The Work Hard Play Hard Roller Coaster

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May 28, 2025

Contents

1 Introduction & Background

This report outlines the design and operational principles of an energy-free roller coaster for kids, a ride that eliminates the need for external power sources by relying solely on human-generated effort and mechanical advantage. Unlike conventional roller coasters that use motors and electrical systems, this design requires the parents to manually pull the coaster to its peak via a rope integrated with a gear-driven mechanism. By optimizing the gear ratios, the system ensures that the force application is efficient by minimizing the physical strain on the parents and maximizing the force of the lift. Once the coaster reaches the peak, the gravitational potential energy does the rest as it converts to kinetic energy and takes the riders through the track.

Previous implementations of energy-efficient roller coasters have demonstrated the viability of alternative energy methods. The Green Dragon roller coaster in the UK, for example, utilizes a boarding platform that all riders stand on. Their combined weight drives the platform down slowly, and using this potential energy, the car is able to be lifted up to the top of the platform and the riders climb another set of stairs to board. Our design builds on this concept but introduces a mechanically assisted ascent by using a gear and shaft driven system to elevate the roller coaster. The incorporation of gears will allow us to vary the gear ratios to adjust torque and speed. Lower gear ratios provide increased mechanical acceleration, enabling a group of riders to apply lower individual force while still generating enough power to lift the coaster. This ensures a balanced distribution of force, and reduces physical strain allowing for a sustainable and repeatable process. This design not only achieves zero energy operation but also enhances user engagement, making the ride an interactive experience and a practical demonstration of mechanical principles.

2 Overall Design

This roller coaster is designed for children and their parents. Parents are tasked with pulling the coaster cars and their kids in them up to the initial descent of the ride. This allows for the ride to not rely on electricity and give the park patrons a workout. Fourteen riders ride the coaster at one time while mainly parents will be designated to pull the coaster up to its peak. Though this ride a designed for children, it is possible for some some adults to ride with their children. Parents tasked with pulling will be pulling on a rope that is in a loop between a winch drum and a pulley. The rope is wrapped around the winch drum multiple times to create enough fiction so when the rope is pulled the winch is advanced and no slippage occurs. The winch is supported by bearings and attached to a pinion via the input shaft. The pinion meshes with an intermediate gear. The intermediate gear, also supported by bearings, is attached to the bull wheel sprocket via the output shaft. The bull wheel sprocket engages with a chain responsible for pulling the cars up the lift hill. Mechanical advantage is generated with gear ratios between the drum, pinion, intermediate gear and bull wheel sprocket allowing people to pull up the coaster with ease. Queuing riders are encouraged to help with pulling to aid the parents of the riding children. A sketch of the full assembly is shown below in Figure [1].

