



LEAN CONTROLLED WHEELCHAIR

CRITICAL DESIGN REVIEW

ME 487 / ECE 480 / CSE 480

Lean Controlled Wheelchair
Tristan Phillips, Caleb Hottes, Charles Friley, Braxton McBride,
Daniel Titus, Mason Nelson, Dillon Schieve
Dan McEntire M.D., Challenged Athletes Foundation
Dr. Hannah O'Hern, Brian Higgins

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1. Executive Summary:

The purpose of this project is to solve a clear limitation in current sports wheelchairs. Right now, the athlete must use their hands to move the chair, which directly interferes with passing, shooting, and ball handling during play. Our project aims to modify an existing sports wheelchair so that it can be controlled through leaning, allowing the user to keep both hands available while still moving quickly and safely. This is intended to improve athletic performance in wheelchair sports like basketball and pickleball.

Our current design uses two tightly connected systems. The first is an electrical system that senses the user's lean, processes that input, and controls power delivery. The second is a mechanical system that mounts the drive components to the existing wheelchair and transfers power to the wheels. Together, these systems are intended to provide hands-free movement while prioritizing the safety, stability, and controllability of the chair. The current prototype concept uses a pressure-sensor seat array, onboard control electronics, dual motor controllers, two BLDC drive motors, a belt-reduced drivetrain, a dog coupling quick-disconnect wheel system, and an onboard 48 V battery.

Our planned design adapts an existing sports wheelchair by adding drive, sensing, and control systems in a way that preserves the chair's agility, stability, and familiarity to the user. The current concept also preserves limited manual repositioning and transportability, prioritizes durable mounting for repeated impacts, and avoids choices that would make the wheelchair unnecessarily bulky, unsafe, or difficult to maintain.

The main risks to the project are purchasing, integration, and overall system complexity. Major components must work together reliably for the wheelchair to perform as intended, and this intricacy creates risk in both control and mechanical implementation. Custom fabricated parts also introduce some risk because they must fit correctly, withstand repeated impacts, and remain practical to assemble and maintain. In addition, the cost of motors, batteries, sensors, and supporting hardware increases the overall cost of the system and may affect long-term feasibility depending on future production goals.

One expected limitation of the lean-controlled sports wheelchair is that athletic body motion and motion-command input are closely linked. For example, reaching laterally to intercept a basketball pass may also create a lean input that commands wheelchair motion. The current design philosophy accepts that the user will need practice to adapt to this new style of play, while future control tuning may reduce unintended motion through calibration, deadbands, and command shaping.

The design analysis indicates that the selected architecture can meet the major performance, runtime, packaging, and safety requirements under the stated assumptions, but prototype testing is still required to validate real-world control behavior, structural durability, and battery runtime.

Overall, this project presents a meaningful opportunity to improve wheelchair athletics by freeing the user's hands without giving up safety or control. The current design direction shows a practical path toward a working system that could improve competitive play, expand what is possible for wheelchair athletes, and provide clear value to the user and other stakeholders.

2. Problem Statement:

Dan McEntire is sponsoring the development of a prototype lean-controlled sports wheelchair for his son, Parker, a wheelchair athlete with lower-limb paralysis who competes in court sports such as basketball and pickleball. In these sports, the athlete must normally use their hands to propel and maneuver the wheelchair, which directly interferes with passing, ball handling, and other athletic actions during play. This creates a significant limitation because movement and gameplay both depend on the athlete's upper body. The project therefore seeks to develop a sports wheelchair that uses controlled upper-body lean to command motion while remaining stable, durable, and practical for competitive use. The central challenge is to provide hands-free maneuvering without sacrificing the safety, responsiveness, and on-court usability required in wheelchair athletics.

3. Stakeholder Analysis:

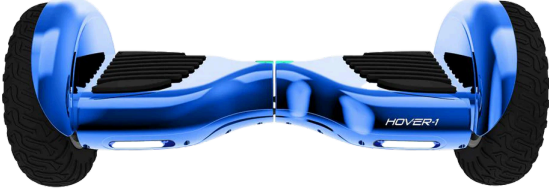
- Operator
 - Aspects of the solution that help
 - The wheelchair is intended to provide hands-free mobility so the operator can use both hands for athletic tasks such as passing, shooting, and ball handling.
 - The system is intended to remain stable, responsive, and safe during rapid movement and expected sport contact.
 - The lean-based control system is intended to feel intuitive and allow precise repositioning during play.
 - Aspects of the solution that hinder
 - The operator will need time and practice to adapt to the control system and develop effective use during gameplay.
 - The system requires sufficient trunk control and core strength to use lean-based input effectively.
 - The added drive components increase system weight and may reduce convenience during manual, unpowered use.
- Family / Guardian
 - Aspects of the solution that help
 - Includes safety-focused features intended to reduce the risk of injury during athletic use
 - Supports the operator's ability to participate more fully in sports and recreation
 - Lower cost than alternative motorized wheelchairs
 - Aspects of the solution that hinder
 - May still create concern due to the added speed, power, and system complexity
 - Will require trust in the reliability of the control, power, and safety systems
 - May require family time, effort, or oversight for transport, charging, maintenance, and troubleshooting
 - Could require future adjustment as the operator grows or as user needs change
- Caregiver / Transporter
 - Aspects of the solution that help

- Can be moved, loaded, and stored without requiring specialized tools
 - Retains some manual maneuverability when unpowered, allowing repositioning during transport or setup
 - Designed to fit through standard doorways or be placed in a transport configuration for everyday handling
 - Allows setup and takedown in a practical amount of time for travel to games, practices, or other events
- Aspects of the solution that hinder
 - Added motors, battery, and structural components increase overall weight and handling difficulty
 - Charging, loading, and securing the wheelchair may be more involved than with a standard sports wheelchair
 - Transport and storage may become more difficult due to the added size, mounted hardware, or fragile components
 - A more complex system may require the caregiver or transporter to learn basic operating and troubleshooting procedures
- Other Athletes and Teams
 - Aspects of the solution that help
 - Is intended to interact safely with other players during normal sport contact and close-quarters play
 - May expand team strategy by allowing a player to move while keeping both hands available for athletic tasks
 - Could broaden future design possibilities for adaptive sports equipment if the concept proves effective
 - Is being designed with stability, controllability, and predictable motion in mind, which benefits everyone on the court
 - Aspects of the solution that hinder
 - Could introduce unfamiliar movement behavior that other players and teams must learn to anticipate
 - May create fairness concerns if the system provides performance advantages beyond what is considered acceptable for play
 - Added weight, speed, or powered motion could increase risk during collisions if not properly limited and controlled

- Teams, coaches, or officials may be hesitant to accept the wheelchair until its safety and on-court behavior are well understood
- Maintainer
 - Aspects of the solution that help
 - Can create opportunities for inspection, repair, adjustment, and long-term technical support
 - Uses organized electrical and mechanical subsystems that can be designed for easier diagnosis and part replacement
 - May use commercially available components that are easier to source than fully custom systems
 - A well-documented design can make maintenance and troubleshooting more practical over time
 - Aspects of the solution that hinder
 - Combines electrical, control, and mechanical systems, making maintenance more complex than for a standard sports wheelchair
 - Custom fabricated parts may be difficult to replace quickly if they are damaged
 - Technicians may need project-specific knowledge to safely diagnose faults or service the system
 - Added wiring, sensors, controllers, and drivetrain components increase the number of possible failure points
- Sponsors
 - Aspects of the solution that help
 - Addresses a clear user need by supporting hands-free athletic mobility
 - Produces design knowledge, testing results, and prototype experience that may be valuable beyond this single wheelchair
 - Can demonstrate meaningful progress toward a novel adaptive sports solution
 - Aspects of the solution that hinder
 - Requires significant time, funding, fabrication effort, and technical coordination
 - May not fully satisfy every performance goal within the limits of a single academic project cycle
 - Prototype-level solutions may still require further refinement before broader real-world adoption

4. Benchmarking:

4.1 Existing Lean or Balance-Based Mobility Systems

Hoverboard Hover-1 Titan - HY-TTN [1] Hover-1™ Titan Hoverboard Hover-1 Rideables	Pros • Typically use dual motors for independent wheel movement. • Small form • Utilize two gyro sensors for determining board angle (relative to the ground)	Cons • The motors are compact due to them being built into the wheels. May not be possible for our solution. • Not very powerful
		
The board works by determining lean angle using two gyro sensors. The processor then spins one or both of the wheels forward or backwards as an attempt to return the board to being level. This causes the entire hoverboard to move forward or backwards. So long as the user continues to lean forward or backwards, the board can not return to being level and therefore continues moving. If our solution uses two primary wheels, then the dual motor and dual sensor design could be applicable.		

iBot Mobility System [2] 2kerr.com/en/balance-wheelchairs/ibot-duo-balance-wheelchair	Pros • Two independently moving wheels • Self Balancing • Lightweight • Foldable • Sharp turning radius	Cons • Uses handlebar to steer similar to a segway
		
This wheelchair is designed to be compact and agile for both daily use and active use. The biggest drawback is that it is not turned using lean controls, but instead by turning a handle bar. Given our application, we would need to incorporate a method of detecting lean angle, as the handlebar does not meet the required specifications.		

Fully Lean Controlled Wheelchair OMEO Evolution 1 [3] Omeo Technology - Omeo Technology	Pros • Controlled by shifting bodyweight around • Optional manual joystick control • Self-Leveling • Relatively fast (12mph)	Cons • Expensive • Relatively large • Heavy (154lbs)
		
The OMEO is an example of a self balancing wheelchair. It only has two points of contact with the ground and uses motors to maintain stability similar to the hoverboard. It determines where and when to move using an Active Seat Control. When the user shifts their bodyweight around by leaning the sensors determine the desired direction. It additionally has a joystick for manual control during times where the lean controller is undesirable. It has a plethora of quality of life features (thick cushion, compartments, usb ports, etc.) that add extra bulk and weight. The all-terrain tires and bulky wheels also add to its weight. We will need to use much smaller, lighter wheels, tires, and have less unnecessary features to keep weight low. Overall, this is a good reference for designing a self balancing wheelchair.		

4.2 Drivetrain and Actuation Options

Planetary Gearbox NEMA 23 Planetary Gearbox 5:1 [4] Planetary Gearbox 5:1 - Spark Robotic	Pros • High torque density (Large torque output for its compact size) • Input and output shafts are on the same axis (reducing size and weight) • Highly efficient (typically has 95% or greater) • Smooth • Reliable	Cons • Expensive compared to other gearboxes • More maintenance required • More complex maintenance • Can trap heat internally under heavy load
		
A planetary gearbox can be more complicated and expensive to use than other simpler gearboxes, but will likely provide a better result. They are efficient, compact, reliable, and provide great performance. Given that our design will require a good amount of torque, this is likely a good gearbox for our application. Additionally, it offers a small formfactor, and relatively low weight which is important due to our limited space to store components. Finally, it is efficient, which would decrease the motor's necessary power draw.		

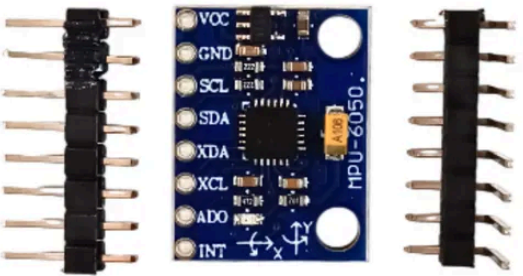
Servo Motor ProTek RC 170TBL "Black Label" PN: PTK-170TBL [5] ProTek RC 170TBL "Black Label" High Torque Brushless Servo [PTK-170TBL] - AMain Hobbies	Pros • Movements are made more often leading to more precision and accuracy • Constant torque across the entirety of the speed range • High maximum speeds • Smooth motion • Power efficient (Only draws power when needed)	Cons • Expensive compared to other motors • More complex tuning required • More components than other motors (Likely will need more maintenance)
		
Servo motors provide constant torque, linear power, great performance, and are power efficient, but cost more. The servo motor makes for a great quality motor, but will require more effort and resources. Given that our wheelchair will need constant, reliable torque, this is likely a good motor for our application. It is also capable of high speeds which will likely be necessary for going down court. The motor is also power efficient, which could decrease our necessary battery capacity.		

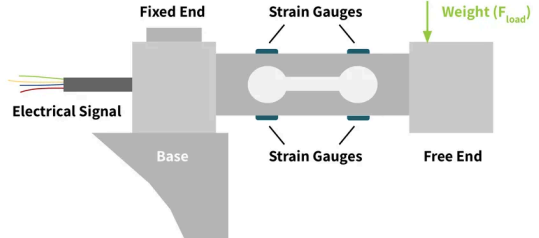
Stepper Motor NEMA34 Integrated Closed Loop Stepper Motor UIM344-NEMA34B [6] NEMA34 Integrated Closed Loop Stepper Motor - Amazon	Pros • Great torque at low speeds • Relatively cheap • Extremely precise and controllable • Less maintenance	Cons • Poor torque at higher speeds • Poor top end speed • Produces a lot of vibrations • Poor energy efficient (Constantly draws power)
		

A stepper motor is cheap and simple to use, but does not offer as great of performance as a servo motor. It does however offer greater torque and precision in low speed scenarios. This would make it exceptional at low speed movements and responsiveness, but would lack on the top end. This may require us to get a relatively large motor, increasing power draw, size, and weight.

4.3 Lean / Posture / User-Intent Sensing Options

Smart Chair to Monitor Sitting Posture	Pros	Cons
[7] https://www.mdpi.com/1996-1944/16/13/4838 	• Capable of sensing posture in chair • Low cost of implementation • No need to modify chair chassis for sensor implementation • Capacitive controlled to distinguish levels of pressures on sensors	• Prototype system • Intended for separate capacity • Unknown responsiveness to movement • Usage of microcontrollers was used for each sensor • When tested some subjects did not set off sensors in positions where others did
This item was a prototype of a chair designed to sense the posture of an office worker through the use of interdigitated capacitive embroidered sensors [7]. This item may lead to a potential solution for the sensing of body weight distribution on the chair to control the motors without modifying the chassis. From the pros and cons this item has a proof of concept of properly sensing a person's posture and position in the chair however depending on the person some sensors may not be in contact which causes misreadings. This benchmark provides a concept of modifying the seat and backrest of the chair to approximate positional movement without the need to remove stabilization wheels.		

Gyroscope/Accelerometer MPU-6050 IMU Module [8] https://demonstrations.wolfram.com/Gyroscope/	Pros • Senses total rotational movement in all directions • Used in self correctional devices like drones • Inexpensive • Commonly used for motion detection	Cons • Implementation may cause mass modification to chair depending on implementation method • Highly sensitive to impacts • Known to have drift issues over time • Unpredictable fluctuations
 Gyroscopes and Accelerometers are commonly used for devices like drones and segways to monitor acceleration and rotational movement for correcting movement for stability. The item is an option for a sensor type to use to account for lean controlling the chair. It would be useful for sensing movement in the chair, however, depending on how it is implemented mass modification of the chair may be needed to implement along with calibration.		

Strain Gauge Load Cell [9] https://www.botasys.com/post/strain-gauge-load-cell	Pros • Senses Weight Distribution and generates electrical signal • Several kinds for different applications • Precise electrical signal	Cons • Would require several to sense several locations for accurate control • Requires at least a few grams or milligrams of force to generate a signal
 This is another sensor type that can be used to detect weight distribution and may be an option for detecting weight distribution in a seat. The sensor works by sensing weight at a point and generating a resistance based off of it, however due to the type of sensor for our project several would need to be used to properly detect weight shifting in real time. This option is one that we will need to look more into to see if this is a better sensor option than some other options on the market.		

5. Needs

Table 1: Needs, Descriptions, and Justification for Lean Controlled Wheelchair

Need ID	Description	Justification	Required
1	The wheelchair must be safe to operate at all times.	The user will participate in high-contact, fast-paced court sports where collisions, rapid acceleration, and tipping are common.	Y
2	The wheelchair must fail safely in the event of power or control system failure.	Loss of power or sensor failure during high-speed athletic activity could cause serious injury.	Y
3	The wheelchair must respond quickly and intuitively to hands-free user input.	The core purpose of the design is to free the user's hands for intuitive athletic performance.	Y
4	The wheelchair movement must be primarily controlled by leaning.	The client explicitly requested posture-based control to eliminate hand propulsion during sport.	Y
5	The wheelchair must be highly stable and resistant to tipping.	Stability is critical in wheelchair athletics to prevent loss of control.	Y
6	The wheelchair must stay upright after collisions to the wheelchair body and user.	Collisions with other athletes are expected in wheelchair sports. The system must maintain upright orientation or recover quickly to prevent interruption of play.	Y
7	The wheelchair must withstand repeated impacts without functional degradation.	Collisions are expected under normal use.	Y
8	The wheelchair should allow limited manual control or override.	Manual control provides redundancy in case of power failure and allows repositioning in non-athletic settings where powered motion may not be	Y

		appropriate.	
9	The wheelchair must fit through, or be easily modified to fit through, standard doorways and common indoor spaces.	Although athletics are the primary function, basic accessibility is required.	Y
10	The wheelchair must operate long enough to support a full athletic session.	The client expects the system to function for the duration of practices or games without interruption.	Y
11	The wheelchair must be suitable for competitive indoor court surfaces.	Basketball and pickleball courts have traction, marking, and floor protection constraints.	Y
12	The wheelchair must not damage regulation court surfaces.	Gym facilities are stakeholders. Damage to finished court surfaces could prevent use in competitive venues and reduce acceptance by organizations.	Y
13	Seating must include pressure-relief features to reduce risk of pressure sores.	The user has lower-limb paralysis and may have reduced sensation and circulation. Extended seated use during athletics increases risk of pressure injury, which is a known medical concern for wheelchair users.	Y
14	The wheelchair remains intuitive to set up and take down without the use of tools.	The user and caretaker must independently transport and store the wheelchair.	Y
15	The wheelchair must have integrated battery power.	It is essential that the wheelchair uses a battery to move.	Y
16	The wheelchair may allow adjustable seating or scalability for different users.	The user is young, and will grow more. Adjustability allows for use when the user is larger.	N

6. Background Information and Research

6.1 Sports Wheelchair Design:

Sports wheelchairs differ significantly from standard daily-use wheelchairs because they are optimized for agility, stability, and rapid maneuvering during athletic play. Their geometry, materials, and seating systems are selected to improve court performance rather than long-term comfort or general indoor accessibility.

One of the most noticeable differences is wheel geometry. Standard wheelchairs commonly use little to no camber, while sports wheelchairs often use rear wheel camber angles in the range of approximately 12 to 20 degrees [10]. This wider wheel stance improves lateral stability and supports quicker turning, which is valuable in sports such as wheelchair basketball. A tradeoff of this wider configuration is that it can increase the overall width of the chair, which reduces compatibility with narrow doorways or other indoor spaces [10].

Sports wheelchairs also commonly use large rear wheels, typically around 25 to 26 inches in diameter [11]. A larger wheel covers more ground per rotation, which can benefit sustained movement, while a smaller wheel generally has lower rotational inertia and can feel more responsive during acceleration. In addition, sports wheelchair tires tend to have a lower profile than those used on standard wheelchairs, which contributes to quick handling on smooth court surfaces. The wheel structure is also designed for contact-heavy use. Spokes are often reinforced, and protective wheel covers may be added to reduce damage during collisions or entanglement with other chairs [11]. Hand rims are shaped for quick and forceful contact during play and are typically more durable than comfort-oriented daily-use rims, although this often comes at the expense of long-term comfort [12].

Frame and seating materials are also selected with athletic performance in mind. Sports wheelchairs are commonly constructed from aluminum, titanium, or, less frequently, carbon fiber [13]. Aluminum is widely used because it offers a favorable combination of strength, low weight, stiffness, cost, and manufacturability. Titanium is often used in higher-end chairs because it offers excellent strength-to-weight ratio, fatigue resistance, corrosion resistance, and vibration damping, but it is more expensive and more difficult to manufacture. Carbon fiber can provide extremely low weight and high stiffness, but it is less common due to cost and concerns about brittleness or impact durability in contact-intensive applications [13].

The seating system of a sports wheelchair also differs from that of a traditional wheelchair. Sports wheelchairs commonly use a bucket-style seat that helps limit unwanted lateral body motion and improves containment during aggressive movement [13]. The seat front is often raised relative to the rear, typically by about 5 to 15 degrees, which helps position the athlete securely and can improve balance and stability during turning [13]. Padding is generally kept minimal because excessive cushioning can allow the athlete to shift too much during impacts or rapid maneuvers. High-tension straps and other containment features are often used to help keep the athlete securely positioned in the chair during competitive play [13].

Taken together, the sports wheelchair is a purpose-built athletic platform designed around rapid acceleration, turning, repeated contact, and secure rider positioning. This is important to keep in mind because any added propulsion and control system must preserve as much of that athletic functionality as possible without compromising it.

6.2 Current Electric Wheelchair Control Methods:

Powered wheelchairs are most commonly controlled through proportional joystick systems. One source reports that approximately 80% of powered wheelchairs use joystick-based control [14]. In this type of interface, the user commands speed and direction through joystick displacement, typically using a hand-mounted control located near an armrest. These systems are widely used because they provide intuitive proportional control and can be tuned through parameters such as sensitivity, acceleration ramps, and dead zones [15]. When such parameters are not properly configured, however, the wheelchair may become difficult to control or behave in a way that is less safe for the user [15].

Although joystick control is effective for many users, it depends on sustained upper-body function, fine motor control, and continuous hand use. For many everyday mobility applications this is acceptable, but it is not ideal in situations where the user's hands must remain available for other tasks. In athletic settings, this limitation becomes especially important because the same upper-body motion needed to propel or steer a conventional wheelchair may interfere directly with ball handling, passing, or defensive movement.

Alternative control methods exist for users who cannot rely on standard hand-operated interfaces. Sip-and-puff systems use changes in air pressure through a tube-like interface to generate control commands [16]. These systems can be highly valuable for users with severe limb limitations who retain respiratory control. Other alternatives include head-array systems, which use switches positioned around the headrest, and chin-operated joystick systems, which allow the user to command the chair through head or chin motion rather than hand motion. These approaches expand accessibility to a broader range of users, but they are generally developed for reliable daily mobility rather than high-speed athletic maneuvering.

Different control methods involve tradeoffs among proportionality, precision, responsiveness, fatigue, training burden, and resistance to unintended activation. Joystick systems provide strong proportional control, but they require hand use. Sip-and-puff systems can be reliable and accessible, but they may impose training demands and can cause fatigue under extended or intensive use. Head-array and chin-operated systems can provide effective mobility access, but they depend on stable head motion and may be more sensitive to unintended movements. Many alternative systems also rely more heavily on switch-like inputs than on continuous proportional inputs, which can limit fine control during rapid maneuvering.

Safety is a central requirement across all wheelchair control interfaces. Standards and established design practice emphasize reliability, fault tolerance, prevention of unintended motion, and preservation of controllability under abnormal conditions. Control systems must minimize false activation while still remaining responsive enough for safe maneuvering. Redundant shutdown mechanisms and fail-safe defaults are commonly integrated to reduce user risk in the event of sensor, controller, or power-system faults.

Existing control methods show that wheelchair input design is a question of how motion is commanded, how reliably user intent is interpreted, how much fatigue the interface imposes, and how safely the system behaves when disturbances or faults occur. These tradeoffs are especially important for a lean-controlled sports wheelchair because the intended application demands athletic freedom and safe, intuitive motion control.

6.3 Target User Populations:

Wheelchair basketball classifies athletes according to functional ability, particularly trunk control and movement capability during play. Players are assigned classifications from 1.0 to 4.5 in increments of 0.5, and each team must remain within a total classification-point limit while on the court [17]. In general, lower classifications correspond to less trunk control and less functional reach, while higher classifications correspond to greater trunk mobility and balance during athletic movement [18].

At the lower end of the classification range, players have little to no active trunk control and are more limited in their ability to lean, rotate, or recover balance during play. As classification increases, players generally gain greater ability to lean forward, rotate the torso, and, at the higher classifications, lean laterally with more control [18]. The half-point classifications are used for athletes whose functional ability falls between the major whole-number classes.

This classification framework is important for the present project because a lean-controlled wheelchair depends on the user being able to produce intentional torso motion in a controlled and repeatable way. A user must be capable of leaning enough to generate a meaningful input signal, while also maintaining sufficient balance and posture control for safe athletic operation. For that reason, the concept is most naturally suited to athletes with relatively high trunk function. Based on the functional demands of lean-based control, the most appropriate target population is likely toward the upper end of the wheelchair basketball classification range, approximately classes 3.5 through 4.5. These users are more likely to have the forward, rotational, and lateral trunk motion needed to operate a lean-controlled system effectively.

This does not mean that athletes in lower classifications could never benefit from alternative hands-free control strategies. Rather, it indicates that the specific control concept considered in this project is not equally appropriate for all wheelchair athletes. Users with more limited trunk function would likely require a different control architecture, such as one based on head, chin, or respiratory input, in order to achieve hands-free mobility safely and effectively.

In addition to functional classification, the athletic environment itself affects the target-user discussion. Court sports involve rapid starts, abrupt stops, close-quarters maneuvering, and repeated contact with other athletes and wheelchairs. These conditions place demands not only on the propulsion system, but also on rider stability, seating security, and control robustness. The intended user population must therefore be considered not only in terms of available trunk motion, but also in terms of whether that motion can be used repeatedly and safely under realistic sport conditions.

Taken together, these considerations show that the target user population for a lean-controlled sports wheelchair is narrower than the full population of wheelchair athletes. The concept is best aligned with users who have substantial trunk control and who can intentionally shift posture in multiple directions during dynamic play. Defining this target population clearly helps bound the design problem and supports later decisions related to sensing, control logic, stability, and safety.

7. Problem Analysis

7.1 Forward-Motion Torque Requirements

7.1.1 Analysis Objective

The purpose of the forward motion torque analysis is to estimate the wheel torque required for a lean-controlled sports wheelchair to achieve competitive forward acceleration during court-based athletic use. This subsection develops a simplified physics-based model that relates required wheel torque to the acceleration performance demands of wheelchair sports.

7.1.2 Representative Performance Targets for Competitive Use

To evaluate meaningful torque requirements, a representative range of forward acceleration must first be established. Literature on competitive wheelchair basketball sprint performance reports a 5 m sprint time of approximately 1.87 s for competitive players, corresponding to an average acceleration of about 2.86 m/s^2 [19, 29]. For this analysis, we will use that number as a reasonable representation of competitive-use forward acceleration. Later sections of the report use these results, along with sponsor input and additional design considerations, to define the final acceleration specification.

7.1.3 Modeling Assumptions and Variable Definitions

Several simplifying assumptions are made to bound the problem while staying solution neutral. The wheelchair is assumed to operate on a flat indoor court surface using driven wheels comparable in size to those of the existing sports wheelchair. The analysis further assumes rolling without slip, negligible aerodynamic drag, and a representative total moving mass that includes both the athlete and the wheelchair system.

The forward-motion torque model is developed using the following simplifying assumptions:

- The wheelchair travels on a flat indoor court surface, so grade effects are neglected.
- The driven wheels roll without slipping.
- Aerodynamic drag is negligible relative to the other forces acting on the wheelchair at the speeds considered.
- The driven wheels have a radius comparable to that of the existing sports wheelchair.
- The total moving mass includes the athlete, existing wheelchair, and added propulsion components.
- The propulsion force is shared equally between two driven wheels.
- Rolling resistance is modeled using a constant coefficient of rolling resistance.

Table 2 defines the variables used in the forward-motion analysis.

Table 2: Variables and Descriptions for Acceleration and Torque Calculations

Variable	Description	Units	Initial Value
r	Effective rolling radius of one driven wheel	m	0.3048
v	Forward speed of wheelchair	m/s	—
a	Forward acceleration of wheelchair	m/s ²	—
ω	Angular speed of driven wheel	rad/s	—
α	Angular acceleration of driven wheel	rad/s ²	—
m	Total moving mass of wheelchair and rider	kg	105
I_w	Moment of inertia of one driven wheel about its axle	Kg m ²	.169
I_o	Additional inertia reflected to the wheel axle from drivetrain components	Kg m ²	—
$I_{eq,w}$	Equivalent rotational inertia at one wheel,	Kg m ²	—
N_d	Number of driven wheels	—	2
F_t	Traction force at the ground from one driven wheel	N	—
F_{rr}	Rolling resistance force	N	—
F_{grade}	Component of gravitational force along direction of travel	N	0
F_{drag}	Aerodynamic drag force	N	≈ 0
C_{rr}	Coefficient of rolling resistance [20]	—	.002
θ	Ground slope angle	deg	0
g	Gravitational acceleration	m/s ²	9.81
τ_w	Torque applied at one wheel axle	N m	—

For this analysis, the wheel radius is taken from the existing sports wheelchair, which uses 24 in diameter wheels, corresponding to a radius of 0.3048 m. The representative total moving mass is taken as 105 kg, based on the athlete at 150 lb, the existing wheelchair at 31.3 lb, and an estimated added propulsion and

mounting system mass. The coefficient of rolling resistance is assumed to be 0.002, consistent with a bicycle-like tire on concrete. All calculations are performed in SI units.

7.1.4 Forward-Motion Torque Model

To relate required wheel torque to forward acceleration, the wheelchair is modeled as a rigid body translating on a flat surface while the driven wheels rotate without slippage. The propulsion force is generated at the tire-ground interface by the two driven wheels. The following free body diagram is used to assist in the analysis.

