8%	5%	610				

Legend	
A	Redline 75" Pit Cart
B	Winter Pit Products
C	DK Hardware

Figure 2: Complete House of Quality

The customer requirements for the SAE Toolbox reflect the needs of both the Baja and Formula SAE teams and are focused on functionality, safety, and usability in off-road and pit environments. First and foremost, the cart must be maneuverable on uneven surfaces such as gravel and grass, ensuring it can be transported to and around the competition site with ease. It must also provide ample storage for SAE-specific tools and spare parts, allowing quick access during repairs and adjustments. A locked compartment is required to secure high-value tools and equipment when unattended. Additionally, the design must integrate a standard ten-pound fire extinguisher in an accessible and secure mount. To support the team's workflow, the cart should include an onboard battery or charging capability for power tools. Visual representation of team sponsors is also important, so space for logos or branded stickers should be incorporated. Finally, the entire system should be operable by a single person, minimizing labor requirements and improving overall efficiency during competitions.

The SAE Toolbox design translates customer needs into measurable engineering requirement constraints for real-world performance. For off-road use, it must have rubber wheels at least 6" in diameter (ideally 8–10") for traction and shock absorption. A single user must operate it with ≤50 lb. of force on a 5° incline when fully loaded. Storage must include a 3'×2'×1' gear compartment and drawers for tools like brake bleed kits. A lockable compartment is required, with a latch that withstands 100 lb. of pull force. The toolbox must mount a standard 10 lb. fire extinguisher and power supply for charging. Its footprint must stay within 30"×60" to fit trailers. Structurally, the frame must support 500 lb. with a safety factor of 2 (verified by FEA) and remain stable on a 10° lateral incline.

2 SUMMARY OF STANDARDS, CODES, AND REGULATIONS

The SAE Toolbox Capstone Project is guided by a combination of engineering standards, safety regulations, and institutional requirements that ensure the design is both functional and compliant in competitive and laboratory environments. These standards apply to the structural integrity of the frame, the safe storage of equipment and hazardous materials, the electrical charging subsystem, and the overall usability of the cart in SAE competition pits.

The frame and welded structure are designed in accordance with the American Welding Society (AWS) D1.1 Structural Welding Code for Steel [1], which specifies joint design, inspection requirements, and allowable stresses for welded members. Material selection follows ASTM standards for mechanical properties of structural steels, such as ASTM A36 [2] for tubing and plate components, to ensure consistent yield strength and safety margins. Load-bearing requirements are based on a factor of safety of at least 2.0, verified through Finite Element Analysis (FEA) in SolidWorks in compliance with ASME design guidelines for pressure vessels and load-bearing frames.

OSHA 1910 [3] guidelines on material handling inform design limits for operator effort, restricting the required push/pull force to ≤50 pounds on a 5° incline, ensuring safe use by a single team member. The overall footprint is restricted to 30 inches by 60 inches, aligning with trailer space constraints and NFPA recommendations for egress and walkway clearance. Handles, drawer slides, and locks are positioned according to ANSI/HFES 100 [4] standards for ergonomics to promote safe and efficient human interaction.

Fire safety compliance is achieved by incorporating a mounting system for a standard 10-pound ABC fire extinguisher in accordance with NFPA 10 [5]: Standard for Portable Fire Extinguishers. The extinguisher is secured in a quick-release bracket to allow immediate access in the event of an emergency, which is a common safety requirement for both shop and track environments. Additionally, sharp edges are minimized or capped to align with OSHA guidelines on workplace safety.

The onboard charging subsystem follows NFPA 70 (National Electrical Code) [6] requirements for low-voltage circuits and battery-powered tools. All wiring and connectors are selected to meet UL-listed standards to ensure safe operation under typical shop and competition conditions. Overcurrent protection and insulated housings are incorporated to prevent hazards associated with short circuits or accidental contact.

The engineering requirements derived from these standards are summarized below: [7]

- **Frame capacity:** Must support ≥500 lb. with safety factor of 2.0 →
$$\sigma_{allow} = \frac{\sigma_y}{n} \tag{1}$$
- **Tilt stability:** No tipping on 10° lateral incline, where h_{cg} is center of gravity height and www is track width →
$$\tan(\theta) = \frac{h_{cg}}{\frac{w}{2}} \tag{2}$$
- **Operator force:** ≤50 lb. on 5° incline where f is rolling resistance coefficient →
$$F = Wsin(\theta) + fWcos(\theta) \tag{3}$$
- **Latch strength:** ≥ 100 lb. pull resistance per ANSI hardware standards.
- **Wheel diameter:** ≥ 6 in. (preferred 8–10 in.) per terrain mobility requirements.

By adhering to these standards, codes, and derived equations, the design ensures compliance with established engineering practice, competition safety expectations, and user requirements. The combination of AWS, ASTM, OSHA, NFPA, ANSI, and SAE guidelines provides a framework for building a toolbox cart that is structurally sound, safe to use, and tailored to the operational needs of the Formula and Baja SAE teams.