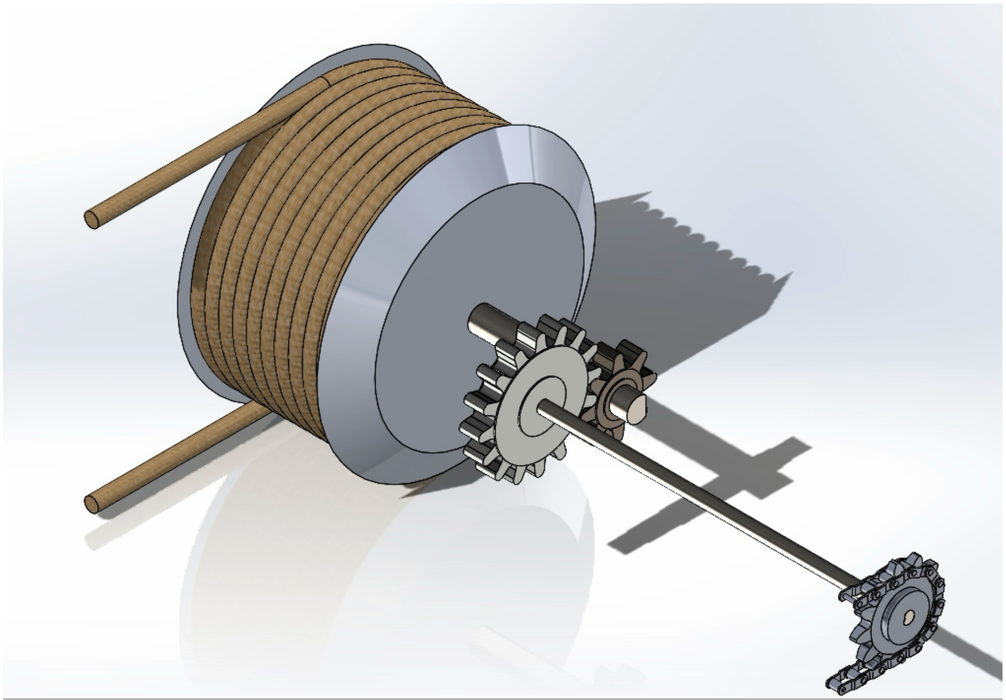


Figure 1: Roller Coaster Lift Mechanism

The chain lifting the cars is a classic way roller coasters are raised. The chain resembles a bike chain but significantly larger. The weight of all the riders is measure before the ride debarks to ensure the maximum weight is not exceeded. Once the roller coaster has made it to the peak it will fly down the ramp and speed through the length of the ride until it is back in the loading station. The ride is designed such that the train returns to the loading dock with very little speed. The final momentum is taken away using a spring that the front cart runs into. The tracks on the station will be slightly sloped down so once the coaster is loaded up the spring is released and the train coasts to the climbing chain where it will be pulled up the lift hill.

2.1 Rolly 95 Train

A specialized *Rolly 95* train assembly is sourced. The electric motors and breaks have been stripped reducing the 1520 kg mass to 1200 kg. The flexible design of the train allows it to take sharper turns. These trains incorporate the most update technologies in construction and safety in respect to international norms EN and ASTM – CSEI – GOST. Known for their low vibration and noise levels, and the smooth linear flow these trains are perfect for a children's ride. All rollercoaster are equipped with adjustable safety bars.

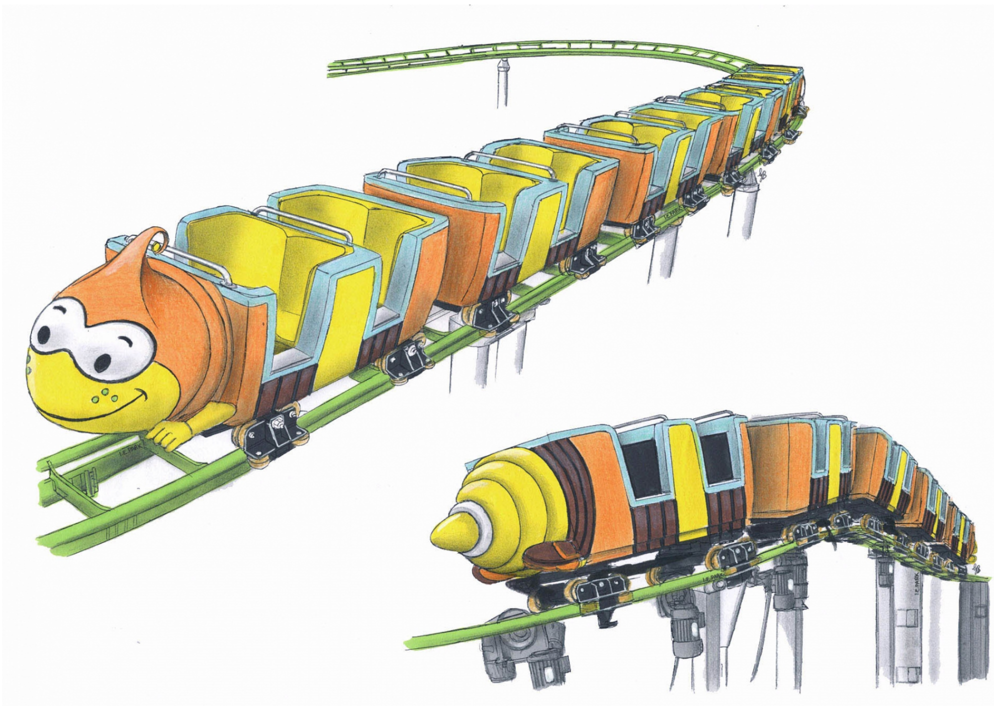


Figure 2: Rolly 95 Train Assembly

2.2 Rope and Drum System

2.2.1 Winch Drum

The rope driven drum mechanism was inspired by winches on sailboats. Sailboats utilize winches to apply enormous forces to a sail via a line secured by friction. *Harken*, an established company specializing in sailing hardware, sells winches rated to carry loads of 750 kg. The loads applied to the winch drum of this rollercoaster will be closer to 170 kg.