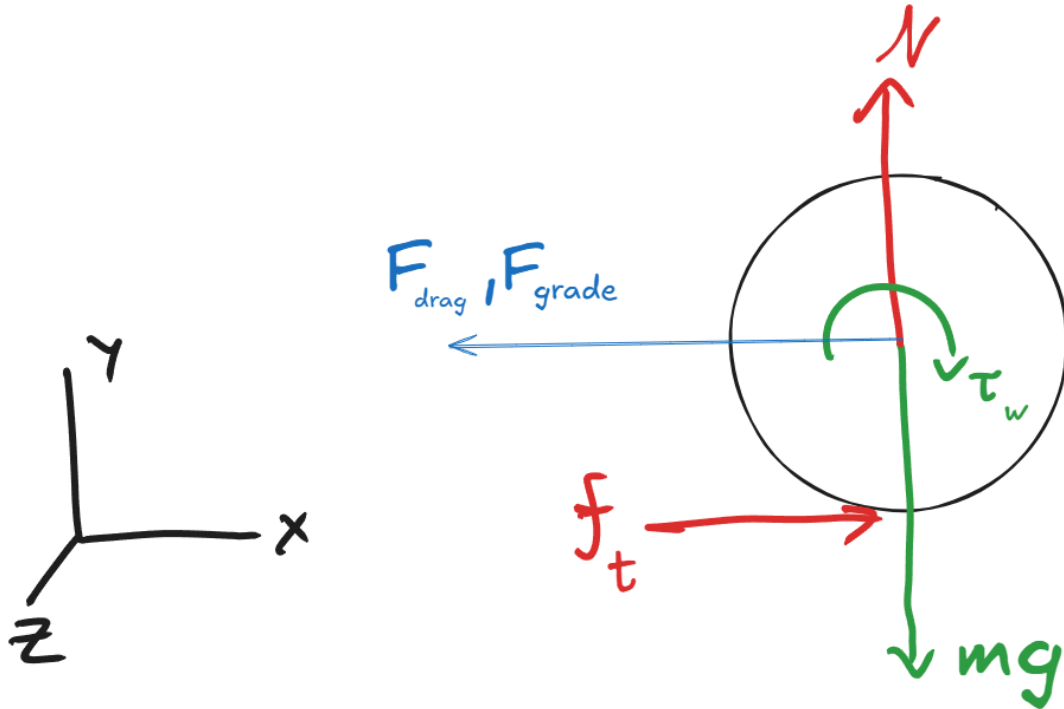


Figure 1: Free body diagram of wheelchair rolling. N is a variable representing the normal reaction force.

Applying Newton's second law in the forward direction gives

$$\Sigma F_x = ma$$

$$N_d f_t - F_{rr} - F_{grade} - F_{drag} = ma$$

For the flat-court case used in this analysis,

$$F_{grade} = mgsin(\theta) = 0$$

and the rolling resistance force is modeled as

$$F_{rr} = C_{rr} mg \cos(\theta) \approx C_{rr} mg$$

Aerodynamic drag is neglected, so

$$F_{drag} \approx 0$$

Substituting these assumptions into the force balance gives

$$N_d F_t - C_{rr} mg = ma$$

Solving for acceleration,

$$\frac{N_d F_t}{m} - C_{rr} g = a \quad (1)$$

Next, summing moments about the axle of one driven wheel gives

$$\Sigma M_{axle} = I \alpha$$

$$\tau_w - F_t r = I_{wheel,eq} \alpha$$

Where

$$I_{wheel,eq} = I_w + I_o$$

Because rolling without slip is assumed,

$$v = r \omega$$

and therefore

$$a = r \alpha \Rightarrow \alpha = \frac{a}{r}$$

Substituting this into the wheel moment equation gives

$$\tau_w - F_t r = I_{w,eq} \left(\frac{a}{r} \right)$$

Solving for traction force,

$$F_t = \frac{\tau_w}{r} - \frac{I_{w,eq}}{r^2} a \quad (2)$$

Substituting Equation (2) into Equation (1),

$$\frac{N_d}{m} \left(\frac{\tau_w}{r} - \frac{I_{w,eq}}{r^2} a \right) - C_{rr} g = a$$

Rearranging to solve for required wheel torque,

$$\tau_w = \frac{mr}{N_d} (a + C_{rr} g) + \frac{I_{eq,w} a}{r} \quad (3)$$

Equation (3) gives the torque that must be supplied at each driven wheel axle to achieve a desired forward acceleration. For the flat-court, two-driven-wheel case considered here, the required torque is governed mainly by total moving mass, wheel radius, and target acceleration. Rolling resistance and equivalent wheel-side rotational inertia are included in the model, but they are expected to have a comparatively smaller effect for the operating conditions considered.

7.1.5 Experimental Determination of Wheel Moment of Inertia

Equation (3) requires the rotational inertia seen at the driven wheel. While the wheel moment of inertia could be estimated from CAD, a simple experiment was performed to obtain a more realistic value for the existing sports wheelchair wheel. A SolidWorks estimate gave a wheel moment of inertia of 0.137 kg m^2 but an experimental measurement was preferred because it better reflects the physical wheel being used in the analysis.

To estimate the inertia, a known mass was allowed to fall while attached to a rope wrapped around the wheelchair wheel. As the mass fell, it applied a torque to the wheel and caused it to rotate. By measuring the falling mass, drop height, and elapsed time, the wheel moment of inertia could be calculated from the equations of motion.

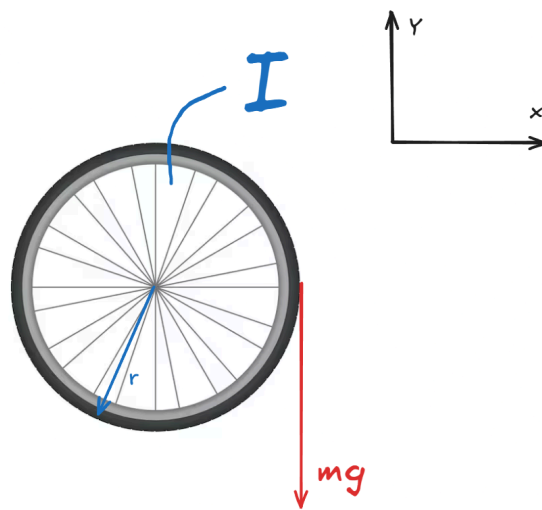


Figure 2: Inertia experiment FBD

For the falling mass, the displacement relation is

$$y = \frac{1}{2}at^2$$

so the linear acceleration is

$$a = \frac{2y}{t^2}$$

Since the measured vertical drop is h ,

$$a = \frac{2h}{t^2}$$

Applying Newton's second law to the falling mass gives

$$mg - T = ma$$

so the rope tension is

$$T = mg - ma$$

The torque applied to the wheel by the rope is then

$$Tr = I\alpha$$

Assuming the rope does not slip on the wheel,

$$\alpha = \frac{a}{r}$$

Substituting this into the torque equation gives

$$Tr = I\left(\frac{a}{r}\right)$$

and substituting $T = m(g - a)$ gives

$$m(g - a)r = I\frac{a}{r}$$

Solving for the wheel moment of inertia,

$$I = \frac{mr^2}{a}(g - a)$$

Finally, substituting $a = \frac{2h}{t^2}$ yields

$$I = \frac{mr^2 t^2}{2\Delta h} \left(g - \frac{2\Delta h}{t^2} \right)$$

which simplifies to

$$I = \frac{mr^2 t^2 g}{2h} - mr^2 \quad (4)$$

Equation (4) gives the moment of inertia of the wheel in terms of the measured falling mass, drop height, and fall time.

The experimental setup is shown in Figure 3. A rope was wrapped around the wheelchair wheel and attached to a known hanging mass. The wheel was then held stationary and released, allowing the mass to fall and accelerate the wheel. A Windex bottle filled with varying amounts of water was used as the hanging mass because it allowed fine adjustment of the applied load. Slow-motion video was used to determine the time required for the mass to fall through a known vertical distance. Multiple trials were performed using two different masses, and the resulting inertia values were averaged.

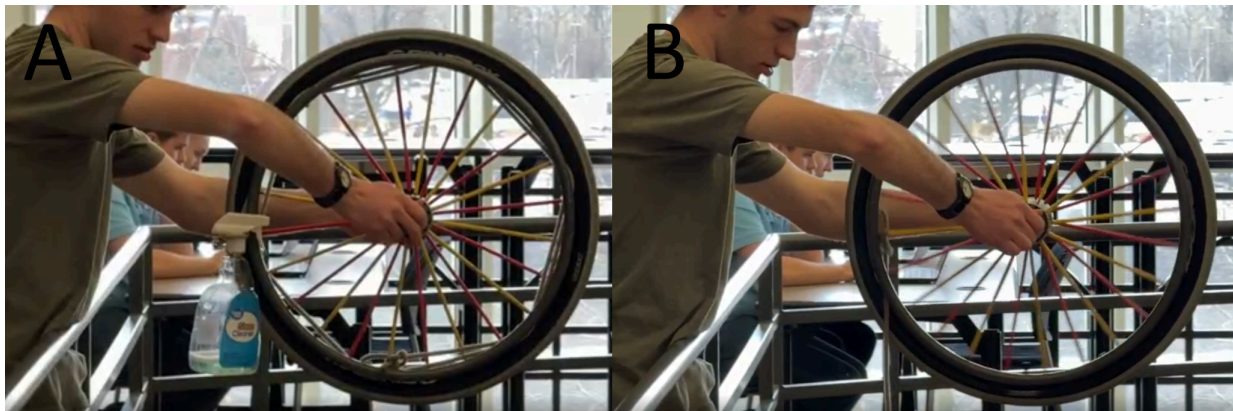


Figure 3: Experimental setup for moment of inertia determination. A depicts the initial state, B depicts the final state.

Trial	Mass (kg)	Time (s)	Moment of Inertia
1	0.1975	2.63	0.204
2	0.1975	2.28	0.149
3	0.3023	1.97	0.163
4	0.3023	1.96	0.161
		Average:	0.170

Using Equation (4), the average experimentally determined moment of inertia of one wheel was found to be $I_w = 0.170 \text{ kg m}^2$. This value differs from the SolidWorks estimate by about 19%. The experiment does include some sources of uncertainty, including neglecting the mass of the rope and omitting the wheel's plastic covers during testing. However, as shown in the following subsection, the required torque is far more sensitive to target acceleration than to moderate

uncertainty in wheel inertia, so this measured value is sufficiently accurate for the problem analysis.

7.1.6 Sensitivity of Torque Requirement to Key Parameters

Equation (3) shows that the required wheel torque depends on several parameters, including the target forward acceleration, rolling resistance, wheel radius, total moving mass, number of driven wheels, and equivalent rotational inertia at the wheel. At this stage of the problem analysis, the parameters with the greatest uncertainty were the equivalent wheel-side rotational inertia and the target competitive-use acceleration. A sensitivity analysis was therefore performed to determine which of these quantities most strongly influences the required wheel torque.

The first parameter considered was the equivalent rotational inertia at the wheel, $I_{eq,w}$. This quantity includes both the experimentally determined wheel inertia and any additional inertia reflected to the wheel from drivetrain components such as the gearbox, motor, or other rotating elements. Because the final drivetrain configuration had not yet been selected, this value was uncertain. To evaluate its influence, Equation (3) was used to calculate required wheel torque over a range of plausible inertia values while holding forward acceleration constant at the representative competitive acceleration of 2.86 m/s^2 and using $C_{rr} = 0.002$. The resulting trend is shown in Figure 4.

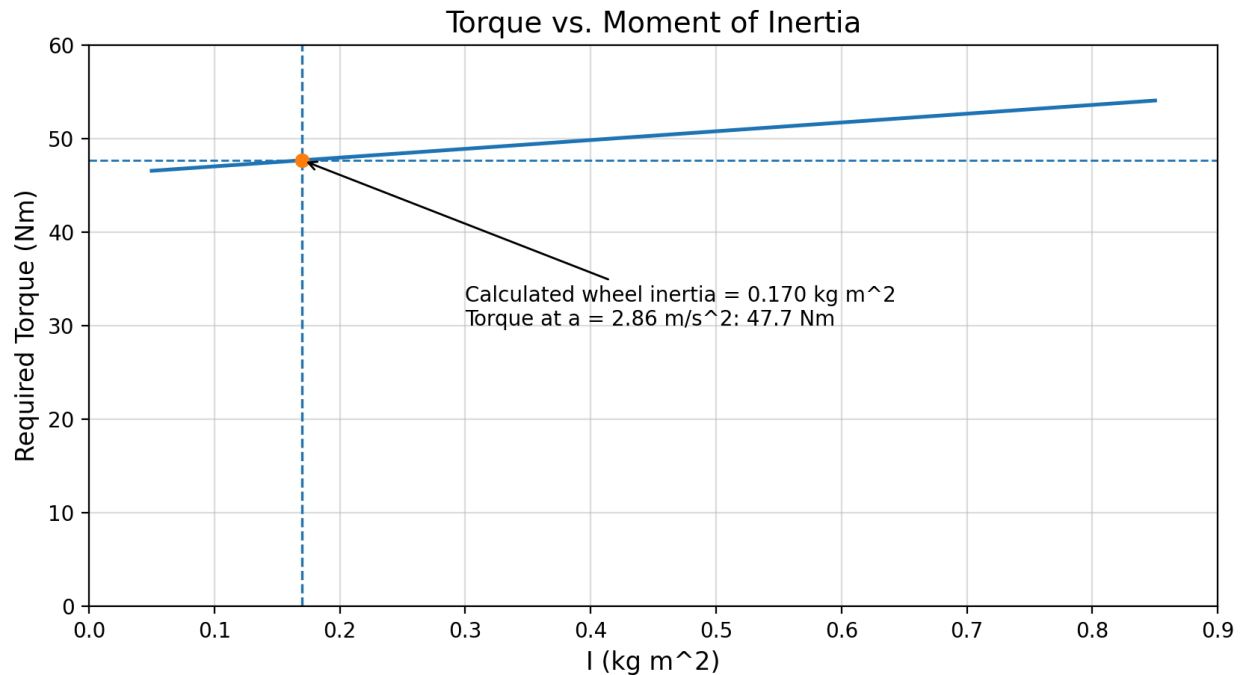


Figure 4: Required torque vs. equivalent wheel-side moment of inertia. Acceleration is held constant at 2.86 m/s^2 and $C_{rr} = 0.002$

As shown in Figure 4, varying the equivalent rotational inertia has only a modest effect on the required wheel torque. This indicates that uncertainty in drivetrain inertia does not strongly influence early torque estimates for the wheelchair. Although the wheel inertia was experimentally measured in the previous subsection, the sensitivity analysis shows that moderate error in that value would have only a small effect on the final torque

requirement. This is useful because it means that the forward-motion torque analysis is not highly dependent on detailed knowledge of the final drivetrain internals.

The second parameter considered was forward acceleration. Using Equation (3), the required wheel torque was evaluated over a range of accelerations while holding the equivalent rotational inertia at 0.20 kg m^2 , which is slightly greater than the experimentally measured wheel inertia and was selected as a conservative estimate to account for additional reflected drivetrain inertia. The resulting trend is shown in Figure 5.

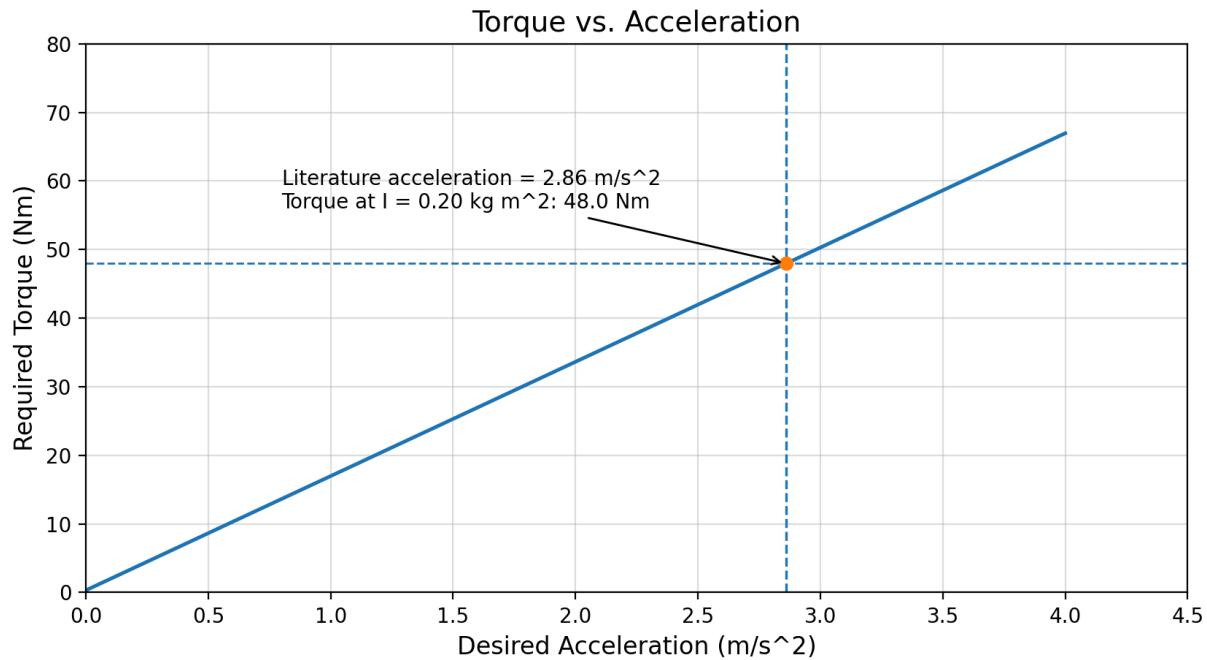


Figure 5: Predicted torque required vs. desired acceleration. Moment of inertia assumed to be 0.2 kg m^2 and C_{rr} is set to 0.002

Figure 5 shows that the required wheel torque changes strongly with target acceleration. This indicates that acceleration is the dominant driver of torque demand in the simplified forward-motion model. In practical terms, this means that early drivetrain sizing is governed much more by the competitive performance target than by uncertainty in wheel inertia. Therefore, a reasonable estimate of representative athletic acceleration is more important to the analysis than a highly precise estimate of equivalent rotational inertia.

Taken together, the sensitivity analysis shows that the required torque is primarily controlled by the desired forward acceleration and the total moving mass, while equivalent wheel-side rotational inertia plays a comparatively minor role. This finding simplifies the remaining problem analysis and supports using Equation (3) to estimate a representative torque range for competitive operation in the following subsection.

7.1.7 Estimated Torque Range for Competitive Use

With the forward-motion model established, Equation (3) can now be used to estimate the wheel torque required for competitive wheelchair sports use. As discussed in Section 7.1.2, competitive wheelchair

basketball sprint performance suggests a representative forward acceleration of approximately 2.86 m/s^2 over a 5 m sprint [29]. To express the result as a practical range rather than a single exact value, it is also useful to evaluate nearby accelerations representative of aggressive court movement.

Using Equation (3),

$$\tau_w = \frac{mr}{N_d} (a + C_{rr}g) + \frac{I_{eq,w}a}{r}$$

and taking

$$m = 105 \text{ kg}, r = 0.3048 \text{ m}, N_d = 2, C_{rr} = 0.002$$

With

$$I_{eq,w} = 0.20 \text{ kg m}^2$$

as a conservative estimate of total wheel-side rotational inertia, the required wheel torque can be evaluated for representative competitive-use accelerations.

For a lower-end aggressive acceleration of

$$a = 2.5 \text{ m/s}^2$$

the required torque at each driven wheel is

$$\tau_w \approx 42.0 \text{ N m}$$

For the representative competitive acceleration taken from the wheelchair basketball sprint literature,

$$a = 2.86 \text{ m/s}^2$$

the required torque at each driven wheel is

$$\tau_w \approx 48.0 \text{ N m}$$

For a slightly more aggressive acceleration of

$$a = 3.00 \text{ m/s}^2$$

the required torque at each driven wheel is

$$\tau_w \approx 50.3 \text{ N m}$$

These results indicate that a viable lean-controlled sports wheelchair intended for competitive court use will likely require approximately **42 to 50 N·m of torque at each driven wheel** in order to achieve strong forward acceleration.

This result is useful because it establishes a solution-neutral performance target for the drivetrain. Before specific motors, gearboxes, or controllers are selected, the analysis shows that the wheelchair must be capable of supplying on the order of tens of newton-meters of torque per wheel, rather than only a few newton-meters. This estimate then serves as a baseline for later design decisions and specification development.

7.2 Power Supply:

7.2.1 Analysis Objective

The purpose of the power supply analysis is to evaluate the performance requirements needed for the system to provide a stable and reliable power source during operation. This section develops a simplified model that relates the voltage, current, and power demands for the system from the torque needed to meet acceleration speculations demanded of wheelchair sports.

7.2.2 Representative Performance Targets for Competitive Use

To evaluate a meaningful power supply requirement, a range of electrical load demand must be established. The system is expected to support continuous and transient operating conditions while maintaining stable voltage regulation and current delivery to all connected components. For this analysis, representative operating power levels and current demands are used to model realistic system usage conditions. These provide a practical basis for estimating the required power needed, voltage stability, and safety margins for the power supply. Later sections of the report use these methods and additional constraints to define the final power supply specifications and component selection criteria.

7.2.3 Modeling Assumptions and Variable Definitions

Several simplifying assumptions are made to bound the problem while staying solution neutral. The power supply is assumed to operate under typical indoor conditions while supplying power to electrical components representative of the final design configuration. The analysis further assumes steady operating voltages, predictable load behavior, and negligible transmission losses over short wiring distances.

The power supply model is developed using the following simplifying assumptions:

- The system operates under standard indoor environmental conditions, so temperature related performance effects are neglected.
- The supplied voltage remains constant within the normal operating range of the connected components.

- Electrical loads are modeled using representative average and peak current demands.
- Power losses in short wiring connections and connectors are negligible to the total system power consumption.
- The total load induces all control electronics, motors, sensors, and auxiliary system components.
- Peak load conditions and average load conditions are considered when estimating maximum current requirements.
- The power supply is assumed to distribute power efficiently to all connected loads without significant imbalance.
- Safety and design margins are incorporated to account for unexpected load variations.

Table 4: Variables and Descriptions for Acceleration and Torque Calculations

Variable	Description	Units	Initial Value
V	Nominal system voltage	V	48
I_{avg}	Average system current draw	A	–
I_{peak}	Peak system current draw	A	–
P_{avg}	Average electrical power demand	W	–
P_{peak}	Peak electrical power demand	W	–
P_{mech}	Total mechanical power output	W	–
P_{elec}	Electrical power accounting for efficiency losses	W	–
η	Drivetrain efficiency	–	0.85
g	Total system gear ratio	–	16:1
t_{op}	Amount of time to operate chair	h	2
Ah	Battery capacity requirement	Ah	–
ω	Average wheel angular velocity	rad/s	260
T_{wheelp}	Peak torque on wheel	Nm	42/48/53
T_{motorp}	Peak torque on motor	Nm	–
T_{motora}	Average torque on motor	Nm	1.43

For this analysis, the wheel radius is taken from the existing sports wheelchair, which uses 24 in diameter wheels, corresponding to a radius of 0.3048 m. The gear ratio is taken as the expected values needed. The total run time is assumed to be 2 hours from the length of an average game, and T_{motora} is an assumed average torque. All calculations are performed in SI units.

7.2.4 Power assumption calculations

The mechanical power for both wheels is given by:

$$P_{mech} = 2\tau\omega$$

where the torque is for the motor, to take the torque of the wheel and convert it to the motor it is:

$$\tau_{motorp} = \frac{\tau}{g}$$

This gives us 3 peak mechanical torques at the motor of:

$$\tau_{motorp} = \frac{42}{16} = 2.625 Nm$$

$$\tau_{motorp} = 48/16 = 3 Nm$$

$$\tau_{motorp} = 53/16 = 3.313 Nm$$

Using these values the peak P_{mech} can be found to be:

$$P_{mech} = 2 * 2.625 * 260 = 1365 W$$

$$P_{mech} = 2 * 3 * 260 = 1560 W$$

$$P_{mech} = 2 * 3.313 * 260 = 1722 W$$

While the average P_{mech} can be found to be:

$$P_{mech} = 2 * 1.43 * 260 = 743 W$$

To convert from P_{mech} to P_{elec} :

$$P_{elec} = \frac{P_{mech}}{\eta}$$

This gives us the following P_{peak} and P_{avg} :

$$P_{peak} = \frac{1365}{0.85} = 1605.88 W$$

$$P_{peak} = \frac{1560}{0.85} = 1835.29 W$$

$$P_{peak} = \frac{1722}{0.85} = 2025.88 W$$

$$P_{avg} = \frac{743}{0.85} = 874.12 \text{ W}$$

From these P values, the assumed current needed will be found by:

$$I = \frac{P}{V}$$

This gives us:

$$I_{peak} = \frac{1605.88}{48} = 33.46 \text{ A}$$

$$I_{peak} = \frac{1835.29}{48} = 38.24 \text{ A}$$

$$I_{peak} = \frac{2025.88}{48} = 42.21 \text{ A}$$

$$I_{avg} = \frac{874.12}{48} = 18.21 \text{ A}$$

To find the assumed capacity size the equation is:

$$Ah = t_{op} * \frac{I_{avg}}{0.8}$$

This will give us a capacity size of:

$$Ah = \frac{2 * 18.21}{0.8} = 45.525 \text{ Ah}$$

7.2.5 Conclusion and Summary:

This power-supply analysis establishes a conservative electrical sizing target for the lean-controlled wheelchair. Based on the assumed 48 V system voltage, 85% drivetrain efficiency, and representative motor torque demands, the system is expected to require approximately 875 W during sustained high-load operation and up to approximately 2.0 kW during peak acceleration events. These values correspond to an estimated average current draw of approximately 18.2 A and a peak current draw of approximately 42.2 A at 48 V.

For a 2.0 hour operating duration with an 80% usable depth of discharge, this conservative estimate gives a required battery capacity of approximately 45.5 Ah at 48 V. This result should be treated as an upper-bound sizing estimate because it assumes a relatively high average motor torque. Later design analysis refines this estimate using representative gameplay acceleration data and compares the resulting current and capacity requirements to the selected battery.

Overall, the power supply must be capable of supporting a 48 V architecture, peak electrical loads on the order of 2 kW, and sustained current delivery high enough to support a full athletic session without excessive voltage sag, overheating, or premature shutdown.

7.3 Control Requirements for Competitive Use

7.3.1 Analysis Objective

The purpose of this subsection is to identify the control-performance demands imposed by competitive court sports on a lean-controlled wheelchair. Unlike a normal powered wheelchair, the proposed system must translate torso motion into propulsion and steering commands while the user is engaged in athletic tasks such as passing, receiving, reaching, and avoiding contact at the same time. The control problem involves generating motion from body lean in a way that remains intuitive, responsive, stable, and resistant to unintended activation during competitive play.

7.3.2 Modeling Assumptions and Context

Several assumptions are made in order to analyze the control problem at a solution-neutral level. The intended operating environment is a flat indoor court surface used for sports such as wheelchair basketball. The intended user population has sufficient trunk control to produce meaningful forward and lateral torso lean. During normal athletic use, the wheelchair will experience both deliberate user-generated posture shifts and unplanned disturbances such as collisions, abrupt stops, and rapid balance corrections. It is also assumed that upper-body motion during sport is not always intended as a command to the wheelchair, since the athlete may lean significantly while catching, passing, contesting the ball, or recovering posture.

These assumptions create a more demanding control problem than that of a normal daily-use powered wheelchair. A standard control system only needs to respond accurately to explicit input commands. In this case, however, the control system must infer driving intent from body motion that may serve both athletic and navigational purposes. As a result, the system must balance responsiveness with selectivity so that the chair remains intuitive to operate without becoming overly sensitive to normal sport-related movement.

7.3.3 Representative Competitive Control Scenarios

Competitive wheelchair sport suggests at least two primary control scenarios and one important non-command scenario.

The first scenario is close-quarters maneuvering. This occurs in and around active play, where the athlete must reposition rapidly for passes, defend nearby opponents, or create small openings for movement. The wheelchair must support fine directional control, rapid turning response, and predictable behavior at low to moderate speeds. Small changes in user posture should produce smooth and controllable changes in motion rather than abrupt or exaggerated commands.

The second scenario is court transition. This occurs when possession changes or when the athlete must move quickly across the court. Strong forward acceleration and stable high-speed control are more important than tight maneuverability. The control system must allow the athlete to command aggressive forward motion without introducing instability, oscillation, or excessive steering sensitivity.

The third scenario is non-command athletic motion. During gameplay, the athlete may lean forward to receive a pass, laterally to intercept the ball, or abruptly in response to player contact. These motions may resemble intentional drive inputs even when no wheelchair motion is desired. This creates a fundamental design choice

in the control philosophy. One approach is to make the wheelchair adapt to the athlete's existing playstyle by attempting to distinguish athletic posture changes from true drive commands. The other approach is to allow these lean motions to produce wheelchair response and instead expect the athlete to adapt to the new movement possibilities and constraints introduced by the chair. Either we design the chair to adapt to the player, or the player must adapt to the chair. In practice, the final design may fall somewhere between these two philosophies, but the control system must be developed with this tradeoff in mind because it strongly affects responsiveness, intuitiveness, and the overall style of play the wheelchair will support.

Taken together, these scenarios show that the control requirements of the wheelchair are not defined by responsiveness alone. The system must also preserve controllability across a wide range of operating conditions and minimize false activation during normal athletic movement.

7.3.4 Control Implications for a Viable Solution

The control problem defined above leads to several important solution-neutral conclusions. First, the wheelchair must provide both fine low-speed maneuverability and stable high-speed response. The control behavior required for close-quarters play is not necessarily the same as that required for full-court transitions. A viable solution will therefore need a control strategy that remains usable across these different operating regimes.

Second, the controller must address the relationship between athletic body motion and drive intent. Because torso motion during sport does not uniquely indicate a desire for wheelchair propulsion, the final design must adopt a deliberate control philosophy. One approach is to reduce unintended activation by distinguishing athletic posture changes from true drive commands through thresholds, deadbands, filtering, or command shaping. Another approach is to allow a stronger coupling between body lean and wheelchair response, accepting that some athletic motions will produce movement and that the user will adapt to the new playstyle this creates. A viable solution may lie between these extremes, but the tradeoff must be considered explicitly because it strongly influences the design approach.

Third, the control system must remain stable and predictable when disturbances occur. Competitive wheelchair sport includes contact, rapid posture correction, and abrupt directional changes. A viable control system must therefore tolerate transient disturbances without producing unsafe or confusing motion. This requirement is especially important because the user's body serves both as the control input source and as part of the physical system being disturbed.

Finally, the control system must preserve a safe fallback operating condition. Because the wheelchair is intended for competitive use, loss of control authority or unintended motion presents a significant safety concern. Any viable design must therefore provide a means of quickly reducing or removing commanded motion if the sensed input becomes unreliable or if continued powered operation becomes undesirable.