3 SUMMARY OF EQUATIONS AND SOLUTIONS

3.1 Brake Sub Assembly - Derek Griffith

The braking sub assembly will contain two calipers mounted on the front wheel hubs of the cart. The cables will be connected to the pull handle of the tool cart via a brake lever akin to a bicycle brake system. To calculate the braking force of the cart in motion, the following equations are required:

1. **Cart Mass (loaded):** m = 227 kg (4)
2. **Cart Velocity:** v = 2 m/s (5)
3. **Stopping Distance:** d = 3 m (6)
4. **Work:** $W = F \cdot d$ (7)
5. **Kinetic Energy:** $E_k = \frac{1}{2}mv^2$ (8)
6. **Braking Force:** $F_b = \frac{1}{2}\frac{mv^2}{d}$ (9)

Using eqn. (1), (2), (3), and (6):

$$F_b = \frac{1}{2}\frac{(227)(2)^2}{3} = 151.33\,N \approx 34\,lbf$$

This force of 34lbf is applied at the contact point between the road and the tire therefore, the actual required braking force will be higher due to the braking being applied to the disc, not the actual wheel itself. Since the wheel and brake rotor are connected, we can use the relation:

$$F_{b1} * r_1 = F_{b2} * r_2 \tag{10}$$

Using equation (7) with a wheel radius of 5 in (0.127 m) and a rotor radius of 4 in (0.1016 m):

$$151.33\,N * 0.127\,m = F_b * 0.1016\,m$$

$$F_b = 189.16\,N \approx 43\,lbf$$

It is important to note that while this is the required braking force for the cart for the velocity above, there is slipping between the brake rotor and the brake pad / piston. This coefficient of friction has been experimentally analyzed by David Gordon Wilson [10]. Based on his data and interpolating the data, we get a μ value of 0.238.

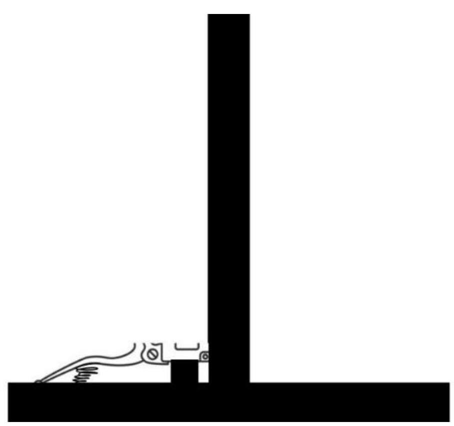

Figure 3: Compression Spring Brake Setup

For the sizing of hydraulic brakes for this special reversed brake system, the important part is the spring that will provide the constant braking force and the mechanical advantage (leverage ratio) at which the spring will act. The current mechanical advantage is estimated to be about 6 based on the available brake handle sizes. Using Shigley's Mechanical Engineering Design [11], an estimated spring ratio was calculated for the prototype:

$$k = \frac{d^4 G}{8D^3 N} = \frac{\left(\frac{1}{8} in\right)^4 (11.5 * 10^6\, psi)}{8\left(\frac{7}{8} in\right)^3 (13)} = 40.3 \frac{\text{lb}}{\text{in}} \tag{11}$$

The spring while under the brake handle was compressed by ¾ in. This means that the force acting on the brake lever is about 30.225 lbf. To calculate the pressure, we use:

$$P = \frac{F}{A} = \frac{6(30.225\, lbf)}{\frac{\pi}{4}(0.4\, in)^2} = 1443.14\, psi \tag{12}$$

With 0.4 in. being the diameter of the piston. For the brake line pressure, braided brake lines will fail at about 11 kpsi. From that pressure we get a Fos of:

$$Fos = 7.62$$

Then using the pressure applied to a $1\, in^2$ brake pad, and with the coefficient of friction value we calculated earlier, we get an applied brake force of 344 lbf to the rotor. From the earlier desired value of 43 lbf, the factor of safety for this system is:

$$Fos = 8$$

3.2 Steering Sub Assembly – Hailey Hein

The steering system is a critical subsystem since it allows maneuverability in competition pits and on uneven terrain. The worst-case scenario for steering occurs when the toolbox must make a sharp turn while fully loaded, placing large forces on the linkage and wheel assemblies.

During a sharp turn, lateral forces act on the wheels due to friction with the ground. When the cart is fully loaded, these forces increase significantly, generating high shear stresses in the steering linkage and torsional loads on the steering shaft. Additionally, the irregular surfaces in competition environments introduce impact forces, which further amplify stresses on joints and welds. These combined effects represent the worst loading the steering mechanism will experience.