Figure 3: Chrome High Capacity Drum Rigging Winch (750) made by Harken

This winch will not be sourced for the rollercoaster design, is referenced to show that it is feasible to engage a drum with a rope simply secured with friction. Winches on sailboats use leverage and mechanical advantage to pull line. Alternatively this roller coaster design requires rope to be pulled to spin a drum. The 6" diameter drum made by harken is designed for 5 wraps of a 22 mm rope. The drum in the rollercoaster design will be much larger, with a diameter of 1 meter. The rope will also be far larger than 22 mm to improve grip. Keeping tension in the rope will increase friction so a spring loaded idler-arm with a pulley mounted to the end is implemented into the rope loop. Below is an image of an idler arm assembly on a supercross BMX bike. An equivalent idler arm assembly system is used where the sprocket is replaced with a pulley and rather than a chain a rope is used.

Figure 4: Idler arm assembly

2.2.2 Rope

The rope selected for the roller coaster needed to interface well with the general public. A 'Tug of War' rope was selected as it is built for people to pull on. Our group plans to use Ravenox 1 inch Twisted Cotton Tug of War Rope because of its great grip, tensile strength of 2,375 lbs, and elimination of rope burns. 1680 N is the planned force put on the rope, equating to 377 lbs, which is a safety factor of over 6. Incorporating a large safety factor will mitigate potential injury to the parents pulling.

2.3 Human Power Input

This rollercoaster utilizes human strength to provide power. In a study, 8 male and 8 female university students of working age participated in various physical exercises, during which their isometric pull and push strengths were measured. The average pull strength of standing men and women was 147.58 N and 85.09 N. The average pull strength of seated men and women was 224.39 N and 120.50 N. All exercises were performed with only the right arm. When pulling the rope people will use their entire body likely resulting in far greater maximum forces. This ride is designed for children so parents will provide most of the pulling power. None the less some of the pulling may be done by children, elderly people, or people with disabilities so the rope should be easy to pull. It is a reasonable assumption to assume that 14 people can easily pull with an average force of 120 N per person resulting in a total force of 1680 N.

The human power transmitted through the rope can be calculated with the formula below. If 1680 N of force is applied to the rope while it is pulled at 1 meter per second,a comfortable pulling rate, 1680 Watt's is supplied to the system.

$$Power = Force \times Velocity \tag{1}$$

2.4 Power Requirements & Mechanical Loss Correction

An empty train of cars weighs 1200 kg. This train will carry a maximum of 14 passengers. Although this rollercoaster is designed for children, up to 7 adults and 7 children will be allowed to ride. With an upper estimate for weight being 50 kg per person, the weight of riders is 700 kg. A full train with 7 adults and 7 children will weigh roughly 1900 kg. The roller coaster will rise 8 meters requiring approximately 150 kJ of energy. This energy value is calculated with the formula below where U is gravitational potential energy, M is mass, g is the acceleration due to gravity and h is the change in height.

$$U_{gravitational} = M * g * h \tag{2}$$

Mechanical losses are inevitable in mechanical systems. Theoretical losses are difficult to calculate so to account for unknown losses a correctional factor will be implemented. Components contributing to energy losses include:

- Frictional losses in bearings and gears
- Rolling friction
- Energy lost due to component deformation or backlash
- Kinetic energy during start up

Frictional losses in bearings and gears cannot be assumed negligible but if properly lubricated quality components are sourced friction is greatly minimized. Rolling friction losses are very minimal due to the extremely slow climb rate of the roller coaster. Component deformation or backlash should be minimized not only for efficiency reasons but more importantly for safety. A large safety factor accounting for fatigue of component will minimize these energy losses. Lastly power requirements calculated using equation [2] do not account for the power of start up. Additional energy is required to overcome the rotational kinetic energy required to accelerate the gears and shafts. Start up energy is accounted for in the correction factor.

The power requirement will be multiplied by a correction factor of 1.25, accounting for 20% energy loss. The corrected energy requirement is 186 KJ, calculated with equation [3]

$$U_{total} = 1.25(M * g * h) \tag{3}$$

If the pulling power is 1680 Watt's it will take roughly 1 minute and 50 seconds for the cars to reach the top. Note that the pulling power is calculated using a low force designed for easy use, and the weight of riders is an upper estimate where 7 adults are riding.

The cars are pulled up a 8 meter high hill at 30 °slope which results in a 16 meter pull distance. With the pulling power and energy requirements stated above the cars would travel 14.4 centimeter per second up the lift hill.