Overall, this analysis shows that the control problem for a lean-controlled sports wheelchair is fundamentally different from that of a typical powered wheelchair. The controller must do more than respond quickly to body motion. It must balance responsiveness, maneuverability, high-speed stability, disturbance tolerance, and resistance to false activation. These conclusions establish the problem-level control demands that later design choices must satisfy.

8. Standards & Regulations:

This section identifies the external standards, test methods, and sport-specific rules most relevant to the design of the lean-controlled sports wheelchair. Because the project combines a powered mobility system with a competitive athletic wheelchair, the design is constrained by both powered-wheelchair safety standards and wheelchair-basketball equipment rules. The standards reviewed in this section were selected because they most directly affect stability, braking, energy use, battery safety, and on-court geometry. These standards and rules, serve as important references for design decisions, specification development, and later validation of the system.

8.1 Determination of static stability

ISO 7176-1:2014 [21], WHEELCHAIRS — PART 1: DETERMINATION OF STATIC STABILITY.

Description:

This standard specifies the requirements for static stability under several conditions such as a flat hard platform with an adjustable slope that is large enough to accommodate the wheelchair to be tested. Along with this it has requirements on the need for slide restraints to prevent an unlocked wheel or anti-tip device from rolling that still prevents a locked wheel from moving or sliding.

Impacts:

This standard establishes the baseline for testing the safety of the wheelchair in static positions and ensures that the chair remains reliable and safe while in a static state.

1. Adjustable platform:

The platform must be adjustable and lie between two imaginary parallel planes 5mm apart with no more than a 0.5 degree angle of variation in slope of or cross slope throughout the test. Must also have a roll restraint to prevent an unlocked wheel from rolling.

2. Slide Restraint:

The slide restraint must prevent a locked wheel or anti tip device from sliding that does not affect the wheelchair's freedom to tip about the contact point of the restrained wheel or post.

3. Tipping Limiter:

The tipping limiter to limit the extent of tipping of the wheelchair relative to the test platform that does not affect the stability of the wheelchair, restrict the wheelchair's freedom to deform, or restrict the wheelchair's freedom to tip sufficiently to detect a zero force under the uphill wheels of the wheelchair.

4. Angular Measurement:

The angular measurement device to measure the angle of the slope of the test platform along with a test dummy that must be secured by the torso, thigh, and lower leg portions in the wheelchair.

8.2 Determination of dynamic stability of electrically powered wheelchairs

ISO 7176-2:2017 [22], WHEELCHAIRS — PART 2: DETERMINATION OF DYNAMIC STABILITY OF ELECTRICALLY POWERED WHEELCHAIRS.

Description:

This standard specifies testing methods and performance requirements related to the dynamic stability of electrically powered wheelchairs. It includes a range of defined procedures that evaluate whether a wheelchair can remain upright while in motion, accounting for factors such as acceleration, turning, momentum, and operator input.

Impacts:

This standard establishes the baseline expectations for the dynamic stability of the wheelchair design. Compliance ensures that the wheelchair will remain stable during normal operation, even when subjected to changes in speed, direction, and operator movement.

1. Dynamic Stability Testing:

The standard defines a variety of stability tests intended to simulate real-world operating conditions. These tests evaluate whether the wheelchair maintains balance and avoids tipping while accelerating, decelerating, or maneuvering. Meeting these requirements ensures that the wheelchair will stay upright during typical usage scenarios.

2. Surface Inclination Considerations:

Some procedures within the standard involve testing the wheelchair on inclined surfaces to evaluate stability on slopes. Since the intended application for this wheelchair is indoor basketball on a flat gymnasium floor, the inclined-surface portions of the testing procedures may be considered less critical for the final design validation.

3. Control System Stability Constraints:

Because the wheelchair will be lean-controlled, the stability requirements in this standard provide important constraints for the control system design. The system must ensure that operator leaning inputs do not cause unsafe tipping conditions and that the chair remains within stable operating limits regardless of how the user shifts their weight.

4. Design Safety Implications:

Adhering to this standard ensures that the final wheelchair design maintains safe operating characteristics during dynamic movement. These requirements guide the mechanical design, center-of-mass placement, and control system behavior to ensure the wheelchair remains stable and safe for athletic use.

8.3 Determination of effectiveness of brakes

ISO 7176-3:2012 [23], WHEELCHAIRS — PART 3: DETERMINATION OF EFFECTIVENESS OF BRAKES.

Description:

This standard specifies testing procedures and performance requirements for the braking capabilities of wheelchairs. It defines methods for evaluating both static and dynamic braking performance to ensure that the wheelchair can remain stationary when required and safely stop while in motion.

Impacts:

This standard establishes the baseline safety expectations for the braking system of the wheelchair. Compliance ensures that the chair can reliably stop and maintain control during operation, which is critical for safe use during athletic activities.

1. Static Braking Performance:

One of the primary tests defined in the standard evaluates the effectiveness of the brakes when the wheelchair is stationary. This test measures the maximum slope on which the wheelchair can remain at rest without sliding when the brakes are engaged. While this test verifies the holding capability of the braking system, it is less critical for this design because the wheelchair is intended for use on a flat indoor gym floor.

2. Dynamic Braking Performance:

The second major test evaluates braking performance while the wheelchair is moving at its maximum velocity. This procedure measures the distance required for the wheelchair to come to a complete stop once the brakes are applied. Meeting this requirement ensures that the operator can maintain control of the wheelchair during high-speed movement and sudden stopping scenarios.

3. Brake Actuation Force Requirements:

The standard includes specifications for the force required to engage the braking system, typically presented in tabulated form. These values provide guidance for designing the brake mechanism so that it can be reliably activated while still remaining accessible and practical for the operator.

4. Design Safety Implications:

Adhering to this standard ensures that the wheelchair braking system provides consistent stopping performance and prevents uncontrolled movement. The requirements guide the selection and design of the braking components to maintain safe control of the wheelchair at all times during operation.

8.4 Energy consumption of electric wheelchairs and scooters for determination of theoretical distance range

ISO 7176-4:2008 [24], WHEELCHAIRS — PART 4: ENERGY CONSUMPTION OF ELECTRIC WHEELCHAIRS AND SCOOTERS FOR DETERMINATION OF THEORETICAL DISTANCE RANGE.

Description:

This standard defines testing procedures used to evaluate the energy consumption and maneuverability performance of electrically powered wheelchairs during operation. The tests measure how efficiently a wheelchair uses energy during continuous driving as well as while performing maneuvering tasks representative of real-world use.

Impacts:

This standard provides a framework for assessing how efficiently the wheelchair converts stored electrical energy into motion. Compliance helps ensure that the wheelchair can operate for an adequate duration while maintaining the level of maneuverability required for athletic environments such as wheelchair basketball.

1. Continuous Driving Energy Test:

One of the primary tests measures the energy consumption of the wheelchair during continuous motion. In this procedure, the operator drives the wheelchair in a circular path for a specified period, and the energy consumed during this activity is recorded. The results provide a metric for evaluating the overall energy efficiency of the propulsion system during sustained movement.

2. Maneuvering Energy Test:

A second test evaluates energy consumption during maneuvering tasks. In this procedure, a course is established using cones, and the wheelchair must navigate between them. The spacing between the cones is used in the calculation of performance metrics. Wheelchairs that are capable of tighter turning and more precise maneuvering will perform better in this test, as they can complete the course more efficiently.

3. Design Performance Considerations:

Because the wheelchair is intended for basketball gameplay, the ability to move quickly across the court while maneuvering around other players is essential. These tests provide useful performance metrics for evaluating how well the design supports rapid directional changes and efficient motion.

4. Design Efficiency Implications:

Adhering to this standard helps ensure that the wheelchair design balances energy efficiency and maneuverability. The results guide decisions related to motor selection, drivetrain configuration, and control system design, reinforcing that efficient operation involves more than simply traveling long distances in a straight line.

8.5 Lead-acid batteries and chargers for powered wheelchairs

ISO 7176-25:2022 [25], WHEELCHAIRS — PART 25: LEAD-ACID BATTERIES AND CHARGERS FOR POWERED WHEELCHAIRS.

Description:

This standard specifies safety and performance requirements for lead-acid batteries and their associated charging systems (rated up to 250 V AC input and 36 V nominal output) designed specifically for electrically powered wheelchairs and scooters. It ensures that the battery set and charger are compatible, reliable, and safe during normal use as well as foreseeable misuse.

Impacts:

Regardless of the final battery solution used, this standard establishes the baseline safety and performance criteria for any lead-acid power solution being considered for the final solution of the wheelchair.

1. Risk Assessment:

The design process must include a formal risk analysis in accordance with ISO 14971. This analysis must account for storage and operating temperature, as well as any foreseeable misuse.

2. Environmental Ingress Requirements:

Any charging solution integrated into the design must meet specific Ingress Protection (IP) ratings. For indoor basketball use, a minimum of IPX1 is required, but if the chair is used in non-dry environments, IPX4 protection must be ensured.

3. Mechanical Fatigue Constraints:

Any on-board charging or battery housing components must be designed to survive the multi-drum and kerb-drop fatigue tests defined in ISO 7176-8. This is a significant constraint for a basketball wheelchair, as the mounting system must prevent connections from working loose during frequent on-court collisions.

4. Interface and Connector Selection:

If a lead-acid system is utilized, the charger must conform to IEC 60335-2-29 and utilize standardized connectors, such as the XLR type defined in IEC 61076-2-103. The design must also include reverse polarity protection to prevent electrical hazards during user-led maintenance.

5. Charger Compatibility and Intelligence:

The standard requires that the charging system be capable of identifying a faulty battery set and preventing unsafe charging. Our future control system will need to support these performance-related safety indicators to inform the athlete of the system status.

8.6 Lithium-ion battery systems and chargers for powered wheelchairs

ISO 7176-31:2023 [26], WHEELCHAIRS — PART 31: LITHIUM-ION BATTERY SYSTEMS AND CHARGERS FOR POWERED WHEELCHAIRS.

Description:

This international standard establishes specific safety requirements and test methods for lithium-ion batteries, battery management systems, and charging systems used in electrically powered wheelchairs. It addresses unique risks associated with Li-ion chemistry, such as thermal runaway and fire propagation. It standardizes how these systems should perform during both stationary charging and active wheelchair operation (including regenerative charging during deceleration).

Impacts :

Because the wheelchair basketball design is motorized and uses a lean-control system, compliance with ISO 7176-31:2023 is critical for both user safety and system reliability under high-performance conditions, particularly if a lithium-ion battery were used in the final solution.

1. Safety and Fire mitigation:

The standard requires that a thermal runaway event within a single cell must not result in a fire of the entire battery system or propagate outside the housing. For a contact-heavy sport like wheelchair basketball, the battery enclosure must be designed to withstand impacts (as per cross-referenced ISO 7176-8) while maintaining this fire containment.

2. Battery Management System (BMS):

The design must include a BMS that maintains safety by preventing damage and managing cell imbalances. The BMS must communicate with the wheelchair controller to report cell voltages, temperatures, and state of charge.

3. Connector Standardization:

If the design includes an external charger, the standard specifies requirements for charger connectors, such as the XLR type. This ensures communication between the charger and control system and prevents electrical hazards during the charging cycle.

8.7 NWBA Official Rules

WHEELCHAIRS — PART XX: NWBA OFFICIAL RULES (RULE 1, SECTION 2: PLAYER EQUIPMENT) [27]

Description:

This Document establishes the legal specifications for wheelchairs used in competitive wheelchair basketball. It ensures a level playing field by regulating wheelchair dimensions and safety features, such as seat height, wheel size, and bumper placement, to prevent injury or damage to the court.

Impacts:

The NWBA standards could act as the primary geometric constraints for our lean-controlled wheelchair. Following these rules is essential for the chair to be as close to “game-legal” as possible and safe for interacting with other players in basketball.

1. Chassis and bumper Design:

The requirement that the front horizontal bar be no more than 11cm off the floor dictates the mounting height of our front end assembly. This will ensure that during a collision, the wheelchair will not “ride up” over other players' wheels/bumper and cause problems.

2. Drive System and wheel selection:

The restriction on the rear wheel diameter (max 69cm including tires) directly impacts our motor torque calculations. We must select motors and gearings that provide the necessary acceleration for the sport while fitting within this specific wheel diameter.

3. Center of gravity and stability:

The regulations regarding seat height (max 58cm to 63cm, depending on player classification) and rear caster height (no more than 2cm from the surface) create a narrow window for our lean-control sensitivity. Since our design relies on “lean” to drive, we must ensure that the “tilt” required to trigger the sensors does not cause the seat rail to exceed the 21-inch height limit or cause the rear anti-tip casters to touch the ground prematurely, which would stall the chair's agility.

4. Integrated Safety:

The NWBA rules regarding “no sharp edges” and “protection of the floor” mean that our motorized drivetrain and battery housing must be fully encased in non-marking, rounded protective shrouding to prevent leaking battery fluids or metal edges from damaging the court or injuring participants.

9. Specifications:

Table 5: Specifications, descriptions, and justifications for Lean Controlled Wheelchair

Spec ID	Need ID Ref.	Description	Justification	Required
A0	1, 3, 5, 10, 11	The wheelchair shall be validated to satisfy all required performance-related specifications for a rider mass of 150 lb.	Several required specifications depend on total moving mass, including acceleration, stability, braking, and runtime. Defining a baseline rider mass prevents the specifications from being interpreted as applying to all possible users without additional validation. A 150 lb rider mass is used as the design and analysis basis for the primary client application, providing a clear condition for prototype verification.	Y
A	1,7	The wheelchair shall withstand representative expected sport-contact loading during prototype testing with 0 fractures, 0 critical components detaching, and 0 permanent deformations that cause failure of any applicable safety, stability, or control specification.	The wheelchair will be used in a high-contact athletic environment where repeated impacts and loading are expected during normal use. Structural failure or permanent deformation could create immediate safety hazards and make the system unusable during play.	Y
B	1,2	Upon sensor fault, controller fault, propulsion power interruption, or triggering of emergency stop device, the wheelchair shall automatically enter a safe state within 0.10 s and reduce commanded wheel speed to less than 5% of maximum within 0.50 s.	The wheelchair must fail safely during electrical or control faults to reduce the risk of unintended motion. A rapid transition to a safe state is necessary because the device may be operating at speed in close proximity to other players.	Y
C	1,5	The wheelchair shall remain upright during maximum acceleration, maximum braking, and maximum turning, with no wheel lifting more than 25 mm off the ground for longer than 0.50 s.	Stability is essential for safe athletic use because tipping or excessive wheel lift can cause loss of control, user injury, or interruption of play. The 25 mm wheel-lift threshold was selected as a conservative engineering limit to identify unacceptable instability during aggressive maneuvers.	Y

D	1,6	The wheelchair shall remain upright and 100% operable after expected frontal or lateral sport contact, and shall not allow dislodging of seated users during such events.	Collisions with other players and wheelchairs are expected during court sports. The system must preserve upright orientation and user containment during routine contact so that normal gameplay does not create unacceptable safety risk.	Y
E	1	The wheelchair shall have 0 exposed sharp edges, burrs, or accessible pinch points in normal operation, and no user-contact surface shall exceed 43°C during operation.	The user will remain in direct physical contact with the wheelchair throughout operation. Eliminating cut, pinch, and burn hazards is necessary to ensure the system is safe for repeated athletic use. [28]	Y
F	3	The wheelchair shall achieve an average forward acceleration of at least 2.85 m/s ² from rest over the first 5 m of travel.	Wheelchair basketball sprint studies report a 5 m sprint time of approximately 1.87 s for competitive players, corresponding to an average acceleration of about 2.86 m/s ² [29].	Y
G	3	The wheelchair shall achieve a governed maximum forward speed of at least 11.0 mph and no greater than 13.5 mph.	Elite wheelchair basketball sprint testing reports peak velocities of approximately 5.3 m/s (≈11.9 mph). A target of 11 mph is therefore slightly conservative relative to elite performance [30].	Y
H	3,4	The wheelchair shall achieve an angular velocity of at least 200 deg/s within 0.50 s from neutral motion.	Elite wheelchair basketball sprint data show athletes reach approximately 3.4 to 4.0 m/s within the first 5 m [30], and wheelchair court-sport agility data report turning radii of approximately 0.94 to 0.98 m [31]. These values correspond to turning rates on the order of 200 to 240 deg/s, so 200 deg/s was selected as a conservative, data-driven target.	Y
I	4	The wheelchair shall provide 100% normal propulsion and steering control through torso-lean input alone during standard athletic operation, without requiring hand propulsion for commanded motion.	The primary purpose of the design is to allow the user to control the wheelchair through torso lean so that the hands remain available for athletic activity.	Y

J	4	The lean-control system shall be calibratable for a given user in 5 min or less.	The control system must be adjustable to the user's available torso motion without requiring excessive setup effort before operation.	Y
K	4	The lean-control system shall detect and respond to intentional torso-lean input within 0.10 s of threshold crossing.	Responsive control input is necessary for intuitive and effective maneuvering during athletic operation.	Y
L	8	The wheelchair shall be 100% manually movable by a caregiver when unpowered on a level indoor surface.	The wheelchair must remain transportable and repositionable when electrical power is unavailable.	Y
M	8	There shall be a backup method for controlled low-speed repositioning of the wheelchair at speeds of at least 1.0 mph when the primary lean-control input is unavailable or disabled.	A backup mobility function is necessary to preserve basic controlled movement during troubleshooting or primary-control failure.	Y
N	9	In its transport configuration, the wheelchair shall have a maximum width of 31.5 in.	The wheelchair must be capable of passing through standard accessible doorway openings during transport. [32]	Y
O	10	The wheelchair shall support at least 2.0 h of normal athletic use per session using onboard battery power, without requiring off-board power during operation.	The wheelchair must support a full practice or game session without interruption so that battery limits do not prevent normal use.	Y
P	11	From maximum operating speed, the wheelchair shall come to a controlled stop in 3.0 s or less on a clean indoor court surface without prolonged uncontrolled skidding or loss of directional stability.	Controlled braking on indoor court surfaces is necessary to maintain user safety and predictable maneuvering during athletic operation. The 3.0 s threshold was selected as a conservative engineering target for safe stopping during sport use.	Y
Q	12	After 2.0 h of representative indoor court operation, the wheelchair shall cause no visible gouging, chipping, tearing, or persistent tire marking on the court surface.	Gym floors are an important stakeholder concern. The wheelchair must be usable in athletic facilities without causing unacceptable surface damage that could prevent adoption.	Y
R	13	The seating system shall provide pressure-management support suitable for prolonged wheelchair use and shall allow the user to perform an	Because the user is at elevated risk of pressure injury during prolonged sitting, the seating system must support routine pressure redistribution and compatible	Y

		independent forward or lateral pressure-relief maneuver without leaving the chair.	pressure-relief maneuvers. [33] [34]	
S	14	The wheelchair shall be capable of setup from transport state to ready-to-use state in 5 min or less without tools.	The wheelchair must be prepared for use efficiently without specialized equipment.	Y
T	14	The wheelchair shall be capable of takedown from ready-to-use state to transport state in 5 min or less without tools.	The wheelchair must be returned to transport condition efficiently without specialized equipment.	Y
U	15	100% of the propulsion batteries, motor controllers, and required electronics shall be mounted on-board the wheelchair during normal operation, with no external trailing power cables or off-board battery pack.	The wheelchair must function as a self-contained powered system during athletic use. External power hardware would create safety, mobility, and usability issues.	Y
V	16	The wheelchair shall allow adjustment of seat width, footrest position, or seat-back position to accommodate users with body dimensions varying by at least 20% from the primary client baseline.	Limited adjustability improves long-term usefulness and allows the system to remain functional as the user grows or if minor fitting changes are needed.	N

10. Ethical and Social Analysis:

The primary ethical lens for this project is virtue ethics. In this context, virtue ethics emphasizes responsible design choices that reflect prudence, care, and a strong commitment to the safety of both the user and the people around them. This has been a central concern throughout the design process. Because the wheelchair is intended for competitive athletic use, the design must balance speed and responsiveness with stability and controllability. For that reason, the team has focused on limiting unsafe behavior through features such as a maximum speed limit, an emergency shutoff, movement only when a user is detected in the chair, and power cutoff in the event of a tip or flip. These measures reflect the importance of designing the system in a way that protects the user even in worst-case scenarios. This emphasis on anticipating hazards and implementing protective measures is consistent with Deuteronomy 22:8, which describes the responsibility to prevent foreseeable harm through thoughtful design and precaution.

The two secondary ethical lenses are rights-based ethics and utility-based ethics. Rights-based ethics emphasizes the user's right to reasonably safe equipment and the family's right to understand the risks and behavior of the system being used. This reinforces the importance of adhering to relevant safety standards, verifying calculations and specifications, and maintaining transparency with the user and family about how the wheelchair operates and what its limitations are. Utility-based ethics focuses on achieving beneficial outcomes for the user. This perspective also aligns with 1 Corinthians 10:24, which encourages individuals to seek the good of others rather than acting only for personal benefit. In this project, that includes allowing the athlete to move without using their hands for propulsion so they can focus more fully on gameplay, ball handling, and rapid response to changing conditions on the court.

Together, these ethical lenses show that the design must support safe operation for the user, reduce unnecessary risk to other players and the surrounding environment, and preserve fair participation in sport. Fairness is especially important because a system that behaves in unexpected or poorly understood ways could create confusion or increase the risk of injury during competition. Proverbs 11:1 also reinforces the importance of fairness and integrity, supporting the need for predictable and equitable system behavior during athletic participation. Overall, the design process has aimed to prioritize user safety, protect others who interact with the wheelchair, and provide a clear and accessible safety mechanism for emergency situations.

11. Design Description:

11.1 System Overview

Our modifications to the wheelchair are organized into two major subsystems: the electrical system and the mechanical system. These tightly integrated systems work together to support intuitive, fully lean-controlled, and safe operation of a sports wheelchair. Figure 6 shows a render of the full CAD model, including the motors and battery.

The electrical system is responsible for central processing, lean-input sensing, user configuration, manual input detection, and electrical energy storage. The mechanical system mounts the drive components to the existing wheelchair, transfers torque from the motors to the wheels, and preserves the structural and transport features needed for athletic use.

To detect lean inputs, the design uses a pressure sensor array on the seat. The pressure sensors provide values that correspond to the operator's pressure distribution, which the controller uses to estimate lean direction and magnitude. These inputs are directed to the Teensy 4.1 microcontroller for processing. Other electrical inputs include the power switch, user adjustment dials for maximum speed and acceleration, and a calibration button. An emergency stop, implemented through a modified circuit breaker, forms the main disconnect between the battery and the electrical system. The block diagram also accounts for disturbance inputs such as collisions with other players on the court. To address these disturbances, the control algorithm may use filtering, damping, or closed-loop compensation to reduce unsafe or unintended motion.

Once the controller has processed the inputs, it determines the motor commands required to produce the desired wheelchair motion. The motor controllers draw power from the battery and regulate power delivery to the motors, forming the main connection between the electrical and mechanical systems. The selected motors are paired with a 7:1 planetary gearbox and an 80:36 timing-pulley reduction, producing an overall gear ratio of 15.55:1. Torque is transferred through the belt drive to the dog coupling mechanism and then to the wheels, producing the wheelchair motion shown as the output of the block diagram.



Figure 6: Overall design of the Lean controlled Wheelchair

11.2 Functional Block Diagrams

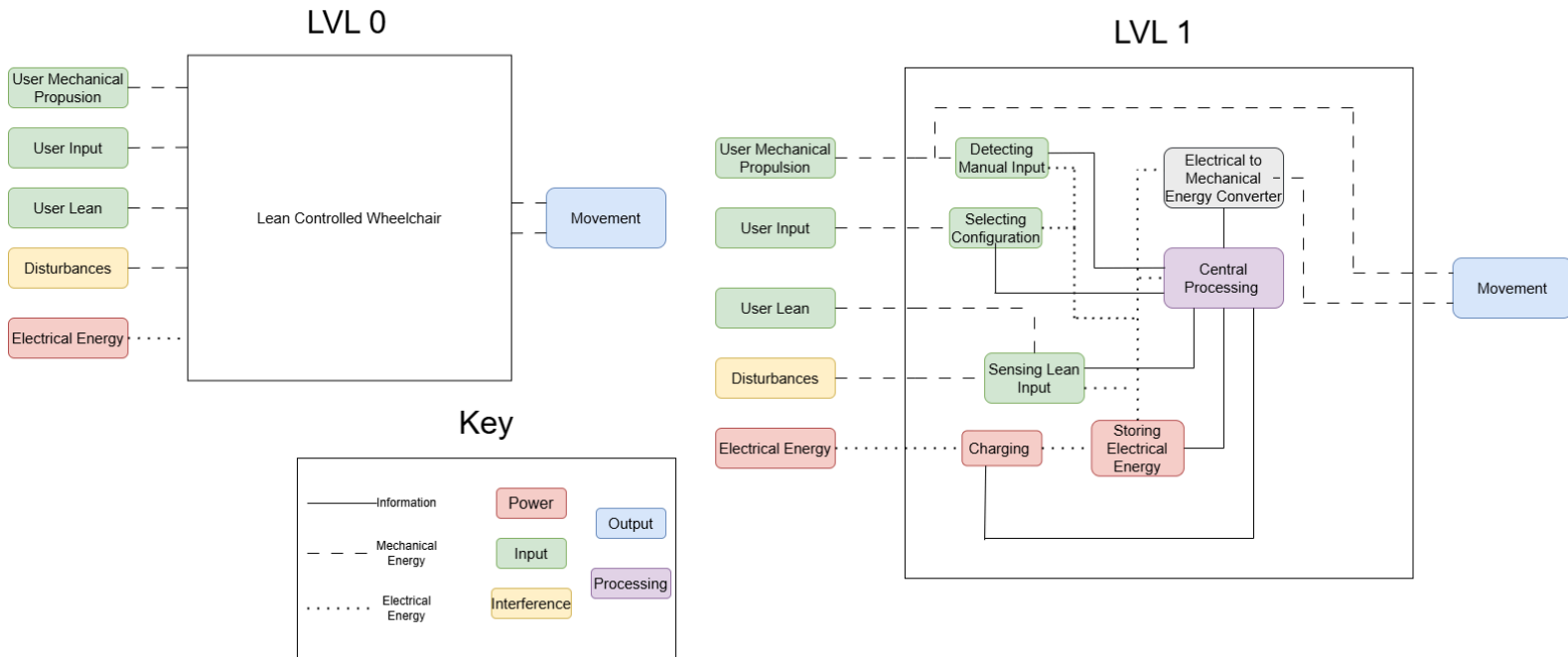


Figure 7: Level 0 and 1 block diagrams

The functional block diagram seen in Figure 7 serves as a conceptual roadmap for the wheelchair's development. By breaking down the system into its core functions, it is possible to map the flow of energy and information through the chair before focusing on individual components. This approach helps connect the selected design back to the required system functions without restricting the explanation to only one component or subsystem.

Level 0:

The Level 0 diagram treats the wheelchair as a single system interacting with its environment. The main goal at this stage is to identify the major inputs the wheelchair must handle in order to produce controlled motion. In addition to user lean and electrical energy, the diagram includes disturbances such as player contact, external impacts, vibration, and changes in wheel-ground traction. Accounting for these inputs early helps ensure that the system architecture considers stability and predictable motion during realistic athletic use.

Level 1:

The Level 1 diagram provides a more detailed view of the internal system logic. It shows how the central processing unit coordinates the user inputs, configuration inputs, lean-sensing inputs, stored electrical

energy, and motor-output commands. In this flow, user lean is sensed and translated into a command, which then determines how the motor controllers use stored electrical energy to produce motion.

This functional breakdown is useful for identifying where safety behavior must be implemented. For example, if the processing unit detects a sensor fault or unsafe input condition, the system must stop sending drive commands or otherwise prevent unintended motor output. In this way, the diagram provides a framework for checking that each subsystem supports safe operation during high-intensity athletic use.

Level 1 Functional Blocks Descriptions:

1. Detecting Manual Input:

Detects manual user inputs, such as buttons, switches, or adjustment controls, and sends those signals to central processing.

2. Selecting Configuration:

Determines the current operating configuration, such as powered mode, manual mode, calibration mode, or other user-selected settings, and sends that state to central processing.

3. Sensing Lean Input:

Converts user body lean into electrical signals that central processing can interpret as motion commands.

4. Charging:

Takes in electrical energy from an external charging source and transfers it to the electrical energy storage system.

5. Storing Electrical Energy:

Stores electrical energy on board the wheelchair and supplies power to the control electronics, sensors, motor controllers, and drive motors.

6. Electrical to Mechanical Energy Converter:

Converts stored electrical energy and motor-control commands into mechanical output for wheelchair motion.

7. Central Processing:

Receives manual input, configuration input, and lean-sensing input, then determines the motor commands required to produce the desired wheelchair motion.

11.3 Electrical Overview

The electrical system senses user lean, processes operator inputs, manages safety controls, and commands power delivery to the drive motors. The major electrical components include the seat-mounted pressure

sensor array, analog-to-digital converters, Teensy 4.1 microcontroller, user-interface controls, VESC motor controllers, BLDC motors, emergency stop, and onboard battery.

11.3.1 Lean-Sensing System

The lean-sensing system uses a 4x4 pressure sensor array placed on either the top of or below the seat cushion, depending on the results of prototyping. These 16 sensors collect pressure-distribution data with enough resolution for lean detection without adding unnecessary complexity or latency to the system.



Figure 8: Example pressure detection using a 4x4 array.

[Try the interactive viewer:](https://calebhottesportfolio.netlify.app/assets/senior%20design/fsr_centroid_interactive_enhanced)

https://calebhottesportfolio.netlify.app/assets/senior%20design/fsr_centroid_interactive_enhanced

The pressure sensor array estimates the user's pressure centroid across the seat. As the operator leans, the force distribution shifts across the array, allowing the controller to infer lean direction and relative lean magnitude. An example depiction of this force distribution is shown in Figure 8.

The farther the pressure centroid shifts from the neutral position, the farther the operator is leaning. The selected lean sensors are capable of detecting small pressure changes, making the system precise, inexpensive, and replaceable. The sensors can also be repositioned on the cushion to better match different operator sizes. For a smaller operator, the sensors can be placed closer together; for a larger operator, they can be spread farther apart while still remaining under the seated user.