Part 1) Steering Handle (Bending, Yield FoS)

Assume a cantilevered handle made from A36 steel [12] 1×1×0.125 in tube, free length L_h = 20 in, end push force F = 50 lbf (OSHA limit case). The steering handle is subjected to a concentrated load. The applied moment is calculated as:

$$\mathrm{M} = \mathrm{F}\,\mathrm{L_h} = 50 \times 20 = 1000\ \text{in} \cdot \text{lbf} \tag{13}$$

The section modulus for the 1×1×0.125 in steel tube is approximately S ≈ 0.114 in³ [13].

The maximum bending stress is obtained by dividing the applied moment by the section modulus:

$$\sigma = \frac{M}{S} \approx \frac{1000}{0.114} = 8.8\, ksi \tag{14}$$

Finally, the yield factor of safety is:

$$FoS_Y = \frac{\sigma_Y}{\sigma} = \frac{36}{8.8} \approx 4.1 \tag{15}$$

This shows that the steering handle passes with significant margin against yielding.

Part 2) Tie Rods (Tension & Buckling FoS)

Assuming two straight rods, Ø 0.25 in steel, effective length L = 24 in. Worst-case steering shock load shared by one rod: P = 200 lbf. The tie rods were evaluated under both tensile and compressive loading. First, the tensile stress is calculated from the cross-sectional area:

$$A = \frac{\pi d^2}{4} = 0.0491\ in^2 \tag{16}$$

$$\sigma_t = \frac{P}{A} = \frac{200}{0.0491} = 4.1\ ksi \tag{17}$$

The corresponding yield factor of safety is:

$$FoS_Y = \frac{36}{4.1} \approx 8.8 \tag{18}$$

Next, buckling resistance is checked using Euler's equation [14]:

$$Pcr = \frac{\pi^2 E\ I}{(K\ L)^2} \tag{19}$$

For d = 0.25 in, L = 24 in, E = 29×10^6 psi, K=1, the critical buckling load is Pcr ≈ 95 lbf.

The buckling factor of safety is:

$$FoS\ buckling = \frac{Pcr}{P} \tag{20}$$

This gives ≈1.9 for 50 lbf compression and ≈1.0 for 100 lbf. Therefore, the design is safe only if the rods are kept in tension or upgraded in diameter.

Part 3) Wheel Hub/Kingpin (Shear FoS on Axle Pin)

The vertical load on each wheel hub was estimated as 250 lbf static, doubled for dynamic loading. The applied shear force is thus P=500 lbf.

The shear area of a Ø 0.5 in pin in double shear is:

$$A = 2\left(\frac{\pi d^2}{4}\right) = 0.393\ in^2 \tag{21}$$

Shear stress is then:

$$\tau = \frac{P}{A} = \frac{500}{0.393} = 1.27\ ksi \tag{22}$$

Allowable shear for A36 is ~21 ksi, giving:

$$FoS_{\tau} \approx \frac{21}{1.27} \approx 16 \tag{23}$$

This confirms that the kingpin design is highly conservative.

By analyzing this case, the steering system can be designed with sufficient strength and rigidity to withstand peak turning loads without bending or failure, ensuring both maneuverability and durability under competition conditions.

3.3 Cabinet Volume Sub Assembly – Hailey Hein

A key requirement of the tool cart is its ability to store essential equipment used during race events. To ensure the design meets this need, we identified the minimum required factor of safety for the shelves holding typical items carried by the team.