11.3.2 Control Hardware and User Interface

Figure 9 shows the major electrical connections between the lean sensors, analog-to-digital converters, Teensy 4.1 microcontroller, user-interface controls, VESC motor controllers, and battery. This schematic provides the electrical-system layout used to supply power and information throughout the electrical system.

The lean-control sensors connect to 16-bit analog-to-digital converters, which convert the sensor voltages into digital values that can be interpreted by the microcontroller. As discussed in the sensor-system analysis section, the selected sensors can withstand the load from the operator and produce voltage outputs within the interpretable range of the ADCs. The resulting signal resolution and measured latency of approximately 18 ms support the response requirements for lean-based control.

The Teensy 4.1 microcontroller serves as the central controller for the electrical system and will be housed in an electrical enclosure to protect the wiring and connections. The Teensy has enough GPIO capacity for the planned lean sensors and user-interface inputs, while still leaving room for later feature expansion. It operates at a 600 MHz clock rate and can handle interrupts effectively, which is useful for a system that must respond quickly to user input and safety-related events.

The Teensy is supplied with 5 V power and provides a 3.3 V source for compatible sensors, buttons, and ADCs. The 16-bit ADCs transmit sensor data to the Teensy through an I2C bus. After processing the input data, the Teensy communicates motor commands to the VESC motor controllers through a CAN bus using a CAN transceiver.

11.3.3 Motor Controllers and Power Distribution

The design includes a personalization panel to allow the operator to adjust wheelchair behavior within the safe performance limits of the system. This panel includes controls for maximum velocity and acceleration, allowing the chair to be tuned for different users, practice settings, or competition levels. For example, a less experienced user may operate the wheelchair with reduced speed and acceleration limits so that the chair remains comparable to the abilities of nearby competitors.

In addition to the personalization controls, the electrical system includes an emergency stop, power button, and calibration button. The emergency stop is implemented using a modified circuit breaker located between the battery and the rest of the electrical system. Activating the emergency stop removes battery supplied power from the system, preventing further motor output. The circuit breaker also provides overcurrent protection for the electrical system.

The power button provides a non-emergency way to turn the wheelchair on and off, allowing the user to safely enter, exit, or reset the chair. The calibration button resets the neutral lean position by zeroing the sensor readings while the operator is seated in a neutral posture. This reduces drift and allows the operator to recalibrate the chair in the event that drift occurs.

In addition to the physical emergency stop, the software will stop sending drive commands if it detects a sensor fault, motor fault, motor-controller fault, or other unsafe input condition. This supports the fail-safe behavior required for safe operation during athletic use.

11.3.4 Battery Integration

The design uses two VESC 6 motor controllers, one for each drive motor. These controllers regulate power delivery from the battery to the brushless DC motors based on commands from the Teensy microcontroller. This makes the motor controllers the main electrical interface between the low-power control system and the high-power propulsion system.

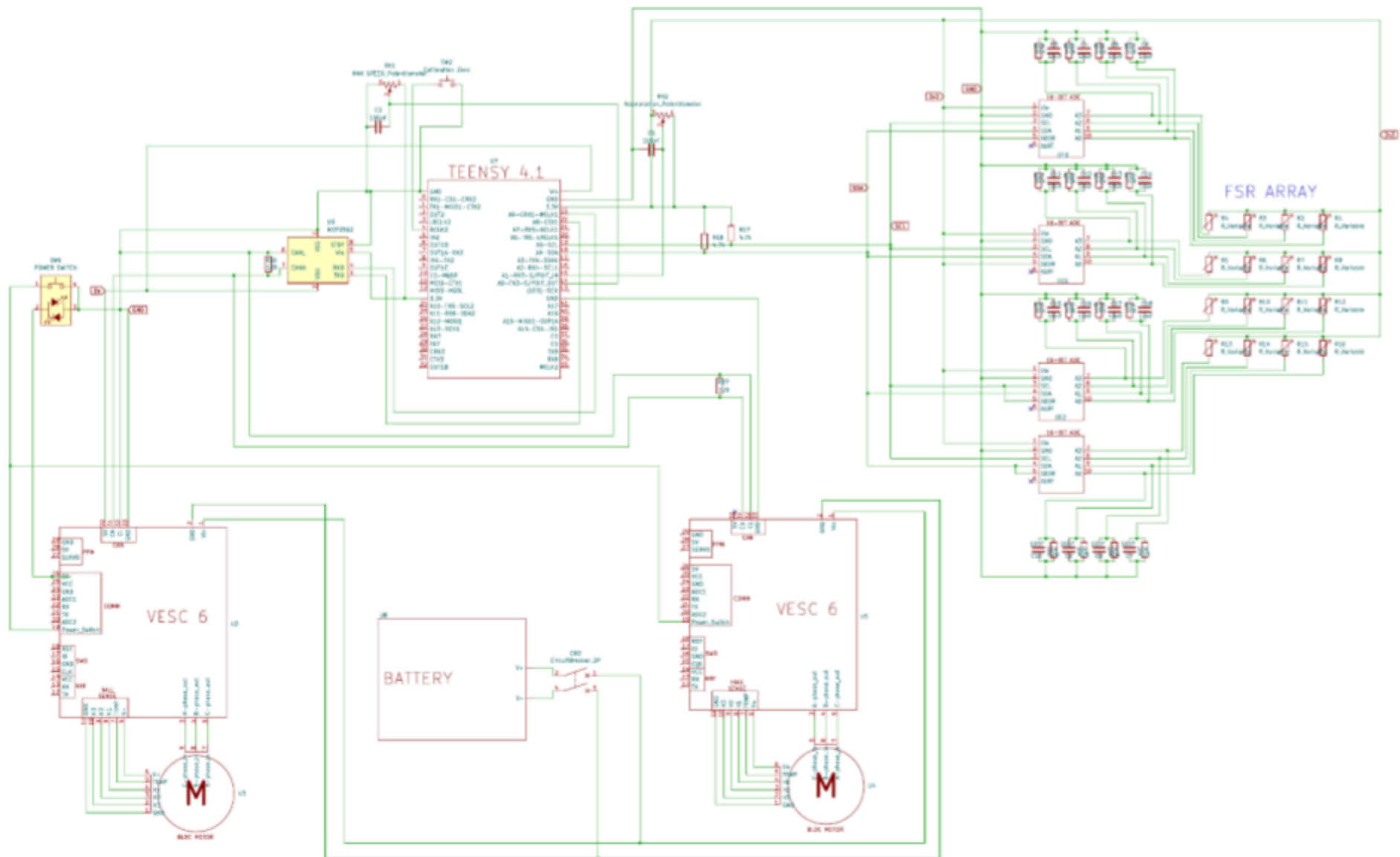


Figure 9: Schematic of the Electrical Components

The VESC controllers also connect directly to the hall sensor wires from the BLDC motors. This allows the controllers to monitor motor position and improve motor-control performance without requiring the Teensy to handle low-level motor commutation. Reducing this processing burden allows the Teensy to focus on interpreting user input, applying control logic, and managing safety behavior.

Each VESC controller also includes onboard sensing features, including an internal IMU. These signals may be useful for detecting abnormal wheelchair behavior such as excessive tilt, impact response, or loss of traction, although the final use of these features will require further validation during testing. The combination of motor-control capability, feedback options, and existing documentation makes the VESC 6 a practical controller choice for the lean-controlled wheelchair.

11.3.5 Battery and Motor Integration

In compliance with Spec U, all propulsion batteries, motor controllers, and required electronics are mounted on board the wheelchair during operation. The selected battery is a Bioenno Power lithium iron phosphate battery with a nominal voltage of 48 V and a capacity of 30 Ah. This battery includes a built-in battery management system and is evaluated in Section 12.3 for its ability to support the runtime requirement in Spec O.

The battery is rated for 30 A continuous current and 60 A peak current for short-duration loading. These current limits are important because the wheelchair is expected to experience brief high-power demands during acceleration, turning, and braking events. The electrical design must therefore ensure that motor-controller settings and operating limits remain compatible with the battery's allowable current output.

The propulsion system uses two 450 W brushless DC motors. Brushless DC motors were selected because they provide good efficiency, reduced internal friction, and improved longevity compared with brushed DC motors. Since the system already uses a DC battery supply, BLDC motors also simplify the overall power distribution. Operating at 48 V reduces the current required for a given power level compared with lower-voltage systems, which helps reduce electrical losses.

11.3.6 Braking

The design will also utilize the motors for braking. This braking is achieved by generating torque with the motors to oppose the wheel rotation to reduce speed in a controlled manner. This reduces the design complexity and overhead cost that would have occurred with a mechanical braking system. This approach supports Spec P by allowing safe and powerful deceleration.

11.4 Mechanical System

11.4.1 Mechanical System Purpose

The mechanical system provides the structural interface between the existing sports wheelchair and the added propulsion system. Its primary functions are to mount the motors and drivetrain components, transfer torque from the motors to the wheels, preserve quick wheel removal for transport, and maintain the stability and durability required for athletic use.

The mechanical design directly supports the specifications related to impact durability, stability, user safety, transportability, and manual handling. The motor and wheel mounting system is designed to be rigid enough to withstand expected sport-contact loading while still allowing the wheels to be removed without tools. The original caster layout is retained to preserve the wheelchair's resistance to tipping, and the final design will include edge finishing and shielding where needed to eliminate exposed sharp edges, burrs, and accessible pinch points.

11.4.2 Wheelchair Frame Interface

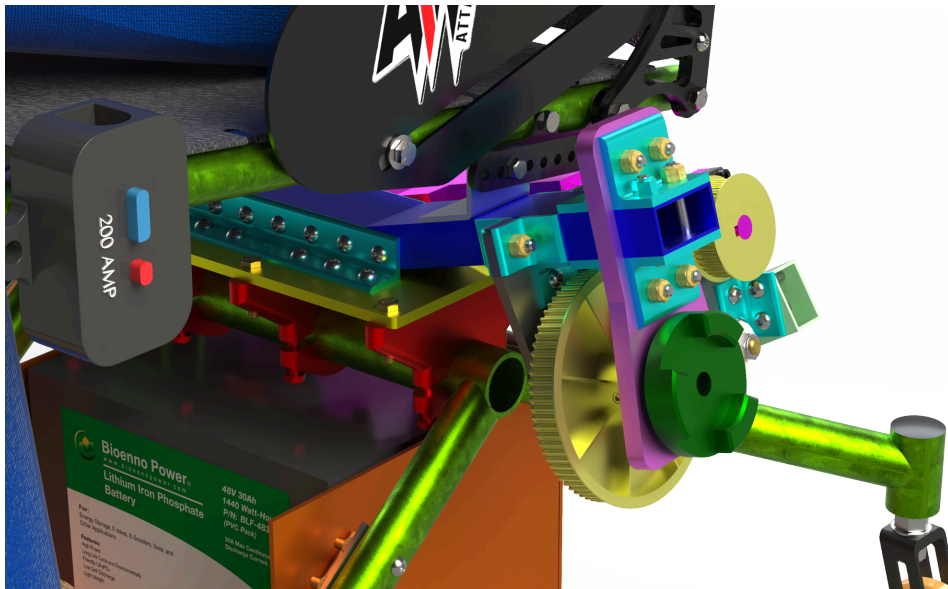


Figure 10: CAD Render of the Motor Mount

The mechanical assembly attaches to the existing wheelchair through four custom clamps, shown in red, and two braces in teal, with two clamps mounted on each side of the frame and the braces in the back. The clamps attach to the main frame tubes and provide flat mounting surfaces for the added drivetrain structure. The clamps are fastened using 1/4-20 bolts, matching the fastener standard already used throughout the existing wheelchair where practical.

The base plate, shown in gold, attaches to the clamps and supports the mitered box-tube structure and angle brackets. These components are bolted together rather than permanently welded wherever practical. This approach preserves modularity during prototyping, allows components to be removed or adjusted, and reduces the risk associated with finalizing a welded geometry before testing. The blue box tube carries loads from the wheel and seat-support structure, so the connection area between the base plate and tube is sized to distribute loading over a relatively large contact region.

The existing wheelchair uses a configurable front geometry with ball joints that allow the seat height and angle to be adjusted. In the modified design, the original vertical connector tube is replaced by a sheet metal plate welded to the added boxtube structure.

11.4.3 Motor and Drivetrain Layout

Several motor placement options were considered, including direct drive at the wheel, mounting the motors under the front of the seat, and mounting the motors behind the main tube in the rear of the wheelchair. Direct drive was rejected because it would require significant structural modification to the existing wheelchair and would require a much more compact motor and gearbox package than the current design. Front mounting was also considered early in CAD development, but it created packaging conflicts with the battery, reduced available space under the seat. The rear-mounted motor layout was selected because it provides a space efficient method for packaging the motors, drivetrain, battery, and frame supports while preserving the core geometry of the existing wheelchair.

Three gearbox approaches were considered: spur, planetary, and worm gears. Worm gearing was rejected because it is impossible to backdrive, which precludes the ability to have manual movement. A custom spur gearbox would be possible, but would add design, manufacturing, lubrication, and safety complexity. A commercial off-the-shelf planetary gearbox was selected because it is compact, self-contained, professionally manufactured, and already matched to the selected motor.

The selected propulsion system uses two 450 W brushless DC motors. On top of the motors is a 7:1 planetary gearbox which feeds into a 80:36 timing-pulley reduction, producing an overall gear ratio of 15.55:1. Based on the motor and drivetrain performance analysis in Sections 12.2.2 through 12.2.5, this configuration is intended to provide the torque and speed needed to satisfy the forward acceleration requirement in Spec F, maximum speed requirement in Spec G, turning performance requirement in Spec H, and motor-braking requirement in Spec P.

Torque is transferred from each motor to the wheel through a 15 mm wide, 5 mm pitch HTD timing belt. The motor-side pulley has 36 teeth, and the wheel-side pulley has 80 teeth. A belt drive was selected because it provides high torque capacity, low noise, low maintenance, and minimal slip when properly tensioned. Compared with a chain drive, the timing belt is better suited for this compact layout because it avoids chain lubrication, reduces noise, and provides smoother operation over the short center distance used in the design.

11.4.4 Motor Mount and Support Structure

The motor and wheel-side drivetrain components are mounted to the same motor mount plate to maintain alignment between the gearbox output shaft, belt drive, dog coupling, and wheel axle. Keeping these components on a common plate helps ensure that the pulley axes remain parallel and that belt alignment is maintained during operation.

The motor mount plate attaches to the main boxtube structure through custom angle brackets and supporting hardware. The clamps are designed to provide a strong connection to the existing wheelchair frame, but the design also includes an additional support beam as a secondary load path. This support beam provides a second point of contact with the wheelchair frame and reduces the moment loads that would otherwise be carried primarily by the clamps. This improves the stiffness and robustness of the drivetrain mounting system during impacts, acceleration, braking, and turning.

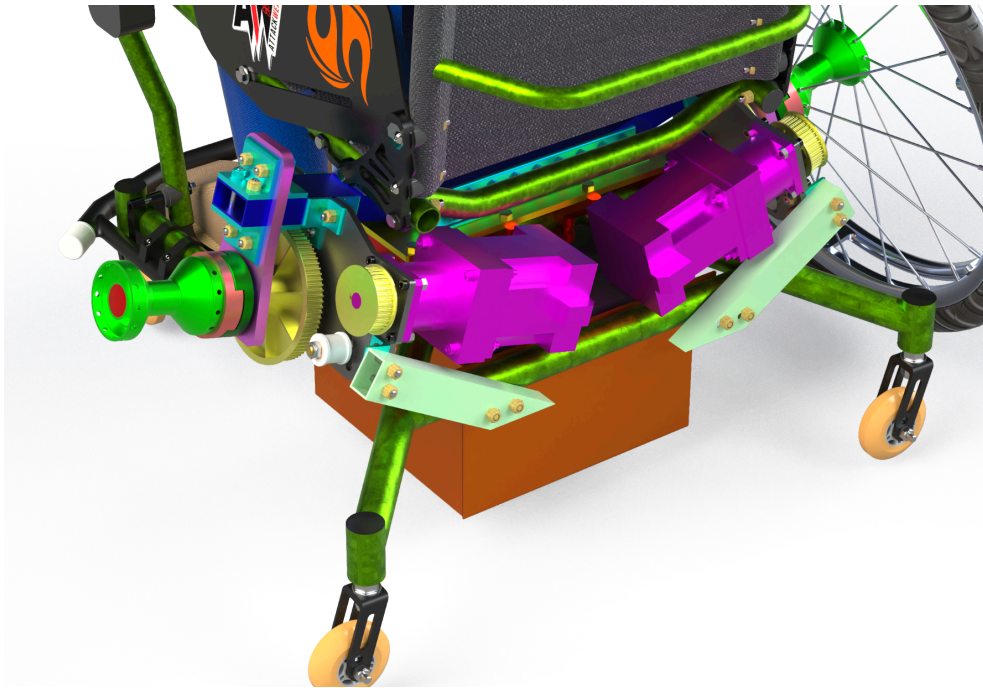


Figure 11: CAD Render of Back of Motor Mount, left wheel spokes and rim hidden

The support structure is intentionally designed with removable bolted connections where practical. This allows the prototype to be adjusted during fabrication and testing while still providing the stiffness required for drivetrain alignment and sport-contact loading.

11.4.5 Battery Mount

The battery will be housed in a bent sheet-metal enclosure mounted below the main frame tube. This location keeps the battery away from the user's legs, places the battery near the electrical system, and lowers the overall center of mass of the wheelchair. A low battery location is beneficial because it reduces the tendency of the chair to tip during acceleration, braking, turning, and contact events.

The battery enclosure will attach to the wheelchair frame at four mounting points. This provides a secure mounting location while distributing loads into the frame. The enclosure must restrain the battery during vibration, impacts, and rapid wheelchair motion, while still allowing reasonable access for charging, inspection, or replacement.



Figure 12: Side View of Wheelchair Showing Battery Mount Location

11.4.6 Dog Coupling and Quick-Disconnect Wheel System

The dog coupling system allows each wheel to be removed from the wheelchair without tools while still transferring motor torque to the wheel during powered operation. This system was modeled after the existing quick-disconnect wheel mechanism so that the modified wheelchair preserves the original transport functionality as much as possible.

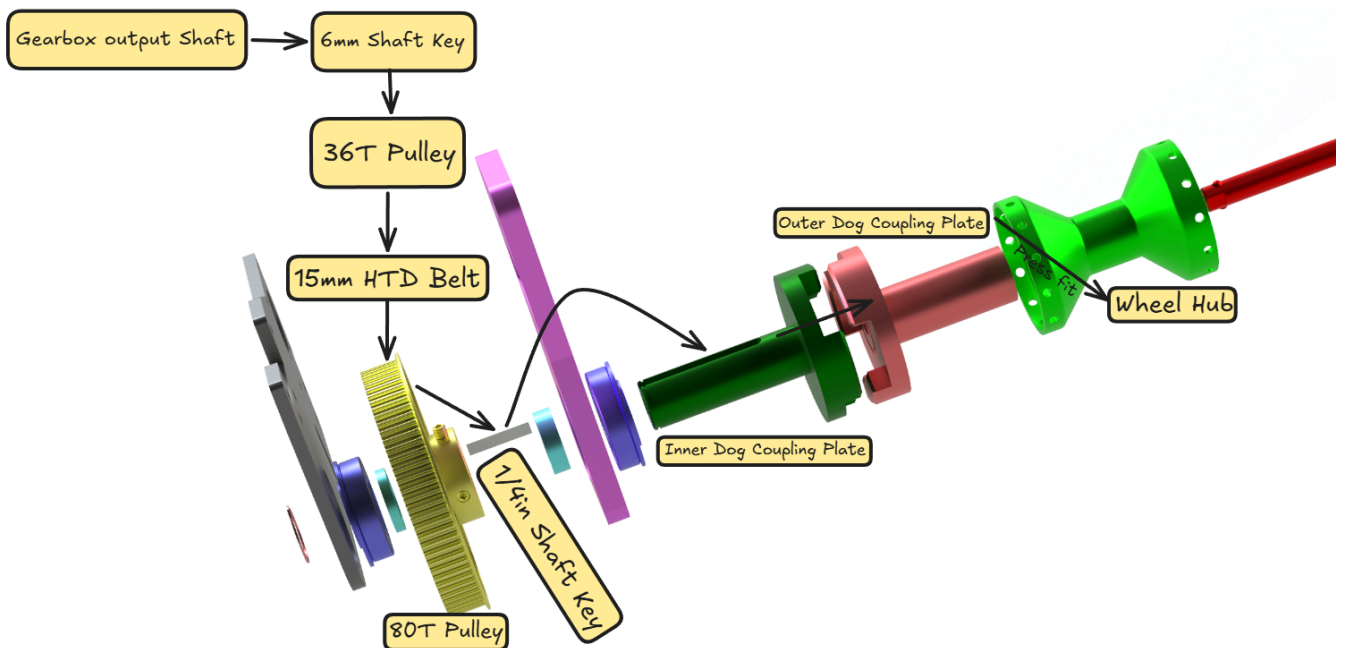


Figure 13: Torque Transfer Path Exploded View

A key design feature is that drive torque is not transmitted through the removable wheel shaft. Instead, torque is transmitted through the keyed pulley and dog coupling plates. This avoids the manufacturing difficulty of using a keyed or hex shaft as both a removable wheel axle and a torque-transfer element. It also preserves the intuitive, toolless quick-disconnect behavior of the original wheelchair.

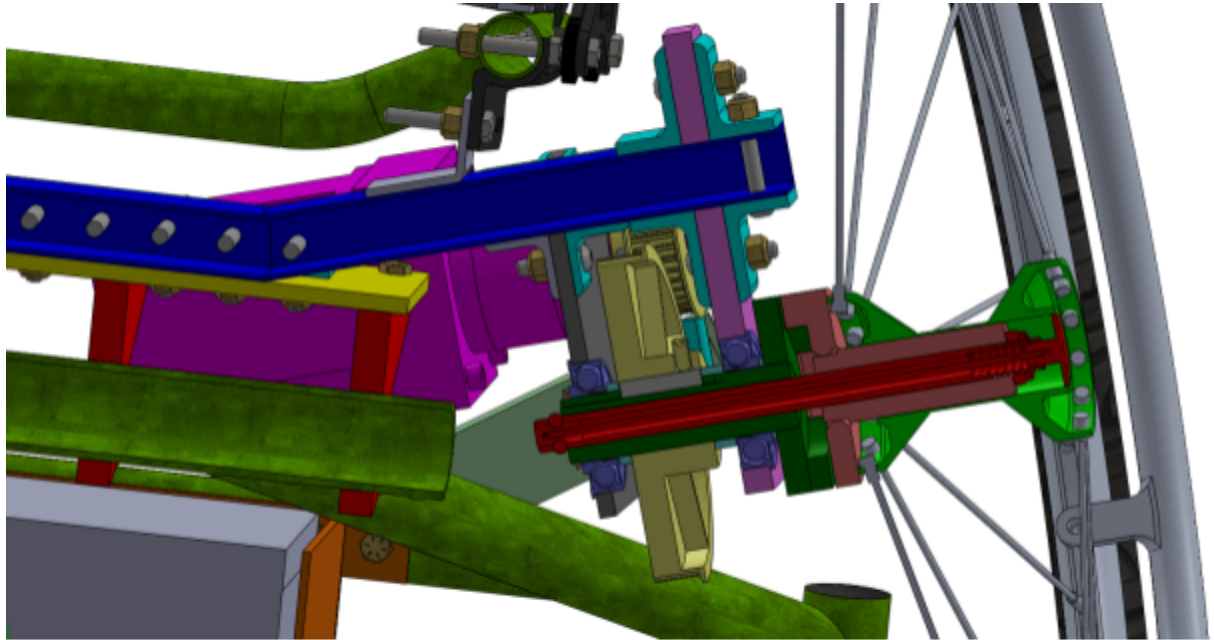


Figure 14: Section View of Dog Coupling Mechanism

When the wheel is removed, the inner dog coupling plate remains supported by bearings in the motor mount assembly. The wheel-side pulley is keyed to the inner coupling plate and retained with setscrews and adjacent spacers. The bearings support the coupling plate radially and keep the drivetrain components aligned when the wheel is not installed.

When the wheel is installed, the removable shaft passes through the coupling assembly and locks in place using the original quick-disconnect mechanism. The outer dog coupling plate is press fit into the wheel hub, while the inner and outer dog coupling plates engage to transfer torque. The shaft primarily supports wheel positioning and loading, while the dog coupling transfers motor torque into the hub. This allows the wheel to remain removable while still supporting powered operation.

The dog coupling plates are custom steel components. The design uses 3 in diameter steel round stock for the coupling body and a smaller welded shaft section where needed. This approach reduces wasted material compared with machining the entire part from a single oversized piece of stock. The coupling teeth will be CNC machined, and the shaft and coupling body will be welded using a custom fixture to maintain alignment.

The coupling geometry is inspired by the function of commercially available jaw-style couplings, such as Lovejoy couplings, but the design omits the elastomer insert because this application requires minimal backlash. The wheelchair operates at low rotational speed, transmits relatively high torque, and changes

direction frequently. For that reason, minimizing backlash is important for responsive control and reducing impact loading during direction changes.

A three-tooth coupling pattern was selected as a balance between strength, manufacturability, and engagement reliability. The inner coupling plate includes a 1 in outer shaft section and a 0.5 in bore for the removable shaft. The outer coupling plate is similar, but its outer diameter is sized to interface with the existing wheel hub. The final press-fit diameter may require adjustment after precise measurement of the hub bore.

The quick-disconnect shaft will be a modified version of the original commercially available disconnects. The only difference is that our version is significantly longer. The original QD will be extended by cutting the old one and welding additional material in the middle. This will also require us to extend the inner spoke which connects the button to the plug at the end. The shaft includes a normally locked spring-loaded mechanism that holds the wheel in place. When the release button is pressed, the locking bearings retract, allowing the shaft and wheel to be removed.

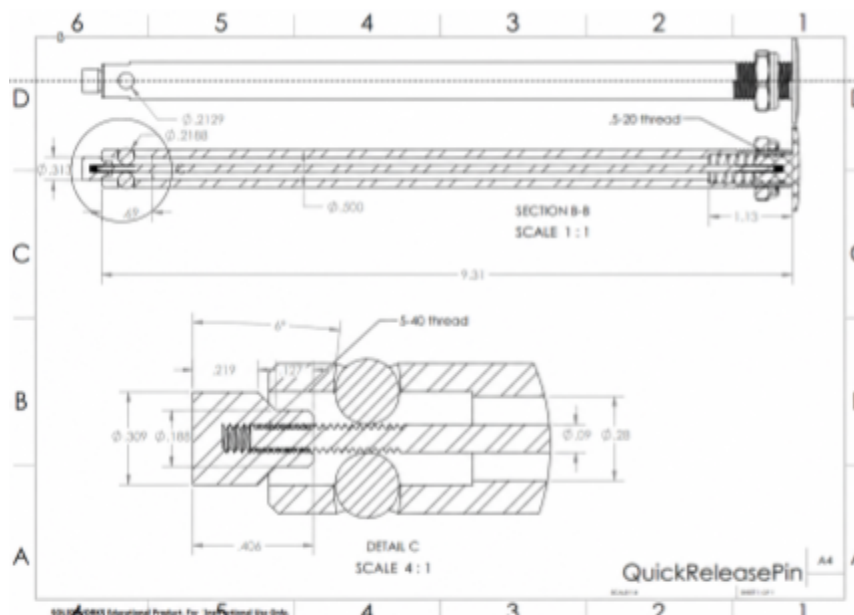


Figure 15: Drawing of the Quick Disconnect

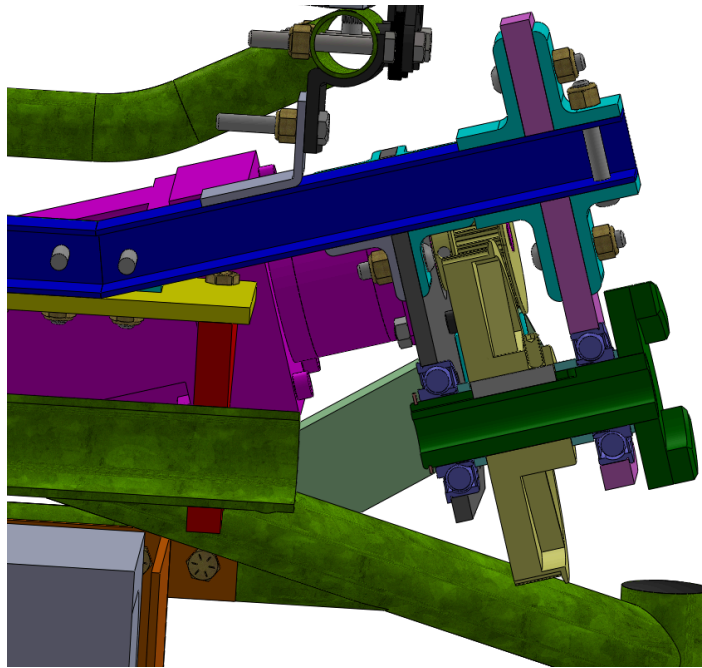


Figure 16: Section View of Dog Coupling Mechanism Without Wheel

The pulley selection is driven by the required gear ratio, shaft geometry, and manufacturability of the keyed connection. The 80T wheel-side pulley uses a 1 in bore with a 1/4 in keyway. This key size was selected based on supplier guidance because it is standard for a 1 in bore and can be broached accurately without shims. A smaller keyway would leave more shaft material, but supplier feedback indicated that a nonstandard broached keyway could be less precise and more likely to loosen over time.

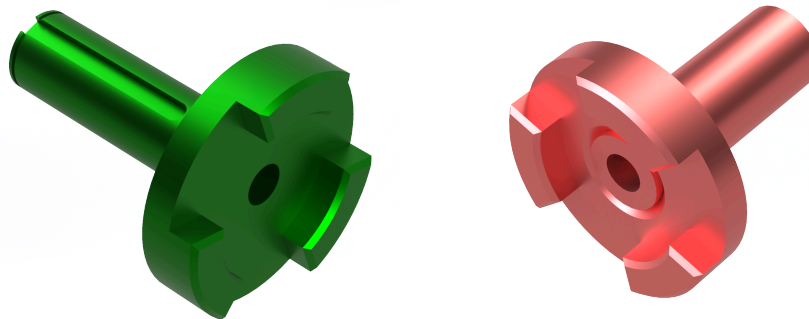


Figure 17: Isolated Dog Coupling View. Left: Dog Coupling Connected to Wheelchair. Right: Dog Coupling Connected to Wheel.

The 80T pulley includes setscrews arranged 90 degrees apart. One setscrew retains the key, while the other presses the pulley against the shaft. This arrangement improves contact compared with placing the setscrews 180 degrees apart. The 36T motor-side pulley mounts to the gearbox output shaft using a 14 mm bore and 6

mm keyway. Together, the 36T and 80T pulleys provide the belt reduction used to reach the final 15.55:1 overall gear ratio.

The drivetrain uses two bearings per side to support the inner dog coupling plate and maintain radial alignment. The bearings provide a low-friction support structure for the rotating coupling assembly and help keep the wheel-side pulley aligned with the motor-side pulley. Roller bearings were selected because they provide a practical combination of load capacity, availability, and ease of mounting for the prototype design.

11.4.7 Stability, Safety, and Transport Features

The design retains the original caster system of the sports wheelchair. These casters act as effective anti-tip supports and help preserve the stability characteristics of the original chair. The added battery mass is located low in the frame, which further lowers the center of mass and supports stability during acceleration, braking, and turning.

The design also retains traditional basketball wheels to reduce the risk of damaging indoor court surfaces. Moving parts, belt-drive components, sharp edges, and pinch-point regions will require shielding or edge finishing before final use. These safety refinements are necessary to satisfy the user-contact and pinch-point requirements of the specifications.

Transportability is supported by the quick-disconnect wheel system. When the wheels are removed and placed on the seat or transported separately, the wheelchair can be reduced to a narrower transport configuration. This supports the doorway-width, setup-time, and takedown-time specifications while preserving the powered drivetrain functionality during normal operation.

12. Design Analysis:

This section evaluates whether the selected design can satisfy the major technical specifications established earlier in the report. The analysis is organized by each subsystem: control system performance, motor and drivetrain performance, battery and power capability, and motor mount structural integrity.

12.1 Control System Analysis

The control system analysis verifies that the sensing, processing, communication, and safety-response portions of the design can support responsive lean-based operation. The analysis focuses on sensor signal quality, filtering, data acquisition timing, control-loop execution, CAN communication, and fault response.

12.1.1 Sensor System Analysis

12.1.1.1 FSR Voltage Divider Signal Range

Each Force Sensitive Resistor (FSR 402S) is paired with a fixed 10 k Ω pull-down resistor. The junction voltage supplied to the ADC is:

$$V_{out} = 3.3V * \frac{R_{pull}}{R_{FSR} + R_{pull}}$$

Table 6: ADS1115 configured at PGA gain 1 (± 4.096 V full scale). LSB = 125 μ V. Positive range: 0 to 32,767 counts:

Condition	Force	R_FSR	V_out	ADC counts (16-bit)
Unloaded	0	>1 M Ω	~0.03 V	~264
Threshold	30g	~200 k Ω	~0.16 V	~1,280
Light seated	1kg	~10 k Ω	1.65 V	~13,200
Average seated	5kg	~1.5 k Ω	2.83 V	~22,640
Max load	10kg	~500 Ω	3.14 V	~25,120

At the 5 kg average operating point, the FSR resistance is approximately 1.5 k Ω , producing a voltage output of 2.83 V. This places the ADC reading in a usable upper portion of its input range without saturating during normal seated loading. At the 10 kg peak condition, representing concentrated loading during a strong lean, the sensor output remains near but below the ADC ceiling.

12.1.1.2 RC Filter Noise Rejection — Spec K

Each FSR signal line has a 10 nF filter capacitor in parallel with the 10 kΩ pull-down. The RC cutoff frequency is:

$$f_c = \frac{1}{2\pi R C} = \frac{1}{2\pi * 10,000 * 10 * 10^{-9}} = 1,592 \text{ Hz}$$

Attenuation at motor PWM frequency (20 kHz):

$$Attenuation = \frac{1}{\sqrt{1 + (20,000/1,592)^2}} \approx 0.079 \rightarrow -22 \text{ dB}$$

Rider weight-shift signals (< 5 Hz) pass with < 0.003 dB attenuation. Motor electrical noise is reduced to approximately 8% of its original amplitude at the ADC input.

12.1.1.3 I2C Bus Timing Validation

The I2C bus operates at 400 kHz (fast mode) with four ADC devices. Estimated bus capacitance with approximately 300 mm of trace is ~130 pF. The 4.7 kΩ pull-up resistors give a rise time of:

$$t_{rise} \approx 0.8473 * R * C = 0.8473 * 4,700 * 130 * 10^{-12} = 518 \text{ ns}$$

The calculated 518 ns rise time is slower than the 300 ns fast-mode target, so the initial 4.7 kΩ pull-up value should be treated as marginal for 400 kHz operation. If bench testing shows unreliable communication, the pull-ups can be changed to 2.2 kΩ, reducing the calculated rise time to approximately 242 ns and bringing the bus within the fast-mode rise-time limit.

12.1.2 Control Loop and Software Architecture

12.1.2.1 Program Overview

The Teensy 4.1 microcontroller runs the control software as a fixed-rate interrupt-driven loop executing at 1 kHz. This means the control logic is triggered by a hardware timer interrupt once every 1 ms rather than running as a free-running loop. The fixed-rate approach ensures consistent timing between sensor reads, centroid computation, and motor commands regardless of how long any individual step takes, which is important for predictable and stable control behavior. Background tasks such as monitoring incoming CAN status frames from the two VESCs are handled in the main program body outside the timed interrupt and do not affect control loop timing.

The software executes in three distinct phases: an initialization phase that runs once at power-on, a recurring 1 ms control loop that handles all sensing, computation, and motor commanding during normal operation, and a fault response routine that can be triggered at any point during the control loop if a safety condition is detected. Figure 18 illustrates the overall program flow. This fixed-loop structure supports predictable sensor

processing and command timing, which is necessary for the wheelchair to feel consistent and controllable during athletic use.

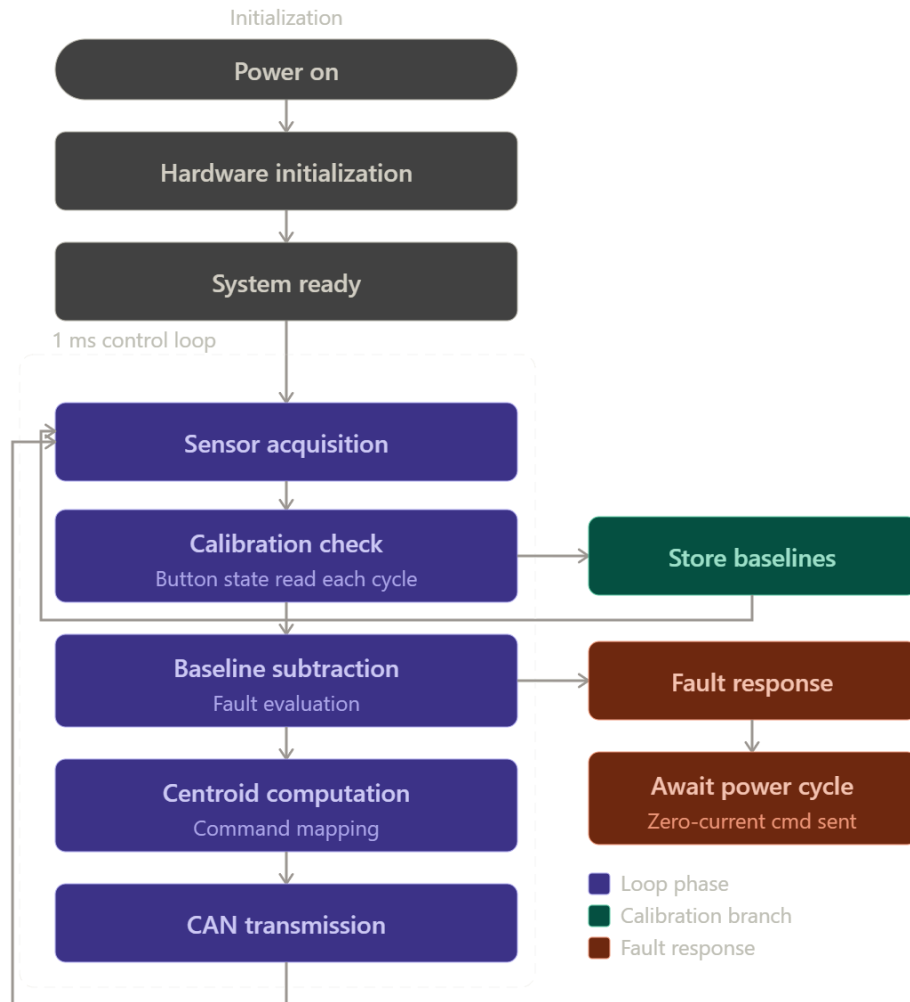


Figure 18 — Software flow diagram showing initialization phase, 1 ms control loop with sequential execution phases, and fault branch

12.1.2.2 Initialization Phase

When the Teensy powers on, the software completes hardware initialization before the control loop begins. This includes configuring the I2C bus for communication with the four ADS1115 ADCs, initializing the CAN peripheral and MCP2562 transceiver for communication with the two VESCs, and reading the initial positions of the speed and acceleration adjustment knobs. Default baseline values for all 16 sensors are initialized to zero, allowing the system to power on without requiring immediate calibration. However, normal use is intended to begin after the rider performs calibration from a neutral seated posture. The illuminated power switch is activated at the end of initialization to signal to the rider that the system is ready.

12.1.2.3 Main Control Loop — 1 ms Execution

Each 1 ms loop cycle proceeds through five sequential phases.

The first phase is sensor acquisition. All 16 FSR readings are collected from the four ADS1115 ADCs over I2C. The ADCs operate in continuous conversion mode so that a fresh reading is always available when the Teensy requests it, avoiding mid-loop conversion wait times.

The second phase is a calibration check. The firmware checks whether the calibration button is currently pressed. If it is, the raw reading of each sensor at that moment is stored as that sensor's baseline value, and the loop advances to the next cycle without computing a centroid or sending a motor command. This prevents a partial or transitional centroid from being transmitted during the snapshot instant. Normal operation resumes on the following cycle.

The third phase is baseline subtraction and fault evaluation. The stored baseline value for each sensor is subtracted from its current raw reading to produce a baseline-corrected value. The corrected readings are summed across all 16 sensors to produce a total cushion load. If this total falls below the ejection detection threshold, the fault response routine is entered immediately, and the remaining phases of the current cycle are skipped. VESC-reported motor temperatures, received via CAN status frames in the background, are also evaluated at this stage. If either motor exceeds the thermal derating threshold, the commanded current limit for that motor is scaled back accordingly.

The fourth phase is centroid computation and command mapping. The baseline-corrected sensor values are used to compute a pressure centroid position as described in Section 12.1.2.7. The centroid's forward-rearward component drives the speed command for both motors, and the centroid's lateral component drives the differential split between left and right motor current. A deadband is applied around the neutral centroid position so that small unintentional shifts do not produce motion. The speed and acceleration knobs are read each loop cycle and applied as scaling factors on the centroid output, giving the rider real-time adjustment of the chair's overall responsiveness without changing the underlying centroid logic.

The fifth phase is motor command transmission. Left and right current commands are encoded into CAN frames and transmitted to the two VESC 6 motor controllers. Both frames are sent within the same 1 ms cycle.

12.1.2.4 Calibration Routine

The calibration button allows the rider to set a personalized baseline that accounts for their individual seated weight distribution on the cushion. Because the FSR sensors respond to absolute force rather than relative changes, riders of different weights or seating positions produce different raw readings in a neutral, upright position. Pressing the calibration button while seated normally captures that neutral distribution as the reference zero for all subsequent centroid calculations. This allows the centroid calculation to reflect intentional lean relative to the rider's neutral posture, rather than simply responding to absolute pressure differences across the cushion. Calibration can be performed at any time without powering the chair off, and

the chair will not move during or immediately after the calibration press because the command transmission phase is skipped for that cycle.

12.1.2.5 Fault Response and Safe State

When a software-detectable fault condition is detected, such as rider ejection from the cushion load monitor, the firmware immediately transmits explicit zero-current commands to both VESCs over CAN before ceasing further command transmission. For a hard emergency-stop activation through the circuit breaker, the circuit breaker directly removes propulsion power from the system as described in Section 12.1.4.1. Sending an explicit zero command rather than simply stopping transmission ensures the VESCs receive an active instruction to reduce output rather than relying on a communication timeout to infer that the controller has stopped. After the zero command is sent, the current decay function described in Section 12.1.4.1 governs the ramp-down of motor output. The system does not automatically resume normal operation following a fault. Power must be cycled to re-enter the initialization phase and return to normal control loop execution.

12.1.2.6 Control Response Latency — Spec K

Spec K requires the control system to detect and respond to intentional torso-lean input within 0.10 s of threshold crossing. The total latency chain from lean threshold crossing to motor current change is:

Table 7: Latency of Control Response

Latency Source	Time (ms)	Notes
ADS1115 conversion time (860 SPS)	1.16 ms	Per sensor
Staggered read cycle (16 sensors)	16 ms max	Worst-case full refresh
Centroid computation	< 0.1 ms	16 multiply-accumulate ops
CAN frame transmission	0.22 ms	Per frame at 500 kbps
VESC command processing	< 1 ms	One PWM cycle at 20 kHz
Total worst-case latency	~18 ms	Well within 100 ms

The worst-case end-to-end latency from lean threshold crossing to commanded current change at the motor is approximately 18 ms. This is 5.5× faster than the 100 ms specification requirement. The primary contributor is the 16 ms staggered sensor read cycle. If lower latency were needed, increasing the read rate to 860 Hz per sensor (each ADS1115 in continuous mode) could reduce this to approximately 5 ms, though this is not required for the current specification.

12.1.2.7 Pressure Centroid Calculation

The 16 FSR sensors are arranged in a 4×4 grid. The pressure centroid is computed as a weighted average of sensor positions, where each weight is the baseline-subtracted sensor reading s_i and (x_i, y_i) is the sensor's physical position on the cushion normalized to $[-1, +1]$:

$$X_c = \frac{\sum(s_i * x_i)}{\sum(s_i)} \quad Y_c = \frac{\sum(s_i * y_i)}{\sum(s_i)}$$

Y_c drives forward speed (deadband $|Y_c| < 0.13$, linear zone up to 0.53, exponential zone to 1.0). X_c drives the differential steering split. The deadband threshold of ± 0.13 was determined from the measured centroid noise floor of a seated rider in a neutral position. This approach converts the 16 individual pressure readings into a two-axis control input while preserving enough simplicity for real-time execution in the 1 ms control loop.

12.1.3 CAN Bus Communication Analysis

12.1.3.1 Bus Timing and Bandwidth

The CAN bus operates at 500 kbps. Each frame takes approximately 222 μ s (111 bits worst-case). Two command frames per 1 ms loop cycle take 444 μ s, leaving 556 μ s for all other processing. CAN bus utilization is approximately 44%, well below the saturation threshold. This indicates that the CAN bus has sufficient bandwidth to send motor commands at the 1 ms control-loop rate while leaving margin for motor-controller status frames and diagnostic communication.

12.1.3.2 Termination Validation

Two 120 Ω termination resistors, one at each end of the CAN bus, produce an effective bus resistance of approximately 60 Ω .

$$R_{term} = 120\Omega || 120\Omega = 60\Omega$$

This matches the expected termination for a properly terminated CAN bus specified in ISO 11898. During assembly, this can be verified by measuring resistance between CAN-H and CAN-L with power off; the measured value should be approximately $60\Omega \pm 10\%$.

12.1.4 Safety System Analysis

12.1.4.1 Emergency Stop and Safe-State Response — Spec B

Spec B requires the wheelchair to automatically enter a safe state within 0.10 s after a sensor fault, controller fault, propulsion power interruption, or emergency-stop activation. It also requires the commanded wheel speed to be reduced to less than 5% of maximum within 0.50 s. In this context, commanded wheel speed refers to the speed command sent from the microcontroller to the motor controllers, not the physical speed of the wheelchair.

The emergency stop is implemented using the main circuit breaker between the battery and the propulsion system. When the circuit breaker is triggered, it opens the battery power path and prevents continued powered motor output. After the circuit breaker opens, the VESC disables output FETs within one PWM switching cycle. At a 20 kHz switching frequency, one PWM cycle is approximately:

This response time is far below the 0.10 s safe-state entry requirement in Spec B. Once propulsion power is interrupted, the motors cannot continue receiving powered drive commands from the battery. The physical wheelchair may continue to coast due to drivetrain inertia and motor back-EMF, but physical stopping distance and stopping time are addressed separately under the braking requirement in Spec P.

For software-detectable faults, such as invalid sensor readings, loss of expected controller communication, or unsafe control input conditions, the firmware reduces the commanded motor output using a decay function. The command decay is modeled as:

$$I_{out}(n) = I_{out}(n - 1) * 0.993 \text{ per } 1 \text{ ms loop cycle}$$

Where n is the number of 1 ms control-loop cycles. The time required for the commanded output to fall to 5% of its initial value is found by solving:

$$n = \frac{\log(0.05)}{\log(0.993)} = \frac{2.996}{0.007} = 428 \text{ cycles} = 428 \text{ ms}$$

Therefore, the firmware reduces the commanded motor output to 5% of its initial value in approximately 428 ms, which is within the 0.50 s requirement in Spec B. This satisfies the commanded-speed portion of Spec B because the control system stops requesting meaningful powered motion from the motor controllers within the required time window.

Spec B is therefore satisfied in two ways. For emergency-stop activation or propulsion power interruption, the circuit breaker provides a hardware safe-state response by removing propulsion power. For software-detectable faults, the firmware provides a controlled command-decay response that reduces commanded output below the 5% threshold within 0.50 s.

12.1.4.2 User Ejection / Seat-Occupancy Detection

The firmware monitors total cushion pressure by summing all 16 baseline-corrected sensor readings. If the total pressure signal falls below 200 ADC counts, the system interprets this as a possible loss of operator contact with the seat and applies the current-decay safe-state response. The threshold was selected to be above the electrical noise floor (< 50 counts measured with cushion unloaded) while well below the minimum loading produced by a seated rider. This threshold will need to be validated during physical testing to ensure normal athletic leaning does not falsely trigger the safe-state response.

12.1.4.3 Thermal Protection

The VESC reports its onboard MOSFET temperature in every STATUS_5 CAN frame at approximately 50 Hz. The Teensy receives these frames via hardware interrupt and stores the values in gTempLeft and gTempRight. The main control loop applies a linear derating function to the commanded current output as the MOSFET temperature approaches its safe operating limit:

$$I_{derated} = I_{commanded} * \max(0, \frac{T_{cutoff} - T}{T_{range}})$$

For VESC MOSFET temperature monitoring, the thresholds are set to:

- $T_{start} = 80^{\circ}C$ — derating begins, current scales linearly downward
- $T_{cutoff} = 100^{\circ}C$ — current commanded to zero, motor stops
- $T_{range} = T_{cutoff} - T_{start} = 20^{\circ}C$

Substituting these values:

$$I_{derated} = I_{commanded} * \max(0, \frac{100 - T}{20})$$

At 80°C the multiplier is 1.0 — no derating. At 90°C the multiplier is 0.5 — current is halved. At 100°C the multiplier is 0 — the output is cut to zero. Between 80°C and 100°C the transition is continuous and proportional, avoiding an abrupt torque step that could destabilise the chair.

Derating is applied independently per VESC. If VESC left reaches 90°C while VESC right remains at 70°C, the left motor is derated to 50% of commanded current while the right motor continues at full output. This asymmetric behaviour is intentional — a symmetric cutback would reduce total drive force unnecessarily when only one controller is thermally stressed.

The VESC 6/75 hardware also has its own independent firmware-level thermal protection that will cut output and set a fault flag if the MOSFET temperature exceeds the VESC Tool configured limit. The firmware derating layer described here acts as a gentler upstream intervention that reduces current demand before the VESC's own hard limit is reached, avoiding sudden fault-state stops during gameplay. This derating strategy protects the motor controllers from overheating while allowing reduced operation before a full thermal shutdown is required.

12.2 Motor and Drivetrain Performance Analysis

The purpose of this section is to verify that the selected motor and drivetrain configuration can meet the wheelchair's major motion-performance specifications. These include the forward acceleration requirement in Spec F, the maximum speed requirement in Spec G, the turning response requirement in Spec H, and the stopping requirement in Spec P. The selected drivetrain uses two 450 W brushless DC motors, each paired with a planetary gearbox and belt reduction.

12.2.1 Motor and Gearbox Selection

The selected motor is the Peaco Support 450 W brushless DC motor in the 48 V PLF-060 planetary gearbox configuration, without a matched controller or integrated brake. The supplier-provided motor and gearbox specifications are summarized in Table 8.

Table 8: Motor and Gearbox specifications

Spec	Motor Value	Gearbox Rating	Post Gearbox Value	Overall Value
Power	450 W	-	-	
Voltage	48V	-	-	
Rated Current	11.72 A	-	-	
Rated Torque	1.43 Nm	9.6 Nm	10.01 Nm	22.88 Nm
Rated Speed	3000rpm	429 rpm	428.5 rpm	187.5 rpm
No-load Speed	3300rpm	-	471rpm	206rpm
No-load Current	5.61A	-	-	
Peak Torque	4.30 Nm	28.9 Nm	30.1 Nm	68.6 Nm
Weight	3.5 kg	-	-	

Using an approximately 16:1 overall reduction, the rated motor torque of 1.43 N·m corresponds to 22.88 N·m at the gearbox output. The peak motor torque of 4.30 N·m corresponds to 68.8 N·m at the wheel output. Because the calculated peak output torque is greater than the listed 60 N·m gearbox rating, the motor controller settings must limit peak torque so that the gearbox rating is not exceeded during operation. Even with this gearbox limit, the drivetrain still provides more torque than the calculated requirement for forward acceleration. It is not expected that imposing this torque limit will have a major impact on player experience.

12.2.2 Torque Requirement Comparison — Spec F

From section 7.1 we have a relation between acceleration and torque:

$$\tau_w = \frac{mr}{N_d} (a + C_{rr}g) + \frac{1}{r} (I_{eq,w} a) \quad (5)$$

Using this equation with the assumed system mass and rider mass, the torque required at each driven wheel to meet the forward acceleration requirement in Spec F is approximately:

$$\tau_{required} = 49 Nm$$

The selected motor and gearbox combination can provide up to approximately 68.8 N·m based on motor peak torque and the rounded 16:1 reduction. Since 60 N·m remains greater than the 49 N·m required to satisfy Spec F, the drivetrain has sufficient peak torque capacity for the forward acceleration requirement.

12.2.3 Maximum Speed Analysis — Spec G

Spec G is a governed-speed requirement: the drivetrain must be capable of reaching at least 11.0 mph, while the controller must limit commanded maximum speed to no greater than 13.5 mph. At this speed of 11 mph, the required wheel speed is approximately 155 rpm. With the rounded 16:1 gear ratio used in this analysis, this corresponds to a motor speed of approximately:

$$155 \text{ rpm} \cdot 16 = 2480 \text{ rpm}$$

This motor speed is below both the rated motor speed of 3000 rpm and the no-load speed of 3300 rpm. Therefore, the selected motor and drivetrain ratio allow the wheelchair to reach the required maximum speed without requiring the motor to operate beyond its expected speed range.

Figure 19 shows the expected motor performance curve for the selected 450 W motor and gearbox configuration. The figure compares the required peak-acceleration torque, continuous torque capability, and maximum-speed operating points. The maximum-speed operating point occurs below the rated motor speed, while the required peak-acceleration torque remains below the usable peak torque limit. This supports the conclusion that the selected motor can meet both the acceleration and top-speed requirements.

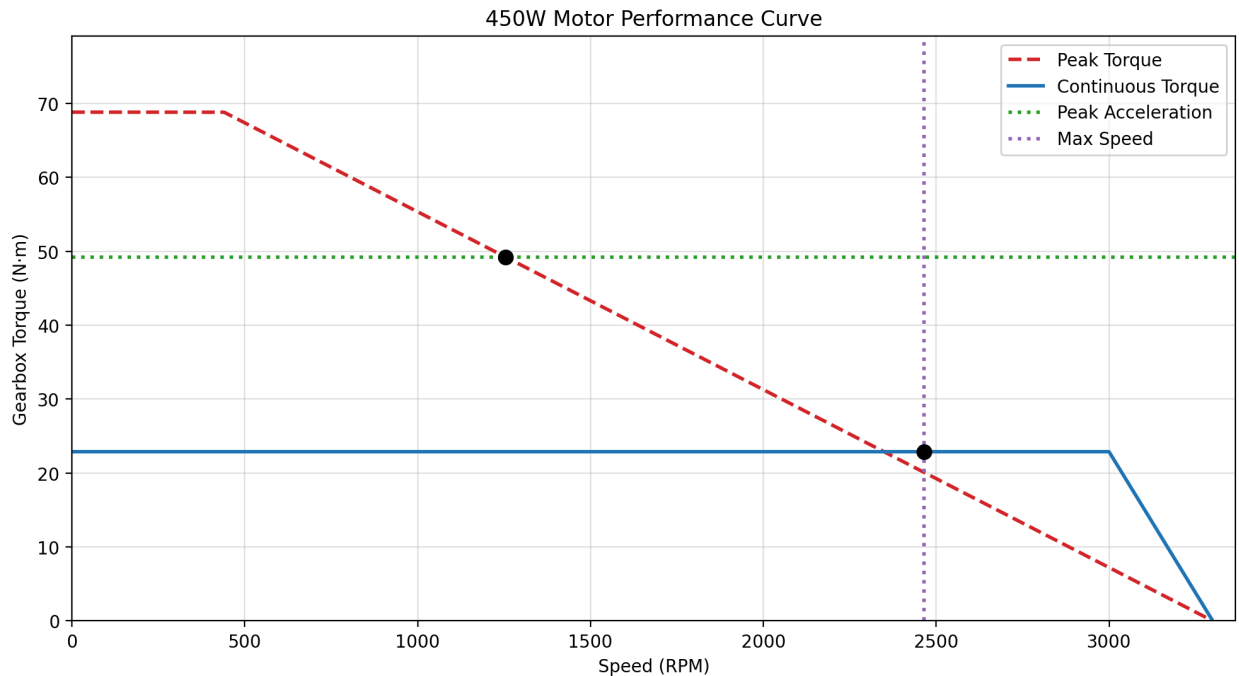


Figure 19: 450W Motor Performance Curve

The motor must also be able to maintain speed once the chair is already moving. According to the problem-analysis equation, the continuous torque of 22.88 N·m corresponds to approximately 1.28 m/s² of forward acceleration for the wheelchair. Converting this to force:

$$F = ma = 105(1.28) = 134 \text{ N} \approx 30 \text{ lbf}$$

This means the drivetrain can continuously provide approximately 30 lbf of tractive force. Because maintaining speed on a flat indoor court requires much less force than accelerating from rest, this continuous force capacity is far more than sufficient to maintain the required maximum speed under normal operating conditions.

12.2.4 Turning Performance Analysis — Spec H

Spec H requires the wheelchair to achieve an angular velocity of at least 200 deg/s within 0.50 s from neutral motion. This corresponds to an average angular acceleration of approximately:

$$\alpha = \frac{200^\circ/s}{0.50s} = 400^\circ/s^2 = 6.98 \text{ rad/s}^2$$

Appendix B derives an equation for the wheel torque required to produce this angular acceleration:

$$\tau_w = \frac{r(I_{tot}\alpha_{tot} + 2l_1 C_{rr} mg)}{2l_1} + \frac{I_w l_1 \alpha_{tot}}{r} \quad (\text{B.3})$$

Running numbers through this will all the standard values used in the problem analysis as shown in Appendix B we find a required pre-wheel torque of

$$\tau_{required} = 13 \text{ Nm}$$

Given that the peak torque is 68Nm, this spec is very achievable.

12.2.5 Braking Performance Analysis — Spec P

Spec P requires the wheelchair to come to a controlled stop from maximum operating speed in 3.0 s or less. The design uses the drive motors to apply braking torque opposite the wheel rotation. This is achieved by reversing the commanded current of the motor controller instead of using a mechanical brake.

If we derive the maximum stopping time of the wheelchair using our physics model from section 7.1, we can map our peak torque to a maximum deceleration of

$$a_{max} = 4.0 \text{ m/s}^2.$$

At a maximum speed of 11 mph, or approximately 4.92 m/s, the estimated stopping time is then

$$t = \frac{v}{a} = \frac{4.92 \text{ m/s}}{4.0 \text{ m/s}^2} = 1.23 \text{ s}.$$

We can also reverse this calculation and find the torque required to meet spec: For the 11mph top speed and the required 3 second stopping time, we would need a deceleration of

$$a = \frac{v}{t} = \frac{4.92 \text{ m/s}}{3 \text{ s}} = 1.64 \text{ m/s}^2.$$

Using the physics model, this maps to a per-wheel torque of:

$$\tau_{req} = 29.0 \text{ Nm}.$$

Thus, the estimated max achievable stopping time is 1.23s, less than half the 3.0 s requirement in Spec P, and to hit a 3.0 s stopping time, we estimate a required torque of 29Nm, which is only slightly higher than the rated value of 22Nm and well below the peak of 68Nm. While the physics model is generally accurate, it does assume rolling without slipping. Actual stopping performance will also depend on tire traction, motor-controller limits, current limits, and stability during deceleration. These factors should be verified during physical testing, but the analysis indicates that the selected motor and drivetrain have more than sufficient theoretical braking capacity.

12.2.6 Motor and Drivetrain Summary

The selected 450 W motor and drivetrain configuration provides enough torque and speed capacity to satisfy the primary motion-performance specifications. The required forward-acceleration torque is approximately 49 N·m per wheel, which is below the gearbox-limited peak torque capacity of 60 N·m. The required maximum-speed motor speed is approximately 2480 rpm, which is below the rated motor speed of 3000 rpm. The turning torque requirement is approximately 13 N·m per wheel, which is below the continuous output torque of the drivetrain. Finally, the estimated motor-braking stop time is approximately 1.23 s, which is below the 3.0 s requirement in Spec P.