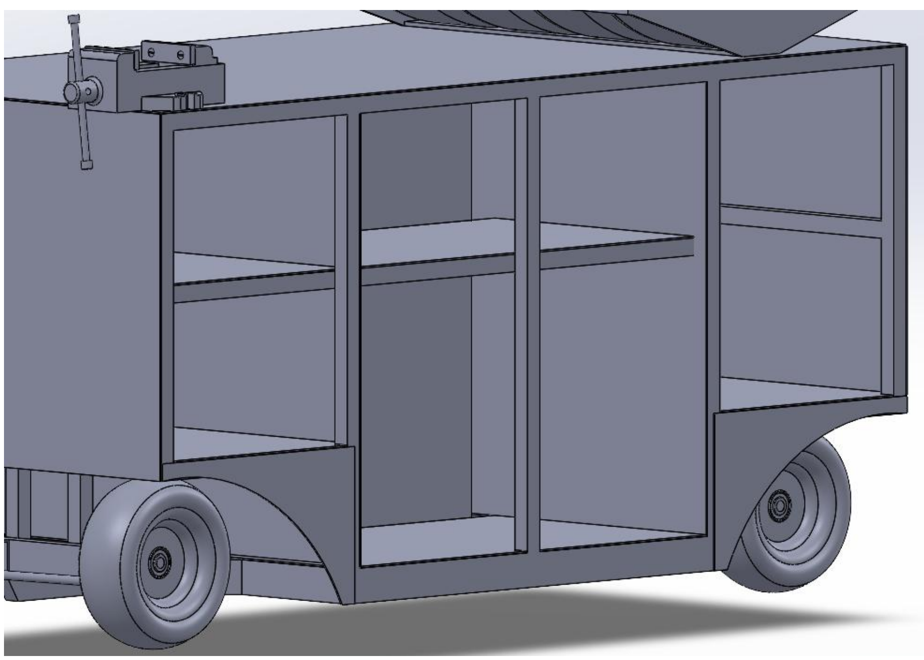

Figure 4: Modeled Available Cabinet Storage Space

When large power tools or dense metal objects are placed on a single shelf, the weight is not distributed evenly, creating high localized stresses. This condition induces bending in the shelf surfaces and shear at the attachment points. In addition, dynamic loads may occur when the cart is moved quickly or over rough ground, increasing the effective forces on the shelving supports. This represents the maximum realistic loading condition for this subsystem.

Part 1) Cabinet Shelves (panel bending, 6061-T6) [15]

The payload on the shelf is modeled as a uniformly distributed load. Per unit width:

$$w = \frac{W}{L} = 3.6\frac{lbf}{in} \tag{24}$$

The moment of inertia for a 0.090 in sheet is:

$$I = \frac{bt^3}{12} = 6.1 \times 10^{-5} in^4 \tag{25}$$

Maximum bending moment is:

$$Mmax = \frac{wL^2}{8} = 29.0\ in \cdot lbf \tag{26}$$

The bending stress is [16]:

$$\sigma = \frac{M\,c}{I} = \frac{29.0\ \times 0.045}{6.1e^{-5}} \approx 21.4\ ksi \tag{27}$$

The factor of safety is then:

$$FoS_{Y} = \frac{35}{21.4} \approx 1.6 \tag{28}$$

Part 2) Latch Pull Strength

The latch requirement is at least 100 lbf. A latch rated at 500 lbf is selected, giving:

$$FoS = \frac{RL}{100} = \frac{500}{100} = 5.0 \tag{29}$$

This provides ample safety margin.

Part 3) Drawer Slides (capacity FoS)

Each drawer is expected to carry up to Wd = 50 lbf. The drawer slides are rated at R = 100 lbf per pair. Thus, the factor of safety is:

$$FoS = \frac{R}{Wd} = \frac{100}{50} = 2.0 \tag{30}$$

All cabinet components are safe under expected loads, though the shelf is the most limiting feature with a modest FoS of 1.6. Analyzing this case ensures that the shelves and cabinet supports are designed to resist bending and shear under concentrated tool loads, preventing failure or deformation that could compromise tool storage or safety.

3.4 Frame Sub Assembly – Hailey Hein

The frame is the backbone of the toolbox and must support all other subsystems. The worst-case scenario for the frame occurs when the cart is fully loaded and moved across uneven terrain, which induces both static and dynamic stresses.

When fully loaded, the frame supports the combined weight of all tools, shelves, wheels, and attachments. As the cart travels across uneven terrain, dynamic loads from bumps and impacts amplify the static weight, leading to bending and torsional stresses in the frame members. Welded joints and high-stress corners are particularly vulnerable under this condition [17]. The frame must also resist potential twisting when one wheel is raised higher than the others on rough ground.

Static Structural Analysis (Steel Frame):

The frame must support the full toolbox and equipment weight while withstanding uneven terrain. A baseline static analysis and redesign checks were performed for critical frame members.

- Material: A36 Steel
- Yield Strength (σ_y): 36,000 psi
- Modulus of Elasticity (E): 29 × 10^6 psi
- Cross-Section: 1" × 1" & 1” x 2” square tubing, 0.125" wall thickness
- Span (L): 60 in
- Load (F): 500 lb (toolbox + equipment)

Part 1) Baseline member (1×1×0.125 in A36 tube):

For a simply supported beam with a central load:

$$Mmax = \frac{F\,L}{4} = 7500\ in \cdot lbf \tag{31}$$

The stress in the 1×1 tube is (1×1×0.125) = 0.114 in^3.

$$\sigma = \frac{M}{S} = \frac{7500}{0.114} = 65.8\ ksi \tag{32}$$

Thus:

$$FoS_{Y} = \frac{36}{65.8} \approx 0.55\ (Fail) \tag{33}$$

Redesign Options:

Increasing tube depth or adding mid-span support improves strength:

1×2×0.125” tube:

$$S \approx 0.332\ in^3 \rightarrow \sigma \approx 22.6\ ksi \rightarrow FoS \approx 1.6 \tag{34}$$

1×3×0.125” tube:

$$S \approx 0.633\ in^3 \rightarrow \sigma \approx 11.8\ ksi \rightarrow FoS \approx 3.0 \tag{35}$$

Add mid-span crossmember:

$$M = 3750\ in \cdot lbf \rightarrow \sigma \approx 32.9\ ksi \rightarrow FoS \approx 1.1 \tag{36}$$

Part 2) Deflection Check (1×2×0.125 option):

Deflection is calculated as:

$$I \approx S\,c = 0.332 \times 1.0 = 0.332\ in^4 \tag{37}$$

$$\delta max = \frac{FL^3}{(48\ E\ I)} = 0.23\ in \rightarrow acceptable \tag{38}$$

This is acceptable. The 1×3 tube reduces deflection further to 0.12 in. The baseline frame fails underloading without support. A 1x2" or 1×3" tube with reinforcement was found to achieve safe stress levels and acceptable deflection. By evaluating this case, the frame design can be validated to ensure it withstands both static and dynamic stresses with an adequate factor of safety, providing structural integrity and preventing failure during competition use.

3.5 Caster Sub Assembly – Haoran Li

The caster assembly was analyzed under the worst-case loading conditions to ensure safe operation when the toolbox cart traverses rough terrain and encounters impact loading. The analysis focused on two primary components: the rim (steel) and the tire (heavy-ply rubber).

Conservative Assumptions:

- Total cart + payload = 500 lbf (consistent with Sec. 3.4).
- Worst case: one front caster carries full impact load, V = 500 lbf.
- Wheel radius: R=5 in.
- Tire width: b_t = 2.0 in.
- Effective tire thickness: t = 1.0 in.
- Friction coefficients: μ_x = 0.6 (longitudinal), μ_y = 0.7 (lateral).
- Materials: Rim = A36 steel (σ_y = 36 ksi); Tire = heavy-ply rubber (P_{allow} = 300 psi, τ_{allow} =300 psi).

Part 1) Steel Rim

The rim was modeled as a thin ring subjected to a quarter-point bending moment:

$$M_{max} = \frac{VR}{4} = \frac{500 * 5}{4} = 625 \tag{39}$$

For rim cross-section width br=2.0 in, thickness hr=0.375 in:

$$S = \frac{b_r * h_r^2}{6} = \frac{2.0 * 0.375^2}{6} = 0.0469\ in^3 \tag{40}$$

$$\sigma_b = \frac{M_{max}}{s} = \frac{625}{0.0469} = 13.3\ ksi \tag{41}$$

Factor of safety:

$$\mathrm{FoS_{rim}} = \frac{\sigma_y}{\sigma_b} = \frac{36}{13.3} = 2.7 \tag{42}$$

Part 2) Heavy-ply Rubber Tire

(a) Compressive stress

Contact length a_c= 1.0 in:

$$p = \frac{V}{b_t * a_c} = \frac{500}{2.0 * 1.0} = 250\ psi \tag{43}$$

$$\text{FoS}_p = \frac{P_{allow}}{p} = \frac{300}{250} = 1.2 \tag{44}$$

(b) Shear stress

Braking load:

$$F_x = \mu_x * V = 0.6 * 500 = 300\ lbf \tag{45}$$

$$\tau_x = \frac{F_x}{b_t * t_t} = \frac{300}{2.0 * 1.0} = 150\ psi \tag{46}$$

$$\text{FoS}_{\tau x} = \frac{300}{150} = 2 \tag{47}$$

Lateral load:

$$F_y = \mu_y * V = 0.7 * 500 = 350\ lbf \tag{48}$$

$$\tau_x = \frac{F_y}{b_t * t_t} = \frac{350}{2.0 * 1.0} = 175\ psi \tag{49}$$

$$\text{FoS}_{\tau x} = \frac{300}{175} = 1.71 \tag{50}$$

3.6 Power Supply Sub Assembly – Yanbo Wang

The power supply system is the core of SAE Toolbox reliability. It must provide continuous electricity to chargers, lights, and auxiliary electronics during competition. A 2500 W inverter generator has been selected as the primary power source. To ensure structural safety, a worst-case load analysis was performed on the generator tray, mounting bolts, hold-down straps, and power cables.

The worst-case scenario occurs when:

1. The generator or batteries experience 3g vertical shock during transport or use, combined with longitudinal and lateral inertial loads.
2. Mounting bolts and straps must resist concentrated impact forces.
3. The tray is subjected to bending stress.
4. Power cables carry maximum current and must be checked for thermal safety margin.

Static Structural Analysis (Power Supply Subsystem)

- Generator weight: 25 kg
- Alternative battery weight: 30 kg
- Shock factor: 3g vertical
- Tray span: 0.30 m
- Material: A36 steel (yield strength 250 MPa)
- Bolts: M8, Grade 8.8 steel
- Cables: AWG 10 copper wire rated 40

Part 1) Generator Tray – Beam Bending

$$P = mgn_g = 25 \times 9.81 \times 3 = 736N \tag{51}$$

$$M_{max} \frac{PL}{4} = 55.2N \cdot m \tag{52}$$

Section modulus:

$$S = 6 \times 10^{-6} m^3 \tag{53}$$

Stress:

$$\sigma = \frac{M}{S} = 9.2MPa \tag{54}$$

FoS:

$$FoS = \frac{\sigma_y}{\sigma} = \frac{250}{9.2} \approx 27.2 \tag{55}$$

Result: Very safe.