These results indicate that the selected motor and drivetrain are capable of meeting Specs F, G, H, and P, provided that the electrical system can supply the required current and that the motor-controller settings are configured to protect the gearbox and battery limits.

12.3 Battery and Power Analysis

12.3.1 Battery Analysis Objective

The purpose of this analysis is to verify that the selected battery can support the wheelchair's expected average power demand, short-duration peak current demand, and two-hour runtime requirement. This section also checks whether the battery can remain fully onboard without violating the packaging constraints of the wheelchair.

12.3.2 Average Power and Continuous Current Requirement

Each brushless DC motor has the following operating characteristics:

- Rated Power: 450 W
- Voltage: 48 V
- Overall drivetrain ratio: 15.55:1

From Spec G and the motor analysis in Section 12.2, the required wheel speed at the governed maximum forward speed is approximately 155 rpm. This corresponds to a motor speed of approximately 2410 rpm. The motor angular velocity is therefore:

$$\begin{aligned}\omega_{motor} &= 2410 \frac{rev}{min} \left(\frac{2\pi rad}{1 rev} \right) \left(\frac{1 min}{60 s} \right) \\ \omega_{motor} &= 252.4 rad/s\end{aligned}$$

Phone IMUs were placed on wheelchairs during wheelchair basketball play to estimate representative acceleration behavior during athletic use. Taking the recorded acceleration data, we applied some basic data processing techniques to remove giant peaks from crashes and computed a representative average continuous acceleration. Then using the physics model from section 7.1, the representative average torque was estimated:

$$\tau_{motor,avg} = 0.83 \text{ Nm}$$

Using this value, the average mechanical power for one motor is:

$$P_{mech,avg,1} = \omega_{motor} * \tau_{motor,avg} = 252.4 * 0.83 = 209.49 \text{ W}$$

In order to convert this mechanical power to an electrical one we must divide by the efficiency of the system which converts electrical energy to mechanical. This is a critical step in a critical calculation, yet we have no way to measure this efficiency. Furthermore we are already using a representative average for acceleration, which means that this efficiency value is also an average. But DC motors have a very much non linear efficiency curve. Under high acceleration and deceleration conditions we will operate in the lower rpm higher torque regions which typically have much lower efficiencies, depending on which motor performance curve you look at. Thus it's possible our assumptions could be overly optimistic. The system for which we need to measure efficiency consists of our 450W BLDC motor, its gearbox, the timing pulleys, and the dog coupling mechanism and its bearings. Assuming a combined motor and drivetrain efficiency of 75% – which was chosen as normal motor efficiency has 85% efficiency, but there is an assumption that there will be an additional 10% loss in efficiency due to the gear shifting and pulley drag – the average electrical power required by one motor is:

$$P_{elec,avg,1} = \frac{P_{mech,avg,1}}{\eta} = \frac{209.49}{0.75} = 279.32 \text{ W}$$

At the nominal 48 V battery voltage, the average current required by one motor is:

$$I_{avg,1} = \frac{P_{elec,avg,1}}{V} = \frac{279.32}{48} = 5.82 \text{ A}$$

This means the battery must continuously supply approximately 559 W, or 11.64 A at 48 V, during representative athletic operation.

To check the short-duration peak current demand, the torque required to satisfy the forward acceleration requirement from Spec F is approximately 49 N·m at each driven wheel. This gives:

$$\tau_{motor,peak} = \frac{\tau_{wheel,peak}}{Gear\ ratio} = \frac{49}{15.55} = 3.15 \text{ Nm}$$

The corresponding peak mechanical power for one motor at the maximum-speed operating point is:

$$P_{mech,peak,1} = \tau_{motor,peak} \omega_{motor} = 3.15 * 252.4 = 795.06 \text{ W}$$

For two motors, accounting for 75% efficiency:

$$P_{elec,peak,total} = \frac{2P_{mech,peak,1}}{\eta} = \frac{2*795.06}{0.75} = 2120.16 \text{ W}$$

The corresponding peak battery current is:

$$I_{peak} = \frac{P_{elec,peak,total}}{V} = \frac{2120.16}{48} = 44.17 A$$

Therefore, the battery must support approximately 12 A continuously during representative athletic use and approximately 44.17 A during short-duration peak loading.

12.3.3 Runtime and Capacity Requirement — Spec O

Spec O requires the wheelchair to support at least 2.0 hours of normal athletic use using onboard battery power. From Section 12.3.2, the estimated average current draw during representative athletic operation is approximately 12 A at 48 V. To avoid fully depleting the battery, this analysis assumes an 80% depth of discharge.

12.3.3.1 Required Capacity — Spec O

Spec O requires the wheelchair to operate for at least 2 hours on a full battery charge.

The required battery capacity is estimated from the average current draw, required operating time, and usable depth of discharge of 80%:

$$Ah_{required} = \frac{t * I_{total}}{DoD} = \frac{2 * 12}{0.8} = 30 Ah$$

Thus, to last for 2 hours, the wheelchair will need a 30Ah battery while running on 48 V.

12.3.3.2 Energy Requirement — Spec O

A 48 V, 30 Ah battery, corresponding to the calculated capacity requirement, stores:

$$E = V * Ah = (48 * 30) = 1440 Wh.$$

Table 9: Battery Spec Requirements

Parameter	Value
Voltage	48 V
Capacity	30 Ah
Total Energy	1440 Wh
Usable Energy (80%)	1152 Wh
Required Runtime	2.0 h

At the estimated average electrical load of 559 W, this corresponds to approximately 2.0 hours of operation. This satisfies Spec O under the stated assumptions, but the margin is limited. Because the estimate depends on representative gameplay loading, the final runtime should be verified through physical testing with realistic start-stop athletic driving.

12.3.3.3 Regenerative Braking

Regenerative braking was considered but was not selected as a required feature for the current prototype. Although it could recover some energy that would otherwise be lost during braking, the benefit is limited by the battery's allowable charge current and the relatively small amount of kinetic energy available during each stop. In this design, regenerative braking would be limited by the battery charge current of 4 A. The braking force would be:

$$F(v) = \frac{P}{v} = \frac{58.2 \cdot 4}{4.92} = 47.3 \text{ N},$$

where P is the power produced by the braking, and v is the velocity of the wheelchair. To stop the wheelchair in 3 seconds, the force of 150 N would need to be applied. This means that only 31.5% of stopping power could be generated from regenerative braking under the best conditions. The kinetic energy of the wheelchair at maximum speed can be calculated as:

$$E = \frac{1}{2}mv^2 = \frac{1}{2} \cdot 90.72[\text{kg}] \cdot 4.92^2[\text{m/s}]^2 = 1098 \text{ J}.$$

The ideal charge this energy could provide in mAh is

$$Q_{Ah} = \frac{1098}{58.2 \cdot 3600} = 5.2 \text{ mAh},$$

from maximum velocity to a complete stop.

Given that only 31.5% of the stopping power could come from regenerative braking under ideal conditions, and assuming 80% recovery efficiency, only about 1.3 mAh could be recovered from a full-speed stop. This is approximately 0.0044% of the 30 Ah battery capacity, meaning it would take roughly 23,000 full-speed stops to recover the battery's full nominal capacity. This recovery is too small to justify the added control complexity for the current prototype, especially because the selected battery already meets the required runtime and current demands.

12.3.4 Selected Battery Capability

The selected battery is a Bioenno Power lithium iron phosphate battery with its specifications shown in Table 10.

Table 10: Actual Battery Specs

Parameter	Value
Voltage	48 V
Capacity	30 Ah

Total Energy	1440 Wh
Charger	58.2 V/4 A
Constant Discharge Current	30 A
Peak Discharge	60 A (30 s)
Weight	31.1 lbs
Dimensions	9.75 in. × 7.5 in. × 6.5 in.

This battery was selected because it matches the required 48 V system voltage, provides the calculated 30 Ah capacity requirement, and has discharge ratings above the estimated average and peak current demands.

12.3.5 Continuous and Peak Current Verification

From Section 12.3.2, the estimated average current demand is approximately 12 A. The selected battery is rated for 30 A continuous discharge, so the expected normal operating load is within the battery's continuous current capability.

The estimated short-duration peak current demand is approximately 44.2 A. The selected battery is rated for 60 A peak discharge for 30 s, so the battery can support the calculated peak load as long as the motor controllers limit peak-current events to short durations.

This satisfies the current-delivery requirement for normal athletic operation and short bursts of high acceleration.

12.3.6 Worst-Case Electrical Loading and Safety Limits

12.3.6.1 Maximum Load Condition

In the worst case scenario, we have:

$$P_{max} = 2121.97 \text{ W}$$

$$I_{max} = 44.21 \text{ A}$$

This occurs when we have maximum acceleration (Spec F), high speed operation (Spec G), and aggressive turning (Spec H).

12.3.6.2 Thermal and Electrical Limits — Spec B

Spec B requires the system to remain safe under fault conditions. Battery-related failure risks include overcurrent, overheating, and excessive discharge. The calculated loads are approximately 12 A during average gameplay and 44.21 A during peak operation. The average load is below the battery's 30 A continuous discharge limit, and the peak load is below the 60 A peak discharge limit as long as peak current draw does not exceed 30 s. Current limits must therefore be implemented in the motor controllers to keep the battery within its safe operating range.

12.3.7 Battery Integration and Packaging — Specs N and U

Spec U requires the propulsion batteries, motor controllers, and required electronics to be mounted onboard the wheelchair during normal operation. Spec N requires the wheelchair to remain no wider than 31.5 inches in its transport configuration. The selected battery is 9.75 in × 7.5 in × 6.5 in and fits underneath the chair without increasing the overall wheelchair width. Once mounted, the battery satisfies Spec U by keeping the power source onboard and supports Spec N by staying within the chair's width constraint.

12.3.8 Assumptions and Remaining Battery Risks

Assumptions used:

- Motor efficiency = 75% (to account for the motors loss of efficiency and gear shifting efficiency loss)
- Average torque = 1 Nm (0.2 Nm of torque higher than found in game to account for losses and longer peak current draw)
- Constant 48 V supply
- Depth of discharge = 80% (typical lithium ion battery)
- Negligible wiring losses

These assumptions increase the required battery size and provide some margin for losses, but runtime should still be verified through physical testing under representative athletic use.

12.3.9 Battery and Power Analysis Summary

The battery system satisfies the relevant specifications:

- **Spec O (Runtime):** 2.0 h achieved with the 30 Ah battery under the assumed average-load condition.
- **Spec U (Onboard Power):** fully self-contained system power provided.
- **Spec B (Safety):** Operates within safe electrical limits of chosen battery.
- **Spec N (Packaging):** Fits within dimensional constraints.
- **Spec F, G, H (Performance):** supports peak power demands.

12.4 Motor Mount Structural Analysis

These calculations and analyses primarily evaluate whether the motor mount structure supports Specification A, which requires the wheelchair to withstand representative sport-contact loading without fractures, critical component detachment, or permanent deformation that causes failure of a safety, stability, or control specification. The full details of the FEA setup and results are included in Appendix A.

12.4.1 Clamp Loading Analysis

The clamps which attach to the main tube are critical in keeping the axle and motor in place because they form the interface between our design and the existing wheelchair.

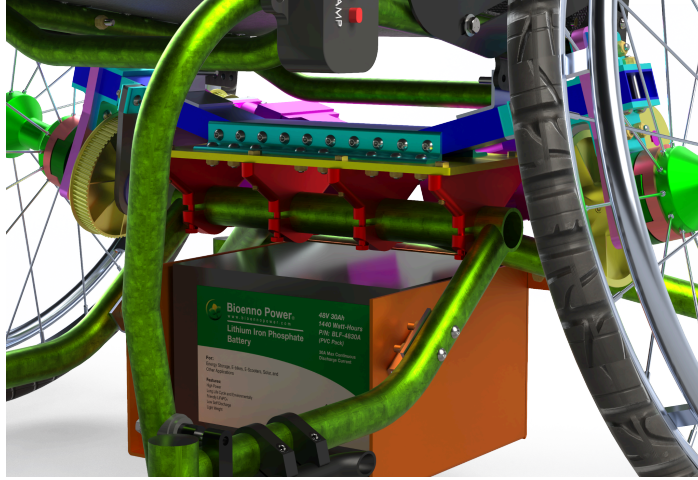


Figure 20: CAD Rendering of the Wheelchair Focused on Clamps

The clamp is very strong in both compression and tension. Under a 500 lbf load, the compressive and tensile FEA studies produced safety factors of 21 and 9.28, respectively. A far more serious concern is that of the clamp slipping around the main tube. To determine the effect of this we can calculate the moment that would be required to rotate a clamp about the tube. Consider the diagram shown in figure 21. In it we have the upper part of a clamp on the main tube. A moment is applied representative of loads from the wheel or crashes, or whatever load could attempt to twist the clamp. The clamping force from the bolts is shown at the bottom. Writing out the statics gives:

$$\sum M = M_{app} - F_{fr}r$$

$$M_{app} = F_{fr}r$$

We know $F_{fr} = \mu_s N$ However, because the contact surface is curved, N is a function of its position. Thus we must integrate. The clamp forces are omitted from the statistics because they cancel out. To consider the entire contact surface we must integrate from α to $180 - \alpha$.

$$N = \int_{\alpha}^{180 - \alpha} \frac{F_{clamp}}{t\theta} \sin(\theta) d\theta$$

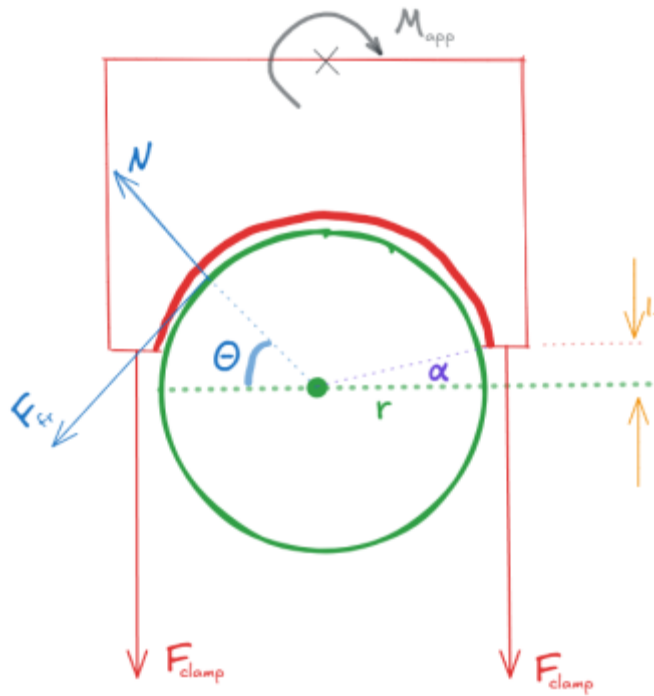


Figure 21: Freebody diagram of Forces on a Clamp

Combining our equations we find that

$$M_{app} = 2 * r \mu_s \int_{\alpha}^{180 - \alpha} \frac{F_{clamp}}{t \theta} \sin(\theta) d\theta \quad (5)$$

Where the thickness of the clamp $t = 0.5\text{in}$, and the tube radius $r = 0.75\text{in}$. Because the two sides of the clamp have the same contact geometry, we can simply multiply by two to find the total moment resisted by a clamp for a given clamping force.

There are still two unknowns, α , a geometric constant, and μ_s , the friction coefficient. For this analysis we will use the coefficient of friction for steel on aluminum: $\mu_s = 0.61$ [35] Alpha can be determined from geometry as

$$\alpha = \tan^{-1}\left(\frac{l_1}{r}\right) = 9.462^\circ$$

These clamps will be held together with two SAE grade 8 ¼-20 bolts, which having a maximum tightening torque of 12 ft-lbs. [36]

From Shigley's Mechanical Engineering Design [37] we have

$$T = K F_i d$$

Eq. 8-27

Where T is bolt torque, K is the torque coefficient, F_i is the bolt preload or clamping force, and d is the bolt diameter. The K factor encapsulates all the physics of bolts, we can use the value from Table 8-15 in Shigley's for Zinc-plated bolts of $K = 0.20$. Thus:

$$\frac{T}{Kd} = F_{bolt} = 2880\text{lbf per bolt.}$$

$$F_{clamp} = 2 F_{bolt} = 5760$$

Using equation 5:

$$M_{app} = 1477 \text{ ft-lbs.}$$

As modeled at the time of writing, pressing on the motor has a 5in moment arm, so this result implies that one can press the motor down with 3574lbf without the clamps twisting. Clearly that will never happen and other parts will certainly fail before the clamps slip. This is an idealized case because the bolts may not be tightened exactly to 12 ft-lbs, the clamp may not seat perfectly on the tube, and the friction coefficient is difficult to verify. Even with these uncertainties, the predicted slip resistance is high enough to suggest that clamp rotation is unlikely to be a primary failure mode.

As an additional precaution, the design includes a support beam that provides a secondary load path and reduces the moment carried by the clamps.

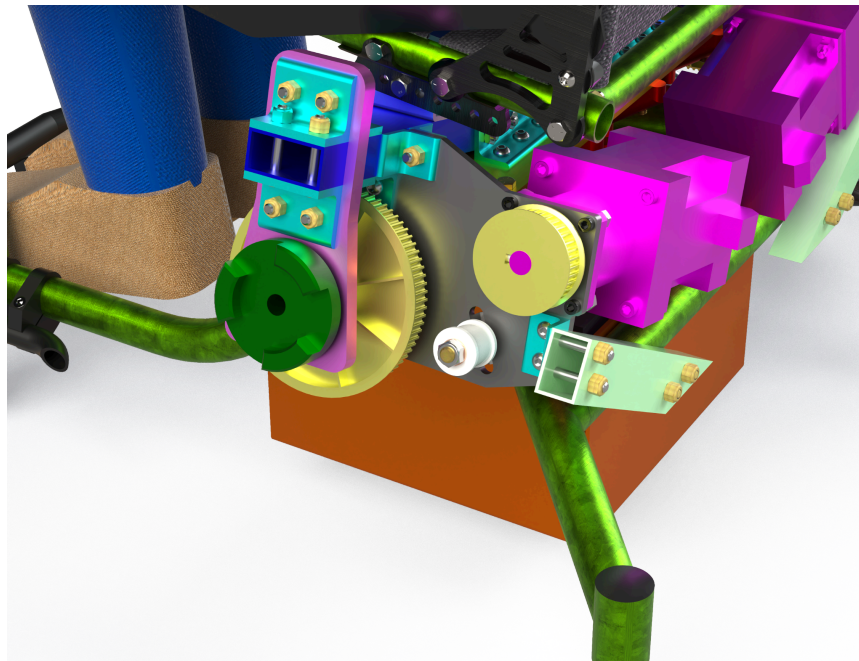


Figure 22: CAD Rendering of the Motor Mount and Support Structure

In conclusion, the motor mount clamps are unlikely to be the limiting structural component in the current design. Based on the FEA results and clamp-slip estimate, the clamps are expected to support Specification A, although final validation should still be completed after prototype assembly.

12.4.2 Motor Mount Plate Bending Analysis

This analysis evaluates whether the motor mount plate can withstand a large force applied to the motor. It is difficult to analyze this because we have not yet finalized how the motor mount plate is held to the wheelchair. For this simulation some sample fixtures were created to represent a gusset and/or angle bracket style mounting solution. The holes which correspond to the rear support were also fixtured. The load applied to the motor was 30lbs, in addition to gravity. This number was picked as a reasonable estimate for loads that could be experienced. Importantly, this was simulated as an assembly meaning that connectors had to be defined between the motor and gearbox and between the gearbox and mount plate. The connectors were M6 and M5 bolts with the preload set to the 10.25 and 6.03 Nm respectively. These values were used as representative tightening torques for bolts of those sizes.

The results show localized stress concentration around the motor and gearbox mounting bolts, as shown in Figures 23 and 24. Under the 30 lbf applied load, the model predicts a small region near yield in the simulated motor/gearbox mounting area. Because the motor was simplified as a solid block of steel and the final mounting geometry is not fully finalized, this result should be treated as a preliminary check rather than final validation. The stress concentrations here can be largely resolved by adding an additional 3D printed support connecting the rear of each motor to the frame. However this part cannot be modeled until we have the physical motors and can get accurate measurements. See section 13.1.3.

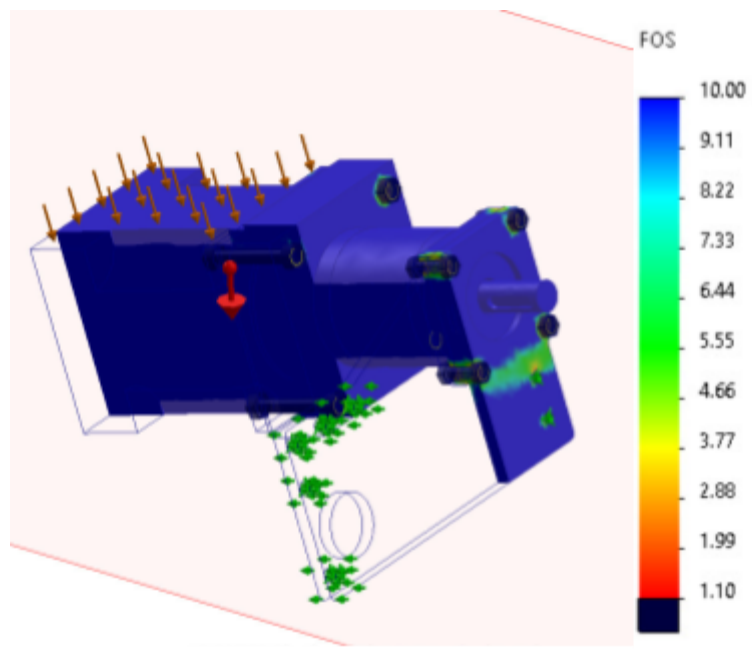


Figure 23: Factor of Safety for motor mount plate bending study

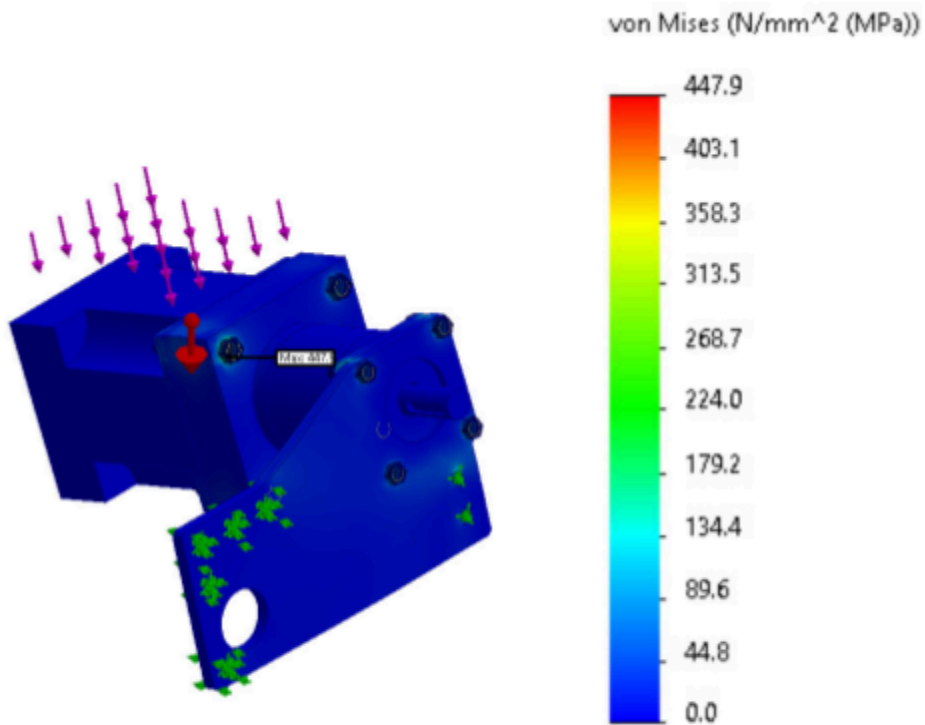


Figure 24: Stress in motor mount plate bending study

12.5 Drivetrain Analysis

This section analyzes the drivetrain and determines whether it can sufficiently handle the loads it is expected to receive during both normal operation as well as from impacts. All values and equations were pulled from Shigley's Mechanical Engineering Design 2024 [38], unless otherwise stated.

12.5.1 Wheel Hub Press Fit



Figure 25: Shows the steel shaft (red) and the wheel hub (green)

A 0.5 inch steel shaft will be press fit into the current aluminum wheel hub. To analyze the capacity of this press fit we needed to calculate the factor of safety for the shaft, the wheel hub, and the maximum torque allowed before slipping occurs. Below is a series of equations used to determine our factors of safety.

$$N_{PressFit} = \frac{S_y}{\sigma} \quad (2.1)$$

$$(\sigma_t)_{inner} = -p \frac{R^2 + r_i^2}{R^2 - r_i^2} \quad (2.2)$$

$$(\sigma_t)_{outer} = p \frac{r_o^2 + R^2}{r_o^2 - R^2} \quad (2.3)$$

$$p = \frac{\delta}{R \left[\frac{1}{E_o} \left(\frac{r_o^2 + R^2}{r_o^2 - R^2} + \nu_o \right) + \frac{1}{E_i} \left(\frac{R^2 - r_i^2}{R^2 - r_i^2} - \nu_i \right) \right]} \quad (2.4)$$

$$\delta = R - H_i \quad (2.5)$$

$$\tau = FR \quad (2.6)$$

$$F = \mu p A \quad (2.7)$$

$$A = 2\pi RL \quad (2.8)$$

Equation 2.1 was used to calculate the factor of safety for each component in the press fit. To calculate the shaft's factor of safety, 57 ksi was used as the yield strength (S_y) as it will be made of AISI 1020 mild steel. The wheel hub is believed to be AISI 6061-T6 aluminum which has a yield strength (S_y) of 39.9 ksi. Equations 2.2 and 2.3 were used to calculate the stresses put on the shaft and the hub, respectively. These equations used the shaft radius (R), inside radius of the shaft (r_i), shaft outer radius (r_o), and pressure (p). The pressure was calculated using equation 2.4. Interference (δ) was calculated to be .0005 inches, using equation 2.5. Young's modulus for the shaft (E_i) was 29 Msi, and 10 Msi for the hub (E_o). Poisson's ratio for the shaft (V_i) is 0.29, and 0.33 for the hub (V_o). Using these values, the factors of safety for the steel shaft and the aluminum hub are 28.0 and 4.9, respectively.

Equation 2.6 was used to calculate the maximum allowable torque before slipping occurs. R is the radius of the shaft, and F is the force in the shaft. Force was calculated using equation 2.7 using the same pressure previously found, the contact surface area found using equation 2.8, and the coefficient of friction (μ). The coefficient of friction was assumed to be 0.35 which is a conservative value as it likely is between 0.45 and 0.61 [39]. The maximum allowable torque through the shaft before slipping occurs is 200 lb*ft. Given that our peak torque is expected to be 51 lb*ft, this gives us a factor of safety of 3.9.

12.5.2 Main Drive Shaft

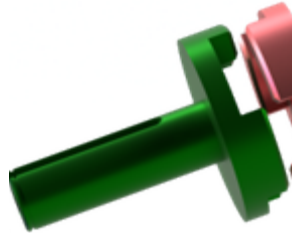


Figure 26: Shows the inner dog couple. The main drive shaft is the green shaft connected to the coupler.

To ensure that the drive shaft can withstand the torque necessary for our application, the torsion stress was analyzed under peak load.

$$\tau = K_t \frac{Tr_{shaft}}{J} \quad (2.9)$$

$$J = \frac{\pi r^4}{2} \quad (2.10)$$

The estimated shear stress from torsion was calculated using equation 2.9. The torque through the shaft (T) is 51 lb*ft, r_{shaft} is substituted with $r_o^4 - r_i^4$, J is the polar moment of inertia, and K_t is the correction factor. The substitution of r_{shaft} is due to the shaft being hollow. The polar moment of inertia was calculated using equation 2.10. A correction factor of 2 was used to account for the keyseat in the shaft. The expected torsion stress that the shaft would experience under peak load would be 6.68 ksi, which gives us a factor of safety of 4.4.

12.5.3 Dog Coupling

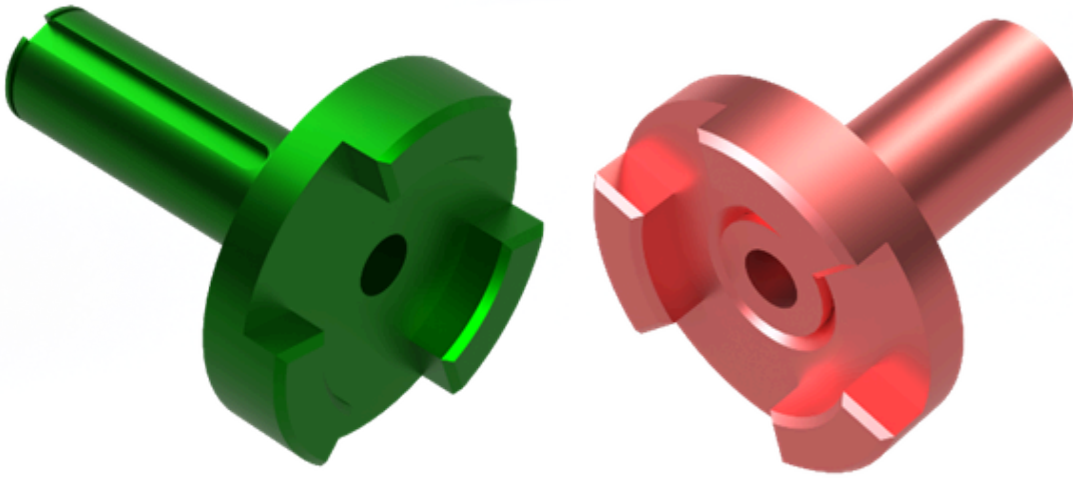


Figure 27: Shows the two dog couplers.

To ensure that the dog couplers can handle the expected forces, analysis was completed for the stresses on the faces of the teeth, the shear forces on the teeth, and the torsion experienced by the shaft.

$$\sigma = \frac{F_{dog}}{A_{dog}} \quad (2.11)$$

$$F_{dog} = \frac{F_t}{n} \quad (2.12)$$

$$F_t = \frac{T}{r} \quad (2.13)$$

$$A_{dog} = w \times h \quad (2.14)$$

$$\tau_{bearing} = \frac{F_{dog}}{A_{dogFace}} \quad (2.15)$$

$$A_{dogFace} = w \times L \quad (2.16)$$

$$\tau_{torsion} = \frac{Tr_{shaft}}{J} \quad (2.17)$$

The stress on the faces of the teeth was calculated using equation 2.11. The force on each tooth was calculated using equation 2.12, where the number of teeth (n) is 3. The forces on all teeth were calculated using equation 2.13, where T is the torque through the shaft (51 lb*ft) and r is the radius of the gear (1.5 inches). The bearing stress was calculated using equation 2.15 and the torsion was calculated using equation 2.17. The polar moment of inertia was calculated using equation 2.10. Using these equations, we found that the stresses on the faces are 69.8 psi on each tooth, the bearing forces are 2.6 ksi, and the shear stress from torsion is 119 psi. The calculated factor of safety for the dog coupling is 10.6.

12.5.4 Keyed Shaft Connection

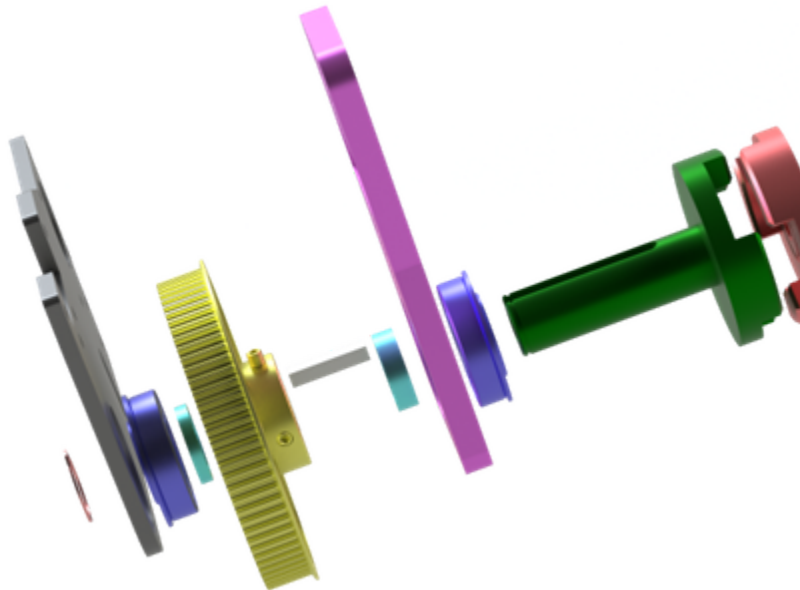


Figure 28. Shows an exploded view of the drivetrain. The silver component is the key, the green component is a shaft with a keyseat, and the yellow component is the pulley with the keyway.

The pulley keyway, shaft keyseat, and key were all analyzed to ensure they would effectively transfer the torque through the drivetrain without breaking.

$$\tau_{key} = \frac{V}{A_{key}} \quad (2.18)$$

$$V = \frac{T}{r} = \frac{2T}{D_{shaft}} \quad (2.19)$$

$$A_{key} = H_{key} \times W_{key} \quad (2.20)$$

$$\tau_{shaft} = \frac{Tr_{shaft}}{J} \quad (2.21)$$

$$\sigma_{bearing} = \frac{F}{A} \quad (2.22)$$

$$F = \frac{2T}{D} \quad (2.23)$$

$$A = L_{key} * \frac{H_{key}}{2} \quad (2.24)$$

Equation 2.18 was used to calculate the shear force on the key, keyseat, and keyway. Equation 2.19 was used to estimate the force that each component experienced. Equation 2.21 was used to calculate the torsion in the shaft. The bearing force was calculated using equation 2.22, and equation 2.23 was used to determine the forces. The stress on the key was found to be 3.9 ksi, the stress on the keyway is 7.2 ksi, and the stress on the keyseat is 2.2 ksi. The shear stress from torsion and bearing forces are 3.32 ksi and 14.4 ksi, respectively. This gives the key a factor of safety of 2.7, the keyway a factor of safety of 5.5, and the keyseat a factor of safety of 13.8.

12.5.5 Drive Shaft Bearings

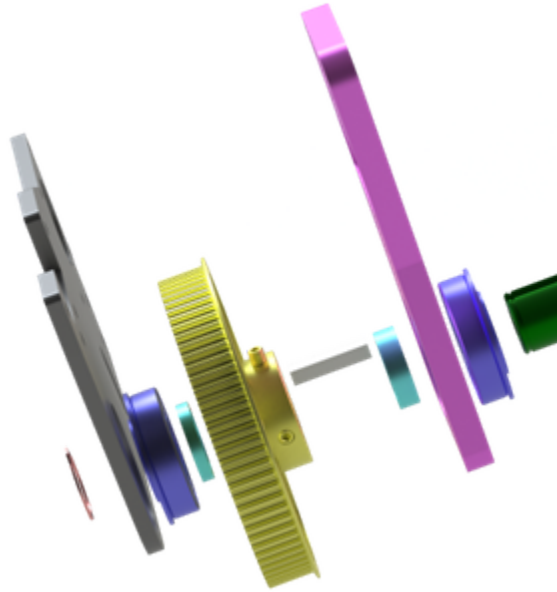


Figure 29: Shows part of the exploded view for the drivetrain. The gray component is the motor mount plate, the purple one is the bearing plate, and the two dark blue components are the bearings.

Similar to the analysis of the wheel hub and shaft press fit, a factor of safety was calculated for the stress experienced by the motor mount plate and the bearing plate, depicted above.

$$p = \frac{E\delta}{2R^3} \left[\frac{(r_o^2 - R^2)(R^2 - r_i^2)}{r_o^2 - r_i^2} \right] \quad (2.25)$$

Both the bearings and the plates are made of steel, and both yield strengths are assumed to be 57 ksi. Equations 2.2 and 2.3 were used to calculate the stresses on the bearings and the plate, respectively. Equation 2.25 was used to calculate pressure this time since both components are made of the same material. The Young's modulus (E) is 29 Msi, δ is .0005 inches, the outer radius of both bearings (R) is 1 inch, the inner radius of both bearings is 0.5 inches, and the outer radii of the plates were assumed to be 1.38 inches and 1.54 inches for the motor plate and the bearing plate, respectively. For the bearing attached to the motor mount plate, a factor of safety of 11.5 was calculated and the plate's factor of safety is 5.9. For the bearing attached to the bearing plate, the factor of safety is 9.7 and the 5.0 for the bearing plate.

13. Feedback Items

During the presentation of this design and its subsequent review we have received various pieces of valuable feedback which will be strongly considered and implemented into the design in the future. The people who gave feedback are Dr. Nardos Ashefani, a robotics and controls specialist at Bastian Solutions in Boise, Dr. O'Hern and Brian Higgins, our instructors, and Dr. Aaron Smith, a mechanical engineer with his PE stamp.

13.1 Mechanical System Feedback

13.1.1 Dynamic component Alignment

One large concern involves aligning the dynamic components of the drivetrain, specifically the bearings. To ensure that the shaft runs smooth and true, it is imperative that the drivetrain components be precisely aligned with little room for tolerance. If the components are not aligned, the inner deg coupling plate will bind at best, and at worst, be impossible to assemble. The reason this is such a large concern is because of stacking tolerances in our design. You can see in figure 30 that both bearings have the blue boxtube as a shared reference, but the mount plates in grey and pink and angle brackets in teal are all somewhat complicated parts that have multiple levels of features that must all be precise in order to achieve a good fit. Another complicating factor is that

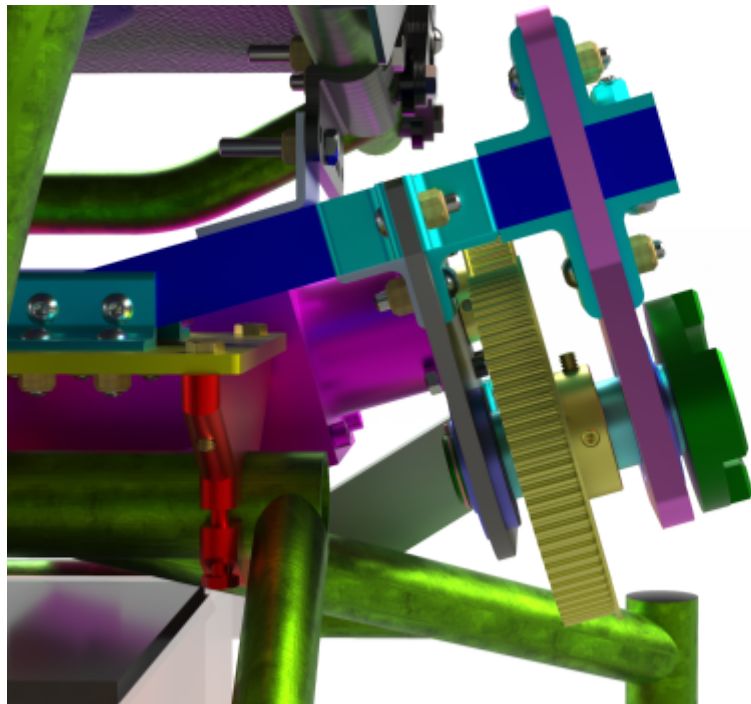


Figure 30: Side view of dynamic components

the angle brackets are currently slated to be welded onto the boxtube and bolted to the motor mount plates. This choice was made because of difficulties with getting threaded fasteners to fit the space constraints, but it comes at the cost of imprecision. The primary mitigation strategy for these issues is to be really careful when machining. Dr. Smith suggested consulting with the professional machinist Phil Boysen who works in the EIS at Boise State University for more advice on solving this issue. Another possibility, if budget allows, would be sending the parts to be professionally machined by a company such as Send Cut Send.

13.1.2 Steel vs. Aluminum Shafts and Fatigue

Steel shafts were recommended as the material of choice for the shafts in the drivetrain, as opposed to aluminum because aluminum does not have an infinite life fatigue limit. Our shafts are undergoing intense fully reversed cyclic stress. Driving across the length of a court once requires 15 revolutions of the wheel. If you roll across the court once every 10 seconds, which works out to around 5mph, and do that continuously for 2 hours, the wheel's shaft will undergo around 10k cycles. Especially over multiple games, fatigue is a serious issue and this makes aluminum shafts unacceptable. Thus we will use steel for all our dynamic rotating components.

13.1.3 Motor Cantilever Support

Currently the motors are only supported by the motor mount plate. This can be seen in figure 31, While that plate itself is well supported, the motors have a lot of mass and will most likely bend the plate due to the significant impacts the chair will continually undergo. To mitigate this, it is recommended that we design some sort of support for the back of the motors. This will certainly be implemented once we have the motors on hand and can iterate a 3D printed support to accomplish this. The support does not have to be extremely strong, it simply has to mitigate the bended loads caused by the motors.

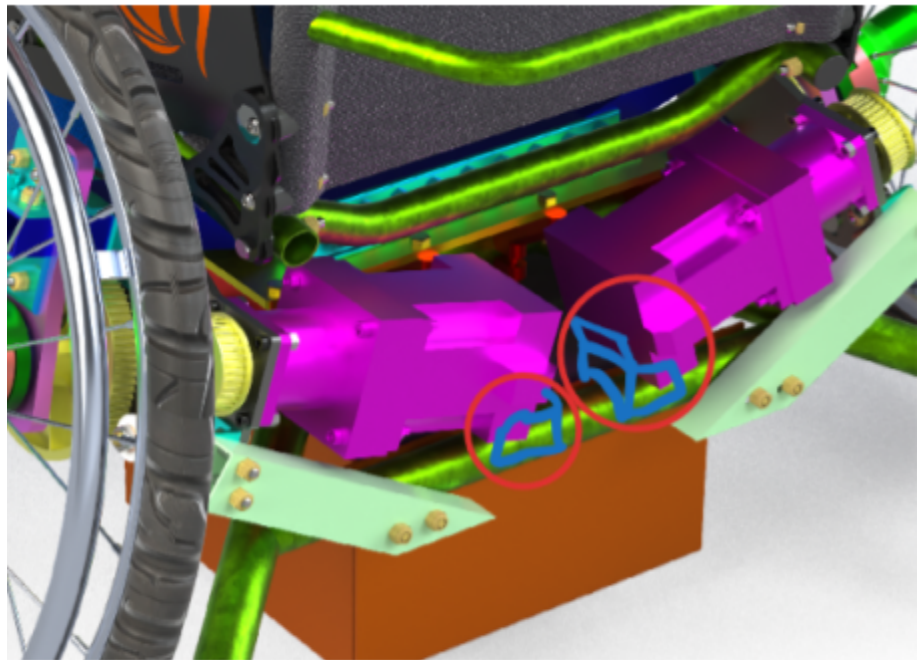


Figure 31: Sketch of possible cantilever support option

13.1.4 Battery Mount

The currently designed battery mount is a bent sheet metal box that is bolted to the frame. However this design is not very well flushed out and does not include a way to hold the battery in place. This issue must be solved and a door mechanism was briefly suggested. Another solution could involve velcro straps to hold the battery down. Whatever the solution is, the battery must be firmly retained inside the battery mount box. Additionally, because the battery is capable of delivering such large amounts of power, much care must be taken to ensure that the frame of the wheelchair is never electrified, causing a safety hazard. To that point we must ensure that all connections to and from the battery are insulated and that the battery connector is held securely in place.

13.1.5 Miscellaneous

Because this is a human operated device a safe testing plan must be developed. It is very likely that things will go wrong initially and it is best if they go wrong without a human in the chair.

Hannah said that disk brakes are probably not a viable solution because of the added complexity, cost, and electrical draw. We should conduct and document some brief analysis to confirm this.

The motors we are currently designing around are sourced from China which raises purchasing concerns. Due to the tight budget we need to be aware of potential tariffs when buying these.

The weight of the system and inertia of the drivetrain could make manual movement of the wheelchair significantly difficult. While this is inevitable, analysis could be done to attempt to quantify this impact. Depending on the result of that analysis it may be propitious to pursue greater weight reduction.

13.2 Electrical System Feedback

13.2.1 Sensor Placement and Interference

Placing the sensors in the correct location will make or break the design. Having a direct and intuitive connection to the chair is the entire purpose of this project. Our sponsor Dan clarified that having a hard plate directly under the cushion, will not negatively affect the playing experience but also brought up the concern that placing the sensors near the top of the cushion exposes them to the sweat and abrasion of the user. Beyond the physical placement of the sensors, we were also reminded of the Garbage In, Garbage out principle. Nardos mentioned that these sensors will likely be prone to noise and Mr. Higgins expounded on the effects of electrical interference on the sensor readings. Because of the high currents passing to the motors from the battery and through the motor controllers, there may be electromagnetic noise picked up by the sensors. Thus it's critical that we ensure we get good readings. A potential solution to this is using differential sensing, an alternative protocol that is similar to what CAN bus uses. Ensuring the wiring is shielded can also mitigate these effects.

Another issue with the sensor placement was the sheer number of wires there are. We are planning on starting with a 4x4 array of sensors, each of which has 2 wires meaning 32 soldered connections. That will get messy fast and it is worth looking into trying to mitigate that. Mr. Higgins suggested talking with Dr. Ben Jorgensen who specializes in sensor arrays and Dr. Carol Baumbauer whose research involves printing flexible circuits. Her research could directly apply to printing a flexible pressure sensor array. However using such specialized components does eliminate the ability to procure replacement parts in the future.

13.2.2 Wire Routing and Circuit Durability

Getting wires to different parts of the chair runs the risk of turning this thing into a rat's nest. Great care must be taken to ensure safe, strong, reliable, and maintainable wire routing. Wires could potentially be routed through the tubes of the frame which results in a very clean look and protects the wires. However in the event something goes wrong or needs to change, it's much more difficult to debug or maintain the electrical system. Another solution is using flexible conduit to capture wires into bundles and further insulate them from the frame.

Due to the large impact loads these circuits are subjected to, a simple breadboard or even prototype board would get shaken to bits, leading Mr. Higgins to strongly recommend that we design and use a printed circuit board. This will drastically reduce the electronics complexity and size and increase its impact resistance at additional monetary cost.

13.2.3 Emergency Stop

Another topic that received a lot of discussion was the state of the wheelchair after the emergency stop is triggered. As designed, the emergency stop cuts all power to everything. Naturally the teensy having no additional power supply or inductance will immediately shut down, leaving the rest of the system uncontrolled. This is potentially a safety hazard. The motors can continue unpredictable powered movement for short periods of time after the e-stop is triggered which could cause a safety hazard. For example if the chair moves at a high rate of speed and the e-stop is triggered and one wheel grinds to a halt but the other keeps going, the wheelchair could suddenly turn or flip over, which is clearly unacceptable. While that is a bit of an extreme case, it's not unrealistic for there to at least be some issues. However we cannot make any definitive judgement about this without testing it. One solution that was proposed was including a small capacitor bank to keep the teensy alive longer than the motors to control them until their momentum is dispersed. Overall we need to analyze this scenario and create a plan to work around it.

13.2.4 Current Vs. Velocity Control

One brief note about control, according to Nardos, using current control as currently designed is non-trivial and requires a significant amount of code while we could use velocity control instead. Using velocity control would simplify our code and allow the motors to passively perform the current control. This is because the motor controller already intelligently handles all the math between our desired velocity and the current that needs to get sent to the motors.

14. Bill of Materials:

Table 11: Bill of Materials for the lean controlled wheelchair. Includes mechanical and electrical components.

Item ID	Vendor	Part Number	Item Description	Link	Quantity	Cost/Unit (\$)	Subtotal	Business Purpose
1	Amazon	B0FM8QKSM1	Silvery Momentary Push Button Switch with Stainless Steel and Aluminum Alloys Witch, 2A 250V AC	Momentary Push Button Switch Amazon	1	\$5.99	\$5.99	Will act as the "calibration" button. Zeroing out the sensors for the baseline when the user is not leaning but sitting in the chair
2	Amazon	XT90-S	Connector Anti-Spark Male Female Connector for Battery, ESC and Charger Lead	Anti-Spark Connectors Amazon	1	\$12.98	\$12.98	Standard power connectors
3	Amazon	B07V1D82HM	24 AWG Solid Core Hookup Wire Kit, 6 Different Colored Jumper Wire 30 ft Each, Tinned Copper Wire with PVC Insulation	Signal Wire Amazon	1	\$15.59	\$15.59	Wire to run signals

4	Amazon	22KC-TZ7128	175 pcs Key Stock 8mm-30mm Stainless Steel	Shaft Key Kit Amazon	1	\$15.99	\$15.99	This is a shaft key kit that will allow us to connect the pulleys to the shafts
5	Amazon	EP-LCB-200A	200A 12-48 Volt DC Self Recovery Circuit Breaker with Manual Reset Button	Circuit Breaker Amazon	1	\$18.99	\$18.99	Will act as the E-stop, disconnecting positive line from the rest of the system
6	Amazon	B0D9VTYRDR	ADS1115 16 Bit 16 Byte 4 Channel Analog to Digital Converter Precised Amplifier Board Module ADC IIC I2C - 3 pack	ADS1115 16-Bit ADC	2	\$9.99	\$19.98	Convert analog signal from FSR sensors to digital to be read by main microcontroller
7	Amazon	B01ABOPO2S	10 AWG, 10 ft Red, 10 ft Black, Stranded Tinned Copper Wire	10 AWG Wire Amazon	1	\$19.98	\$19.98	Extra wire to run power from the battery
8	Bioenno Power	BLF-4830A	48V, 30Ah LFP Battery (PVC, BLF-4830A)	Battery Bioenno Power	1	\$580.00	\$580.00	Main powersource for the entire system
9	CTR Electronics	22-494848	100 ft 22 AWG Twisted Pair Cable, CAN Bus	22 AWG Wire CTR Electronics	1	\$38.00	\$38.00	Wire to run CAN signals

10	DigiKey	CF14JT4K70CT-ND	4.7k Ohms ±5% 0.25W, 1/4W Through Hole Resistor Axial Carbon Film	4.7K Ohm Resistor DigiKey	2	\$0.10	\$0.20	Pull up resistor that pulls SDA and SCL line high.
11	DigiKey	S120CACT-ND	120 Ohms ±1% 0.25W, 1/4W Through Hole Resistor Axial	120 Ohm Resistor DigiKey	2	\$0.12	\$0.24	Two resistors on can bus lines give 60 Ω effective bus termination.
12	DigiKey	490-11886-ND	0.1 μF ±10% 50V Ceramic Capacitor X7R Radial	100nF Capacitor DigiKey	2	\$0.53	\$1.06	Filters the noise the wiper from the potentiometer creates as it slides across the resistive material.
13	DigiKey	13-MFR-25FTF 52-10K2CT-ND	10.2 kOhms ±1% 0.25W, 1/4W Through Hole Resistor Axial	10K Ohm Resistor DigiKey	16	\$0.10	\$1.60	Pull down resistors for each FSR
14	DigiKey	PDB181-K420K-103B-ND	10k Ohm 1 Gang Linear Panel Mount Potentiometer 1.0 Turns Carbon 0.2W, 1/5W PC Pins	Potentiometer DigiKey	2	\$1.29	\$2.58	Acts as a "knob" for the user to control max speed and acceleration
15	DigiKey	399-4148-ND	10000 pF ±10% 50V Ceramic Capacitor X7R Radial	10nF Capacitor DigiKey	16	\$0.30	\$4.80	In parallel with 10kΩ resistor, it helps filter outside noise to give clean signal from FSR

16	DigiKey	1528-4422-ND	Connector Kit Header, Receptacle	Connector Kit DigiKey	1	\$8.95	\$8.95	Standard signal connectors
17	DigiKey	141-PVHC4F23SS-ND	Pushbutton Switch SPST Anti-Vandal Panel Mount, Front	PVHC4F23SS E-Switch DigiKey	1	\$14.95	\$14.95	Acts as the main on/off switch for the whole system.
18	DigiKey	CKN11777-ND	Rotary Switch 4 Position SP4T 2.5A (AC), 350mA (DC) 125 VAC Panel Mount, Through Hole	Rotary Switch DigiKey	2	\$10.04	\$20.08	The rotary switches will be used to allow the user to customize the wheelchairs behavior. For example, changing maximum acceleration and maximum speed.
19	DigiKey	MCP2562	High-speed CAN Transceiver with Standby Mode and VIO Pin	MCP2562 Microchip Technology	1	\$1.43	\$1.43	Acts as the interface between a CAN controller and the physical CAN bus wires.
20	Digikey	1528-2335-ND	Resistive Analog Solder Tab	FSR Sensors Digikey	16	\$3.95	\$63.20	Force Sensitive Resistor used as the input for the users lean
21	Grainger	799KD9	Carbon Steel Rod: A36, 1 1/4 in Outside Dia, 12 in Overall Lg, Mill, Hot Rolled	1-1/8" Steel Rod Grainger	1	\$16.06	\$16.06	This rod is the stock for the dog clutch system

22	McMASTE R-CARR	90591A151	Zinc-Plated Steel Hex Nut, Medium-Strength, ISO Class 8, M6 x 1 mm Thread Size	M6 x 1mm Nut McMASTER-C ARR	1	\$3.42	\$3.42	Used in conjunction with screws to mount the gearbox to the motor mount.
23	McMASTE R-CARR	90630A110	High-Strength Steel Nylon-Insert Locknut, Grade 8, 1/4"-20 Thread Size	1/4"-20 Nut McMASTER-C ARR	2	\$4.58	\$9.16	Used in conjunction with screws to fasten the battery plate, rear motor support, and the motor mount.
24	McMASTE R-CARR	97753A124	High-Strength Grade 8 Steel Hex Head Screws, Black Corrosion-Resistant-Coate d, 1/4"-20 Thread Size, 1-1/2" Long	1/4"-20 1-1/2" Screw McMASTER-C ARR	1	\$9.29	\$9.29	Mounts the upper part of the motor mount to the underside of the wheelchair seat.
25	McMASTE R-CARR	92620A544	Zinc Yellow-Chromate Plated Hex Head Screw, Grade 8 Steel, 1/4"-20 Thread Size, 1-1/4" Long, Fully Threaded	1/4"-20 1-1/4" Screw McMASTER-C ARR	1	\$10.69	\$10.69	Used to fasten the lower part of the motor mount to the motor clamp.
26	McMASTE R-CARR	91255A540	Black-Oxide Button Head Hex-Drive Screw,	1/4"-20 3/4" Screw 	1	\$12.31	\$12.31	Used to fasten the motor mount plate to the motor mount angle bracket.

			Alloy Steel, 1/4"-20 Thread Size, 3/4" Long	McMASTER-CARR				
27	McMASTER R-CARR	90044A125	Black-Oxide Alloy Steel Socket Head Screw, 1/4"-20 Thread Size, 2" Long, Fully Threaded	1/4"-20 2" Screw McMASTER-CARR	1	\$13.22	\$13.22	Used to fasten the battery plate to the wheelchair frame.
28	McMASTER R-CARR	91257A553	Zinc Yellow-Chromate Plated Hex Head Screw, Grade 8 Steel, 1/4"-20 Thread Size, 2-3/4" Long	1/4-20 2-3/4" Screw McMASTER-CARR	1	\$13.32	\$13.32	Used to fasten the rear motor support.
29	McMASTER R-CARR	91255A547	Black-Oxide Button Head Hex-Drive Screw, Alloy Steel, 1/4"-20 Thread Size, 2-1/2" Long	1/4"-20 2-1/2" Screw McMASTER-CARR	1	\$14.48	\$14.48	Used to fasten the mitered box tube to the motor mount angle bracket.
30	McMASTER R-CARR	93839A029	High-Strength Steel Thin-Profile Hex Nut, Zinc Yellow-Chromate Plated, 1/4"-20 Size	1/4"-20 Nut McMASTER-CARR	1	\$16.00	\$16.00	Used in conjunction with screws to mount the mitered box tube to the wheelchair.
31	McMASTER R-CARR	91290A242	Black-Oxide Alloy Steel Socket Head Screw,	M5 x 0.8mm 20mm 	1	\$17.51	\$17.51	Used to fasten the gearbox to the motor mount.

			M5 x 0.8 mm Thread Size, 20 mm Long	McMASTER-CARR				
32	McMASTER-CARR	60355K691	Steel Ball Bearing, Sealed, Trade Number R8-2RS, for 1/2" Shaft Diameter	Steel Ball Bearing 1/2" ↓ McMASTER-CARR	4	\$11.18	\$44.72	Ball bearings for the shafts connecting the wheels to the output pulleys.
33	Online Metals	1165	1/2" x 1" Aluminum Rectangle Bar 6061-T6511-Extruded 12" (1 ft)	1/2x1 Bar Online Metals	1	\$6.91	\$6.91	This stock is for the motor mount Clamps
34	Online Metals	971	1" x 1" x 1/8" Aluminum Angle 6061-T6 24" (2 ft)	Angle Bracket Online Metals	1	\$7.81	\$7.81	Aluminum angle material used to create brackets.
35	Online Metals	15488	3/4" Alloy Steel Round Bar 4140-Annealed Cold Finish 12" (1 ft)	3/4 Round Bar Online Metals	1	\$11.25	\$11.25	This rod is the stock for the dog clutch system
36	Online Metals	14469	1" x 1 1/2" x 1/8" Aluminum Rectangle Tube 6061-T6-Extruded	1x1.5 Boxtube Online Metals	1	\$17.58	\$17.58	Boxtube to create a rear support for the motor.

			24" (2 ft)					
37	Online Metals	18003	1" x 2" x 1/8" Aluminum Rectangle Tube 6061-T6-Extruded 24" (2 ft)	1x2 Boxtube Online Metals	1	\$21.94	\$21.94	Boxtube for mounting the motor to the wheelchair.
38	Online Metals	1168	1/2" x 2" Aluminum Rectangle Bar 6061-T6511-Extruded 24" (2 ft)	1/2x2 Bar Online Metals	1	\$29.72	\$29.72	This stock is for the motor mount Clamps
39	Online Metals	20725	1/8" x 6" Aluminum Rectangle Bar 6061-T6511-Extruded 24" (2 ft)	1/8x6 Bar Online Metals	1	\$34.51	\$34.51	Sheet metal for the battery mount.
40	Online Metals	1149	1/4" x 6" Aluminum Rectangle Bar 6061-T6511-Extruded 36" (3 ft)	1/4x6 Bar Online Metals	1	\$65.24	\$65.24	This stock is for the motor mount plate
41	PEACO Support	PEACO-D5BLD 450-30S	450W Brushless DC motor (.5 HP), 1.43 Nm torque,	450W Brushless DC Motor, 48V 	2	\$663.45	\$1,326.90	Motor and gearbox that will provide rotational power

			3000 rpm speed, 48V rated voltage, 5.61 A, 4.3 Nm peak torque, 3.5kg, 7:1, planetary gearbox	Peaco Support				to spin the wheels on the wheelchair.
42	Sparkfun Electronics	DEV-16771	600 MHz ARM Cortex-M7 processing power, 8MB flash, Ethernet, a microSD socket, and a USB host port	Teensy 4.1 SparkFun Electronics	1	\$31.50	\$31.50	Main processing for sensor input and communication with motor controllers
43	Torque Transmission	TBD	Belt Pulley, 80 Teeth, 1 Inch Bore	Custom Part	2	\$47.00	\$94.00	This belt pulley will transmit power from the belt to the dog clutch.
44	Torque Transmission	TBD	Belt Pulley, 36 Teeth, 14mm Bore	Custom Part	2	\$32.00	\$64.00	This belt pulley will be connected to the output side of the motor, and will transmit power from the gearbox to the belt.
45	Torque Transmission	TBD	Belt, 15mm wide, 575mm long, 5M	Custom Part	2	\$35.00	\$70.00	This belt will allow the pulleys to transfer rotational forces to one another.
46	Trampa Boards	VESC 6.91	Brushless motor controller, 6th gen., 6061-T6	VESC 6 MkVI TRAMPA	2	\$249.75	\$499.50	Acts as the brain and power

			anodized aluminum heatsink enclosure, continuous 80A and burst 120A output, input 60V					regulator for an electric motor, managing its start/stop functions, speed, direction, and torque.
47	Amazon	FF-PETG CF-BK	1.75mm Carbon Fiber Filament, Lightweight and Heat-Resistant	Filament Amazon	1	\$29.99	\$29.99	Allows for precise parts to be printed for the idler pulley
Total							\$3,337.62	

The projected cost of our solution is \$3,337.62. Including taxes, and the estimated shipping costs, it will cost \$3,611.62. Our total budget is \$4,000 which means the subtotal and total are 83% and 90% of our budget, respectively. The pulleys and belts purchased from Torque Transmission (rows 43, 44, and 45) are custom parts. Because of this, they do not have part numbers and we cannot provide a link to purchase them. Additionally, we do not have shipping costs for these items.