Part 2) Mounting Bolts – Shear

Total load 736 N shared by 4 bolts:

$$V = \frac{736}{4} = 184N \tag{56}$$

M8 bolt stress area:

$$A_s = 36.6mm^2 \tag{57}$$

Shear stress:

$$\tau = \frac{V}{A_s} = 5.0MPa \tag{58}$$

Allowable shear ≈ 290 MPa.

$$FoS = \frac{290}{5.0} \approx 58.0 \tag{59}$$

Result: Extremely safe.

Part 3) Tie-down Straps – Tension

Two straps share the load:

$$T = \frac{736}{2} = 368N \tag{60}$$

Each strap rated at 400 N.

$$FoS = \frac{400}{368} = 1.09 \quad (61)$$

Result: Marginal, at risk of overload. Straps should be upgraded or reinforced.

Part 4) Power Cables – Thermal Margin

Peak current: 20 A vs. AWG 10 rating of 40 A.

$$FoS = \frac{40}{20} = 2.0 \quad (62)$$

Result: Safe.

Part 5) Battery Tray (Alternative Case)

Battery mass: 30 kg.

$$P = 30 \times 9.81 \times 3 = 883N \quad (63)$$

$$M = 66.2N \cdot m, \qquad \sigma = 11.0MPa, \qquad FpS = \frac{250}{11.0} = 22.7 \quad (64)$$

Result: Safe but adds weight and cost.

The analysis of the power supply subsystem shows that the tray and bolts have a very high safety margin, indicating strong structural performance. The cables are also determined to be safe under the conditions evaluated. However, the hold-down straps represent a critical weak point, with a factor of safety of approximately 1.1. These straps should be upgraded to higher-strength alternatives or increased in number to improve reliability. While the proposed battery option is structurally safe, it is both too heavy and too costly, and therefore not recommended. Overall, the power supply subsystem is structurally sound under worst-case conditions, but reinforcing the straps is essential to ensure robust and reliable operation.

3.7 Total Summary Table - All

Table 1 provides a comprehensive summary of the calculated minimum factors of safety (FoS) for all major sub-systems and their components within the project. This table consolidates the critical design data into a single reference, making it easier to evaluate the overall structural and functional integrity of the system. Each entry identifies the responsible owner, sub-system, specific part, and the governing load case scenario, along with the material used and the method employed to calculate the FoS. By including these details, the table highlights both components with substantial safety margins and those approaching critical limits where design improvements or reinforcements may be necessary. This information allows for quick identification of potential weak points and serves as a guide for prioritizing future testing, analysis, and design modifications.

Table 1: Summarized Calculations

Owner	Sub-System	Part	Load Case Scenario	Material	Method of Calculating FOS	Minimum FOS
Derek Griffith		Brakes				
		Part 1 – Rotors	43 lb braking force	304 Stainless	Stopping distance	8
		Part 2 – Brake lines	1443.14 psi	304 Stainless	Brake line yield	7.62
Hailey Hein		Steering				
		Part 1 – Handle	50 lbf end load, L=20 in (worst-case push)	A36 steel	Beam bending, yield FoS	4.1
		Part 2 – Tie Rods	200 lbf axial (shock), tension	A36 steel	Axial stress yield, FOS	8.8
		Part 2 – Tie Rods In compression	50–100 lbf compression, L=24in	A36 steel	Euler buckling, FOS	1.0-1.9
		Part 3 – Axle/Kingpin	500 lbf peak wheel reaction, double-shear	A36 steel	Shear stress FoS	16
Hailey Hein		Cabinet Volume				
		Part 1- Cabinets shelves	100 lbf distributed, span 27.8 in	6061-T6	Strip bending, yield FoS	1.6
		Part 2 – Cabinet Latch	100 lbf required pull	Steel	Capacity ratio RL/100	5 (for 500 lbf latch)
		Part 3 – Toolbox Drawers	50 lbf contents per drawer	6061-T6	Capacity ratio R/W	2 (for 100 lbf-rated slides)
Hailey Hein		Frame				
		Part 1 – 1x1" Steel	500 lbf centered, span 60 in, un-braced/un-supported	A36 steel	Beam bending, yield FoS	0.55
		Part 2 – 1x2" Steel	500 lbf centered, span 60 in, un-braced/un-supported	A36 steel	Beam bending, yield FoS	1.6
Haoran Li		Casters				
		Part 1 – Rim	500 lbf vertical impact load on single caster, wheel radius = 5 in	Steel	Yield stress method	2.7
		Part 2 – Tire	500 lbf vertical load with braking (300 lbf) and lateral (350 lbf) forces	Heavy-ply Rubber	Stress comparison method	1.2
Yanbo Wang		Power Supply				
		Part 1 - Generator Tray	25 kg @ 3g vertical, span 0.30 m	A36 steel	Beam bending	27.2
		Part 2 - Mounting Bolts	25 kg @ 3g, shared by 4 × M8 bolts	Grade 8.8 steel	Shear stress	58.0
		Part 3 - Tie-Down Straps	25 kg @ 3g, 2 straps	Nylon/steel	Tension capacity ratio	1.09
		Part 4 - Power Cables	20 A peak vs 40 A ampacity	Copper	Thermal ampacity margin	2.0
		Part 5 - Battery Tray (alt)	30 kg battery @ 3g	A36 steel	Beam bending	22.7

4 FLOW CHARTS AND OTHER DIAGRAMS

In line with the rubric's requirement for diagrams, the CAD drawings serve as visual system diagrams to explain the physical layout and functionality of the design. Figure 14 provides the dimensioned drawing of the frame, which functions like a flow diagram for fabrication, mapping dimensions and material placement to guide manufacturing. These diagrams ensure that the design's functionality and integration of subsystems are clearly communicated for planning and construction.

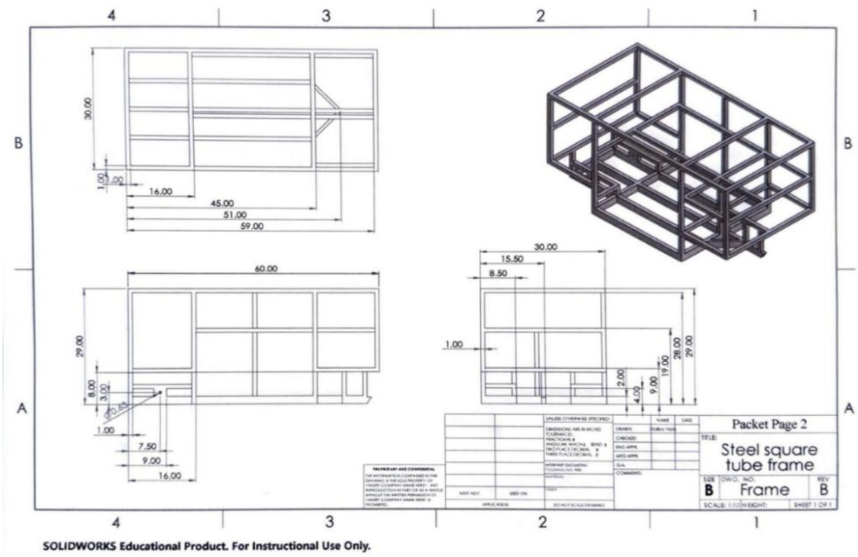

Figure 5: Steel Frame CAD Part Drawing

Figure 6 illustrates the complete brake fluid routing within the system. Beginning at the brake fluid reservoir, the diagram traces the path of hydraulic fluid as it travels through brass lines to a T connector, where the flow divides to feed both brake calipers. This flow chart clarifies the functional sequence of components and shows how hydraulic pressure is distributed evenly between the two brake assemblies, ensuring balanced braking performance. By visualizing the connections and flow paths, this diagram helps identify potential failure points, maintenance access points, and areas where future upgrades or redundancy may be incorporated.

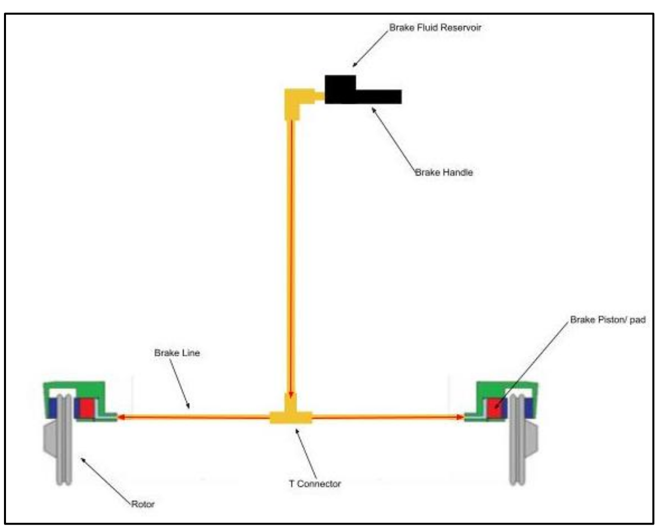

Figure 6: Brake Fluid Flow Chart

Figure 7 presents a detailed overview of the electrical power distribution architecture. Power originates from the battery pack and passes through a main fuse for overcurrent protection before reaching a contactor with pre-charge circuitry. From there, current flows to the DC bus, which distributes high-voltage power to the motor controller and a DC-DC converter. The converter steps down the voltage to 12 V for auxiliary loads such as sensors, ECU, fans, and lighting. An emergency stop switch is also highlighted in the diagram to demonstrate how the system can be rapidly isolated in case of fault or maintenance. This flow chart not only clarifies how electrical power is managed but also shows the functional hierarchy of safety devices, power conversion, and control elements.