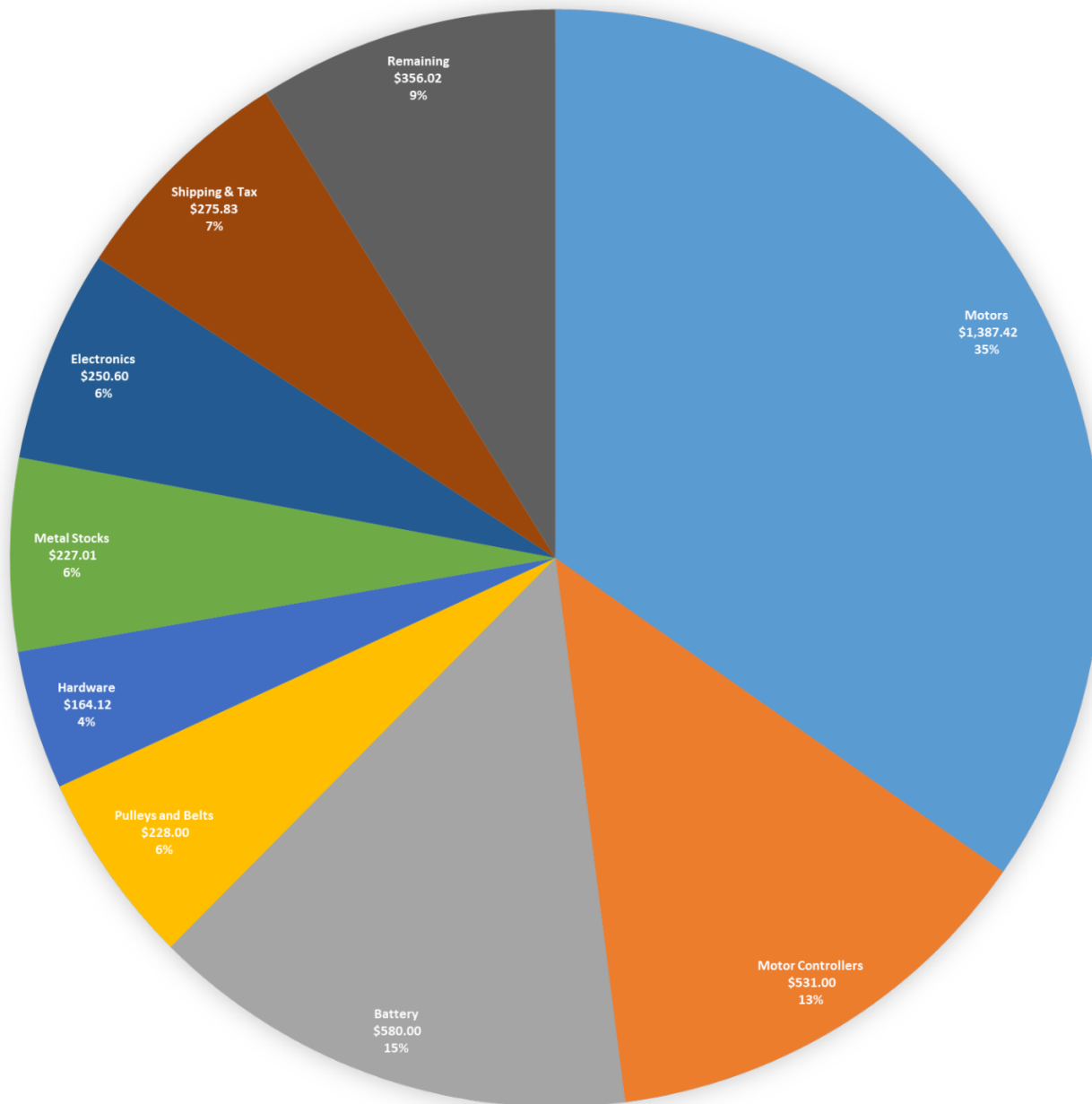


Figure 11: Pie Chart Giving a Visual Representation of the Bill of Materials

15. Schedule:

15.1 Timeline

15.1.1 Aug 24 – Sept 6, 2026 [Week 1 + 2]

At the beginning of the semester, the team will complete final design refinements and initiate purchasing of all critical components. Early efforts will prioritize long-lead items such as drivetrain components, motors, and specialized electronics to ensure timely delivery. In parallel, manufacturing drawings will be finalized to enable immediate fabrication.

15.1.2 Sept 7 – Sept 20, 2026 [Week 2 + 3]

During this period, the team will continue purchasing remaining components and begin fabrication of custom mechanical parts using the machine shop. Initial components will be received and inspected, and early-stage assembly and test fitting will begin on the wheelchair platform.

15.1.3 Sept 21 – Sept 27, 2026 (Prototype ~50% Complete) [Week 5]

By this time the team will produce a medium fidelity prototype with the following properties: Major drivetrain hardware will be mounted to the wheelchair, and basic electrical and control systems will be integrated sufficiently to enable early-stage testing. The system will be capable of initial validation through either limited physical motion or be ready for subsystem integration and testing.

15.1.4 Sept 28 – Oct 4, 2026 [Week 6]

Following initial integration, the team will complete a formal testing plan and continue refining subsystem integration. Electrical packaging, wiring, and safety features such as the emergency stop will be incorporated as testing readiness improves.

15.1.5 Oct 5 – Oct 18, 2026 [Week 7 + 8]

The team will focus on full system integration, including mechanical, electrical, and control subsystems. Iterative testing and debugging will be conducted to resolve issues identified during early prototype operation.

15.1.6 Oct 19 – Oct 25, 2026 (Functional Prototype) [Week 9]

By this time, the team will demonstrate a functional prototype capable of lean controlled motorized motion. Core electrical systems and safety features will be operational, enabling repeatable and safe human testing of system behavior.

15.1.7 Oct 26 – Nov 20, 2026 [Week 10 + 11 + 12 + 13]

The remaining development period will focus on iterative refinement, performance improvement, and system reliability. The team will conduct repeated testing to improve control response, mechanical robustness, and overall usability of the system.

15.1.8 Nov 23 – Nov 29, 2026 (Thanksgiving Break) [Week 14]

Limited progress is expected during this period due to the academic calendar.

15.1.9 Nov 30 – Dec 11, 2026 [Week 15]

Final refinements will be completed, and the prototype will be prepared for demonstration. Documentation, validation results, and presentation materials will be finalized for sponsor review and project showcase.

15.1.10 Dec 14 – Dec 18, 2026 (Final Deliverables) [Week 16]

The project will conclude with final design documentation, presentation, and handoff to the sponsor, including all relevant design files and testing results.

15.2 Schedule Risks

The primary risks to the project schedule are associated with component procurement, fabrication dependencies, and subsystem integration. As no major components are currently in hand, delays in ordering or shipping could impact early assembly and integration efforts. To mitigate this risk, the team will prioritize purchasing of critical components at the beginning of the semester and identify alternative suppliers where possible.

Fabrication of custom components in the machine shop presents an additional risk due to potential scheduling constraints and the possibility of rework during iteration. The team will mitigate this by finalizing manufacturing drawings early and allowing buffer time for adjustments.

Finally, integration of mechanical, electrical, and control systems represents a key technical risk. As system functionality depends on successful coordination between subsystems, delays in one area may impact overall progress. To address this, the team will pursue parallel development across disciplines and aim to achieve early-stage integration by the 50% prototype milestone to enable iterative testing and reduce late-stage risk.

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17. Appendices:

17.1 Appendix A: Motor Mount FEA Analysis

In this appendix we discuss in detail the analysis of the motor mount and include FEA results and setup details.

Motor Mount Clamp Compressive Study

In this study we consider the upper part of one clamp being pushed down onto the tube it sits on. Thus the primary constraint is simply the tube as a roller slider on the bottom of the clamp. The force is applied to the top face as 500 lbf. This value was chosen because it is likely far above what could reasonably be expected. The internal threads of the model were removed to enable meshing of the part. The results are clear, with a FoS of 21, this failure mode is not a concern.

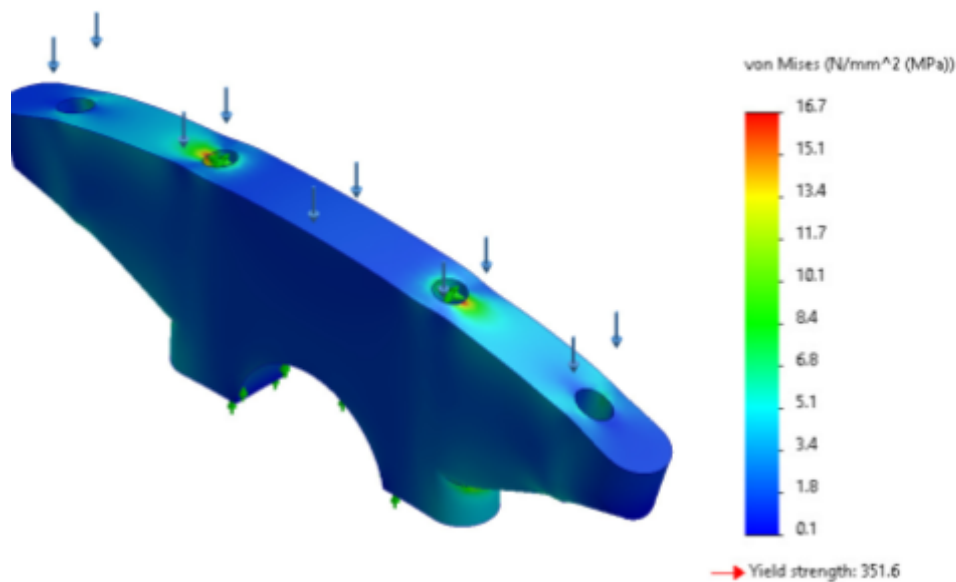


Figure A.1: Stress of compressive loading of the motor mount clamp

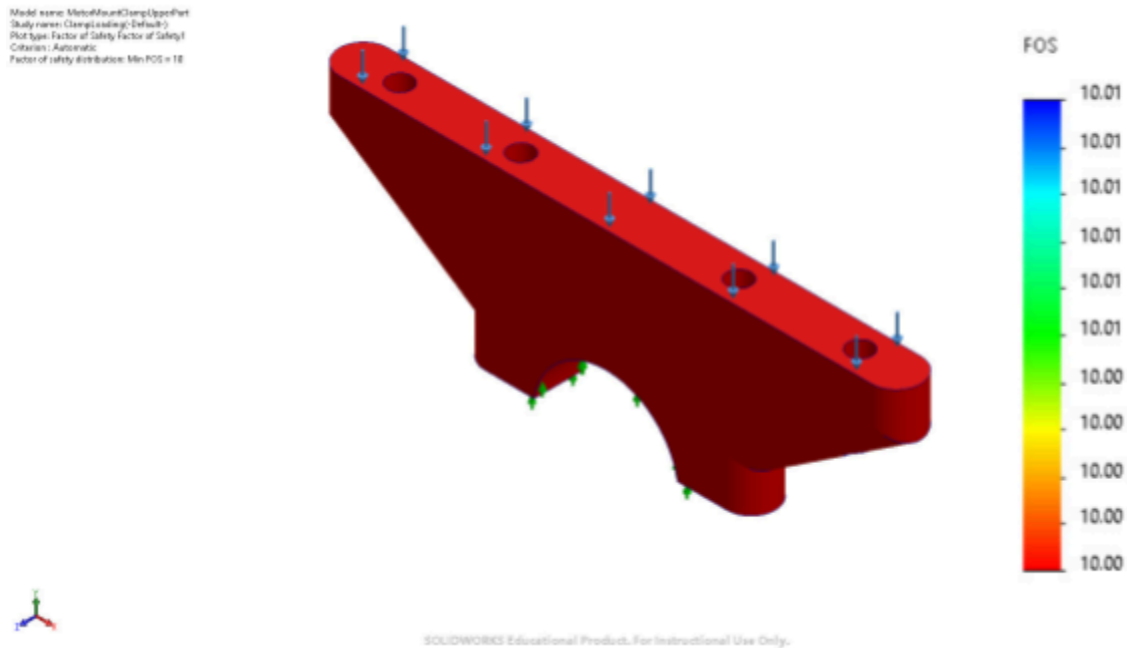


Figure A.2: Safety factor of compressive loading of the motor mount clamp

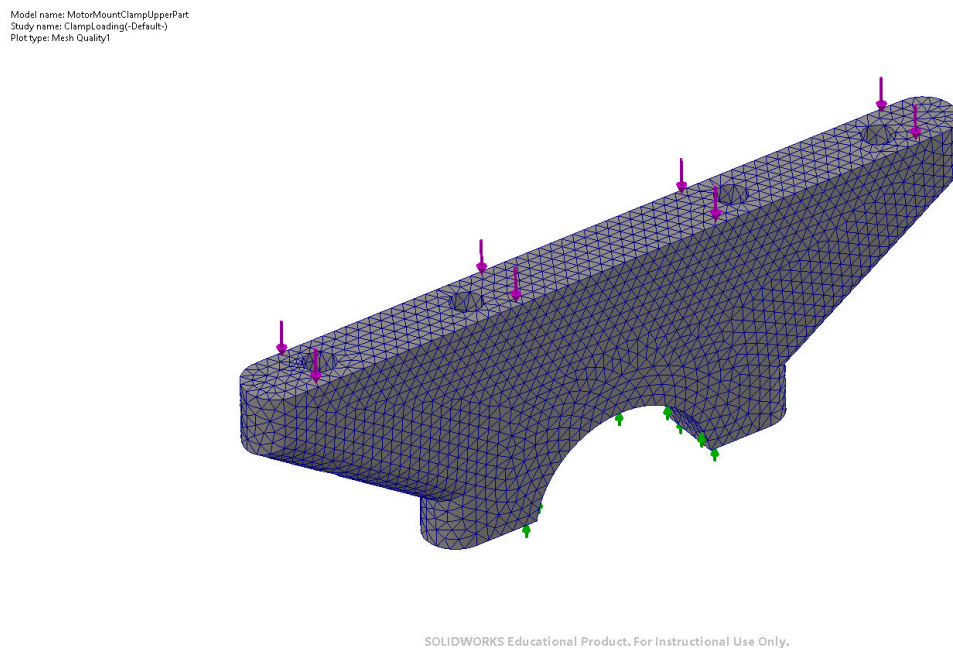


Figure A.3: Mesh for motor mount clamp studies

Motor Mount Clamp Tensile Study

In this study we consider the upper part of one clamp being pulled up from the tube it sits on. The 500lbf force is applied by pulling up on the bolts that the motor mount plate attaches to. This is a slight assumption

as the force is really applied on threads, but the features are too complex to mesh within a reasonable amount of time, let alone solve. Similarly, the model is fixtures using the clamp bolts. The bolt holes are constrained axially and radially representing the threads and pinning of the bolt. The results are clear, with a FoS of 9.28, this failure mode is not a concern.

The mesh is the same as the compressive study.

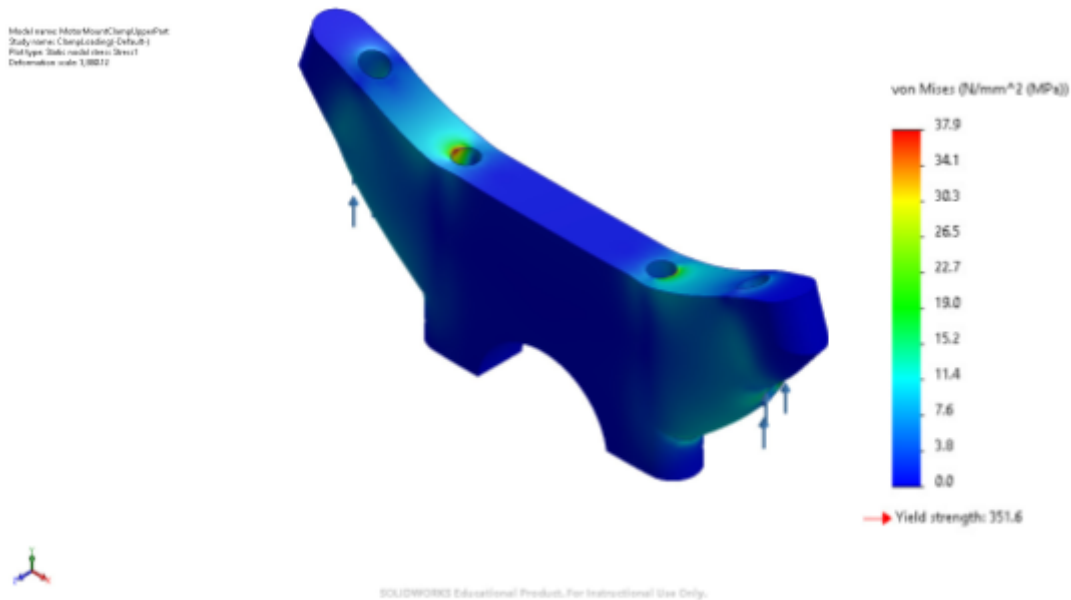


Figure A.4: Tensile Stress on Motor Mount Clamp

17.2 Appendix B: Determination of torque for angular acceleration

Spec H requires that the wheelchair be capable of reaching 200deg/s of angular velocity in .5s. Following a very similar derivation to that found in the problem analysis we will find an equation relating angular acceleration to the torque which must be applied to a wheel to achieve it.

Total moment of inertia found using CAD model. The human was set to a weight of 150lb, batteries and motors had their masses overridden to match supplier documentation. The mass moment of inertia about the vertical axis, taken at the center of mass was found to be 4.54 kgm². While the model is not fully solidified and the components were imperfectly mirrored, the total weight was 102kg, which closely matches the 105kg used in our analysis. We assess that, considering the amount and gravity of changes left to make, accuracy of the model of the human, and all other considerations, that this estimation of the moment of inertia is very accurate and approaches levels of accuracy that could be obtained through experimental methods.

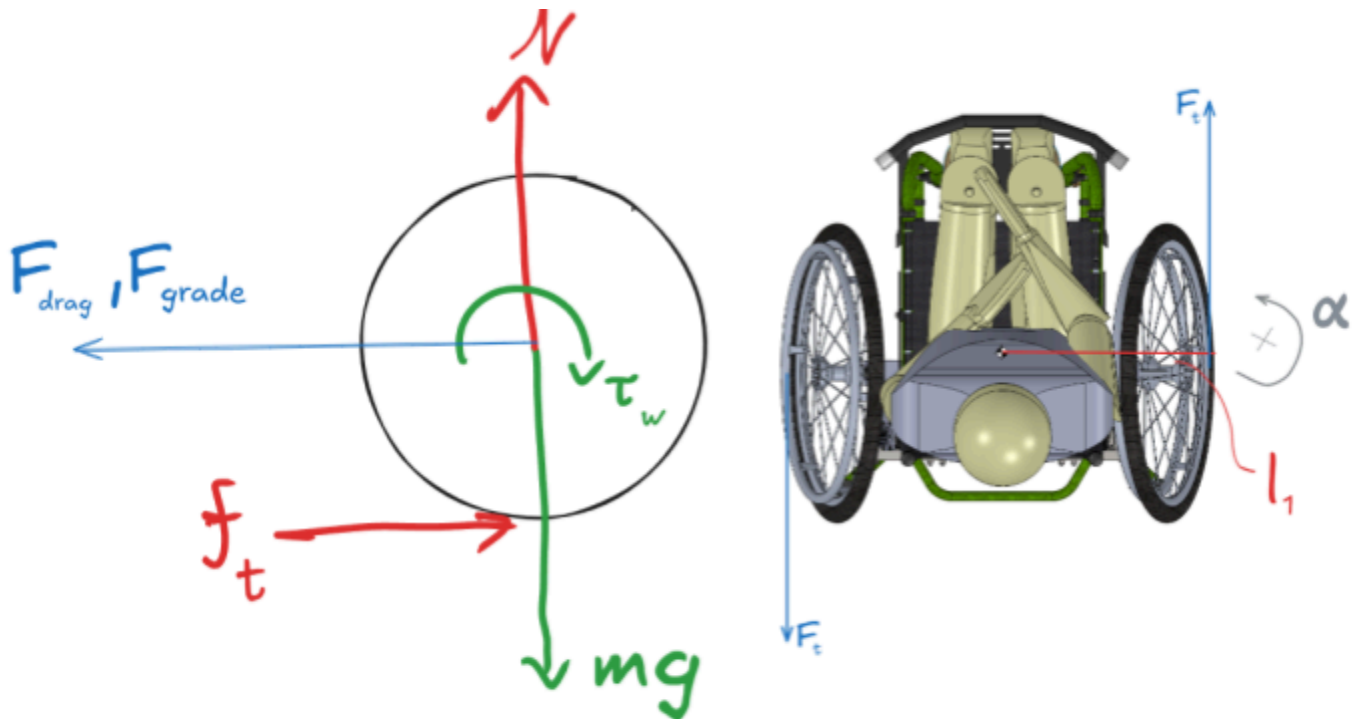


Figure B.1: Free body diagrams of wheelchair rolling and Traction Forces

As seen in Figure B.1 we consider the torque applied to the wheel, the traction force with the ground, gravity and drag for arguments sake. F_{grade} is the component of gravity in the horizontal direction, which will be promptly set to 0. We will assume that the wheels roll without slipping.

Sum of moments about the center of mass:

$$F_{grade} = mgsin(\theta) = 0$$

$$F_{drag} = \frac{1}{2} \rho C_d A v^2 \approx 0$$

$$F_{rr} = C_{rr} mg \cos(\theta) = C_{rr} mg$$

$$\Sigma M_{com} = I \alpha$$

$$\Sigma M_{com} = 2l_1 F_t - 2l_1 C_{rr} mg = I_{tot} \alpha \quad (B.1)$$

Sum of torques about axle:

$$\Sigma \tau = I \alpha$$

$$\tau_w - F_t r = I_{wheel,eq} \alpha$$

$$\text{Where } I_{wheel,eq} = I_w + I_o$$

Utilizing our no slip assumption we know $\alpha = \frac{a}{r}$. But in this case is the linear acceleration of the point of contact of the wheel about the center of mass. Thus we can rewrite α as:

$$\alpha = \frac{l_1 \alpha_{tot}}{r}$$

Substituting

$$\tau_w - F_t r = I_{wheel,eq} \frac{l_1 \alpha_{tot}}{r}$$

$$\tau_w - I_w \left(\frac{l_1 \alpha_{tot}}{r} \right) = F_t r$$

$$F_t = \frac{\tau_w}{r} - \frac{I_w l_1 \alpha_{tot}}{r^2} \quad (B.2)$$

Sub eq. 2 into eq. 1

$$2l_1 \left(\frac{\tau_w}{r} - \frac{I_w l_1 \alpha_{tot}}{r^2} \right) - 2l_1 C_{rr} mg = I_{tot} \alpha_{tot}$$

$$\frac{\tau_w}{r} - \frac{I_w l_1 \alpha_{tot}}{r^2} = \frac{I_{tot} \alpha_{tot} + 2l_1 C_{rr} mg}{2l_1}$$

$$\tau_w = \frac{r(I_{tot} \alpha_{tot} + 2l_1 C_{rr} mg)}{2l_1} + \frac{I_w l_1 \alpha_{tot}}{r} \quad (B.3)$$

We can find the required angular acceleration as follows:

$$200 \frac{deg}{s} = 3.49 \frac{rad}{s}$$

$$\omega = \alpha t \Rightarrow \alpha = \frac{\omega}{t} = 6.98 \frac{rad}{s^2}$$

We can now evaluate this equation at the following values to find the torque that must be applied at each wheel to achieve the specified angular acceleration:

Table B.1: Angular Velocity Variables

Variable	Value	Note
r	.3048 m	Geometrical fact
I_{tot}	4.54 kg m ²	Accurate estimation from CAD
α_{tot}	6.98 rad/s	Required by Spec H
l_1	0.415925 m (16.4in)	CAD measurement
C_{rr}	.002	From Problem Analysis
m	102kg	CAD measurement
g	9.81 m/s ²	Physical constant
I_w	0.170 kg m ²	Experimentally determined during problem Analysis

Evaluating at these values gives that the torque supplied at each wheel should be 13 Nm.

17.3 Appendix C: Motor Performance Curve Generation Code

```
import numpy as np
import matplotlib.pyplot as plt

# Pre-gearbox motor specs
rated_torque_motor_nm = 1.43
peak_torque_motor_nm = 4.30
rated_speed_motor_rpm = 3000.0
no_load_speed_motor_rpm = 3300.0
gear_ratio = 16.0

# Convert to gearbox-side torque for plotting
rated_torque_gearbox_nm = rated_torque_motor_nm * gear_ratio    # 22.88 Nm
peak_torque_gearbox_nm = peak_torque_motor_nm * gear_ratio      # 68.80 Nm

# User-requested reference lines
horizontal_ref_nm = 49.202
vertical_ref_rpm = 2464.0

rpm = np.linspace(0, no_load_speed_motor_rpm, 1200)

# Continuous torque curve
cont_torque_motor = np.where(
    rpm <= rated_speed_motor_rpm,
    rated_torque_motor_nm,
    rated_torque_motor_nm * np.maximum(0, (no_load_speed_motor_rpm - rpm) /
    (no_load_speed_motor_rpm - rated_speed_motor_rpm))
)
cont_torque_gearbox = cont_torque_motor * gear_ratio

# Peak torque curve
peak_flat_end_rpm = no_load_speed_motor_rpm * (2000.0 / 15000.0)
peak_torque_motor = np.where(
    rpm <= peak_flat_end_rpm,
    peak_torque_motor_nm,
    peak_torque_motor_nm * np.maximum(0, (no_load_speed_motor_rpm - rpm) /
    (no_load_speed_motor_rpm - peak_flat_end_rpm))
)
peak_torque_gearbox = peak_torque_motor * gear_ratio

# Intersections
mask = rpm >= peak_flat_end_rpm
peak_intersection_x = np.interp(horizontal_ref_nm,
```

```

peak_torque_gearbox[mask][::-1], rpm[mask][::-1])
peak_intersection_y = horizontal_ref_nm

cont_intersection_x = vertical_ref_rpm
cont_intersection_y = np.interp(vertical_ref_rpm, rpm, cont_torque_gearbox)

fig, ax = plt.subplots(figsize=(10, 5.6))

# All lines different colors
line1, = ax.plot(rpm, peak_torque_gearbox, '--', linewidth=2, color='tab:red',
label='Peak Torque')
line2, = ax.plot(rpm, cont_torque_gearbox, linewidth=2, color='tab:blue',
label='Continuous Torque')
line3 = ax.axhline(horizontal_ref_nm, linestyle=':', linewidth=2,
color='tab:green', label='Peak Acceleration')
line4 = ax.axvline(vertical_ref_rpm, linestyle=':', linewidth=2,
color='tab:purple', label='Max Speed')

# Intersection points (kept out of legend)
ax.plot(peak_intersection_x, peak_intersection_y, 'o', color='black',
markersize=8, label='_nolegend_')
ax.plot(cont_intersection_x, cont_intersection_y, 'o', color='black',
markersize=8, label='_nolegend_')

# Labels and title
ax.set_title('450W Motor Performance Curve')
ax.set_xlabel('Speed (RPM)')
ax.set_ylabel('Gearbox Torque (N·m)')

ax.set_xlim(0, no_load_speed_motor_rpm * 1.02)
ax.set_ylim(0, max(peak_torque_gearbox.max(), horizontal_ref_nm) * 1.15)

ax.grid(True, alpha=0.35)

# Legend in top right
ax.legend(handles=[line1, line2, line3, line4], loc='upper right')

plt.tight_layout()

out_path = "/mnt/data/450W_motor_performance_curve_torque_only.png"
plt.savefig(out_path, dpi=200, bbox_inches='tight')
plt.show()

print(f"Saved: {out_path}")

```

17.4 Appendix D: Price Per Pressure Sensor

As the sensor price is not readily available on the market, we contacted the manufacturer and have this email from them with the given information on how much the price of the sensors cost:

Thank you for contacting us. It's a pleasure to meet you.

Your project—a tilt control wheelchair system for wheelchair basketball—sounds very promising, and we are happy to support your graduation project.

Regarding the FSR402S-30-10000, it is a standard single-point force sensor suitable for basic force detection.

However, depending on your application scenario (detecting the athlete's position and movement on the seat), we suggest you consider the following:

1. Applicability of the FSR402

If you only need a few discrete sensing points and the goal is to detect the presence or relative pressure changes, then the FSR402 can be used.

However, if you require other functions, such as pressure distribution, etc., the FSR402 may have limitations.

2. Recommended Solutions

For your application, we recommend two solutions:

Solution A: Multiple FSR Sensors (Discrete Layout)

Place multiple FSR402 (or similar models) in key locations.

Lower cost, simpler integration.

Suitable for basic position detection.

Solution B: Custom FSR Matrix Sensor (Recommended)

Use a matrix array to capture pressure distribution on the cushion.

Better for detecting posture, motion, and center of gravity shift.

We can support custom sizes, shapes, and sensor grid designs.

3. Pricing and Samples

For FSR402, the unit price is \$2 USD per piece. For student projects, we are happy to provide samples and can offer competitive pricing accordingly.

However, please note that shipping costs are the responsibility of the applicant.

17.5 Appendix E: Additional CAD Renders

Here are a variety of pictures of the CAD model so as to avoid clogging up the design description.