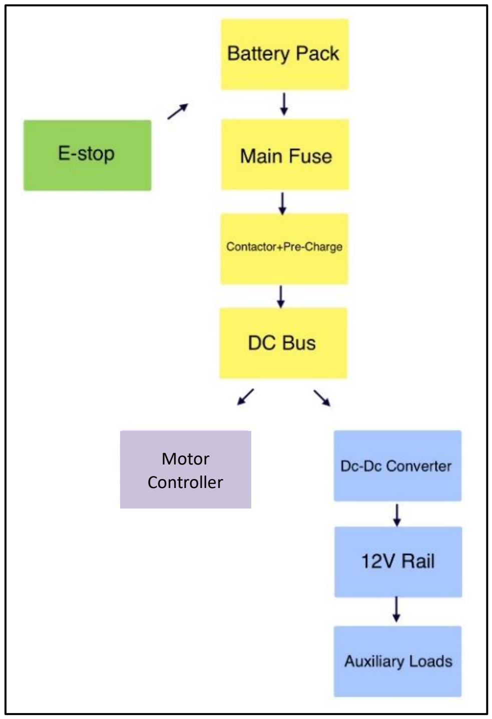

Figure 7: Electrical Power Distribution Flow Chart

Using flow charts and diagrams throughout the report ensures that complex systems are presented in a clear, accessible, and organized manner. These visual tools make it easier to understand the sequence of operations, interconnections, and critical components of each subsystem. For engineering teams, they serve as quick-reference guides during fabrication, assembly, troubleshooting, and testing. Incorporating these visuals also strengthens the communication of design intent to stakeholders, from faculty reviewers to shop technicians, ensuring consistency between design, analysis, and construction. Ultimately, the inclusion of flow charts and diagrams bridges the gap between conceptual design and practical implementation, helping ensure reliability, safety, and ease of maintenance for the final product.

5 MOVING FORWARD

While significant progress has been made in defining requirements, developing the design, and validating major subsystems, several areas still require additional work and analysis to fully verify that the toolbox cart will meet all customer and engineering requirements.

Although preliminary Finite Element Analysis (FEA) has been conducted on the frame, more detailed simulations are needed to confirm long-term durability under repeated loading. Fatigue life calculations should be re-run on weld joints and high-stress members to ensure the cart will not fail under the cyclic conditions of repeated transport over uneven terrain. Additional sensitivity analyses are also required to verify stability under varied loading scenarios, including off-center tool storage and tire placement.

The steering and braking concepts have been prototyped and show promise; however, complete validation has not yet been achieved. The disc brake system must be tested under full load to ensure it can bring the cart to a stop and maintain position on a 5° incline, in accordance with OSHA ergonomic safety limits. Braking torque calculations should be revisited using measured friction coefficients for the chosen components to confirm the design meets performance requirements. Similarly, the steering geometry analysis should be refined to ensure that the projected 38-inch turning radius is consistently achievable under real-world conditions.

The cart is expected to be operable by a single person with no more than 50 pounds of applied force, but this requirement has only been verified theoretically. Physical testing with a fully loaded prototype is necessary to confirm rolling resistance assumptions and validate ergonomic calculations. If performance deviates from expectations, adjustments to caster size, bearing type, or frame reinforcement may be required to meet operability goals.

The drawer and storage subsystems also require additional validation. While lockable compartments and drawers have been designed, further testing is needed to ensure that they remain secure under vibration and incline conditions. An inclined-plane test should be conducted to verify that the drawer locking mechanisms can withstand sudden stops and shock loads. Additionally, latch pull strength calculations should be compared against manufacturer specifications to confirm compliance with the 100-pound minimum requirement.

A final decision to use a 2500W inverter generator as our power supply has been reached. This generator will supply power for tool operation and battery charging during competition, providing a reliable and portable energy source that eliminates the need for grid access. With this, the next steps include analyzing thermal loading, wiring safety, and overcurrent protection in accordance with NFPA. Verification of the generator's output capacity against SAE team tool requirements is necessary to prove sufficient loads.

Finally, several validation tests remain to be conducted to close the loop on subsystem performance. These include a hill stopping and holding test to evaluate brake reliability and drawer locking under incline conditions, a mock tech inspection to simulate race-day scenarios and confirm tool accessibility and organization, and a loaded weight test with a 500-pound payload to verify single-user operability, caster strength, and overall maneuverability during transport and trailer loading.

In summary, while the team has developed a comprehensive and promising design, further structural calculations, braking torque analyses, ergonomic validation, electrical safety checks, and full prototype testing are required to fully convince the client, instructors, and end users that the cart will perform reliably. Completing these outstanding tasks in the upcoming semester will ensure that the final product meets all safety, performance, and usability requirements for the NAU SAE Formula and Baja teams.

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