With the rope being pulled at one meter per second and the cars travel at 14.4 centimeter per second. The sizes of the drum, pinion, intermediate gear, and bull wheel sprocket are selected to provide mechanical advantage to allow this system to operate smoothly.

2.5 Gears

Gears create a system that provides mechanical advantage by transmitting torque and altering rotational speed and force. This mechanical advantage is divided over 3 stages.

1. The rope driven drum transmits motion to the pinion (gear 1) via the input shaft
2. The pinion (gear 1) meshes with the intermediate gear (gear 2)
3. The intermediate gear (gear 2) transmits motion to the bull wheel sprocket (gear 3) via the output shaft

2.5.1 Stage *I* - Input Shaft Assembly

The first stage is a single rigid body that includes the input shaft, drum, and pinion.

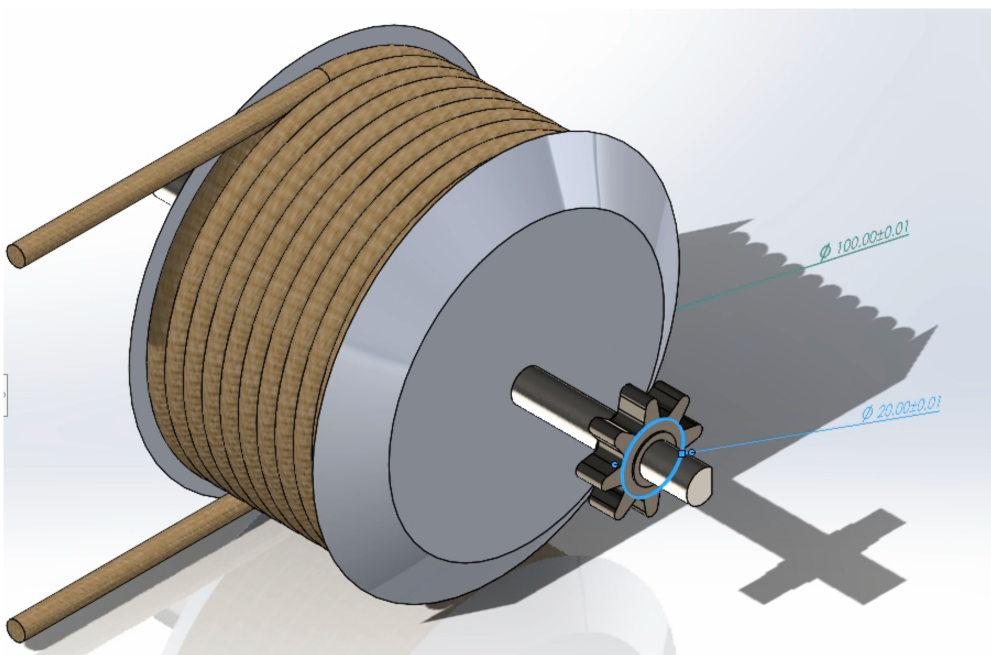


Figure 5: Input Shaft Assemble

By varying the radius of the drum and the pinion the force and velocity can be traded due to conservation of power. Equation [4] shows the relationship between tangential velocity (v)

and radius (r) of the drum and pinion both rotating at the angular velocity of the input shaft ($\omega_{input\ shaft}$). Equation [5] shows the sum of torques applied to the input shaft when rotating at a constant rate. Equation [6] is a combination of equations [4] and [5] where conservation of power is applied. The trade off between tangential velocity and force is shown. Note that force is applied to the drum and pinion in opposite directions.

$$\omega_{input\ shaft} = \frac{v_{drum}}{r_{drum}} = \frac{v_{pinion}}{r_{pinion}} \tag{4}$$

$$\Sigma\tau_{input\ shaft} = 0 \quad \rightarrow \quad F_{drum} \cdot r_{drum} = -F_{pinion} \cdot r_{pinion} \quad (Steady\ State) \tag{5}$$

$$F_{drum} \cdot v_{drum} = -F_{pinion} \cdot v_{pinion} \quad (Steady\ State) \tag{6}$$

1680 N of force is applied to the drum at 1 m/s via the rope. With a drum radius of 50 cm the angular velocity of the shaft is 2 rad/s and 840 N-m of torque are applied to the shaft via the drum. With a pinion of 10 cm radius the tangential velocity of the pinion is 0.2 m/s and 8400 N of force are applied to the teeth of the pinion. Force is magnified by a factor of 5.

2.5.2 Stage II - Pinion & Intermediate Gear

The second stage consists of the pinion meshed with the intermediate gear.

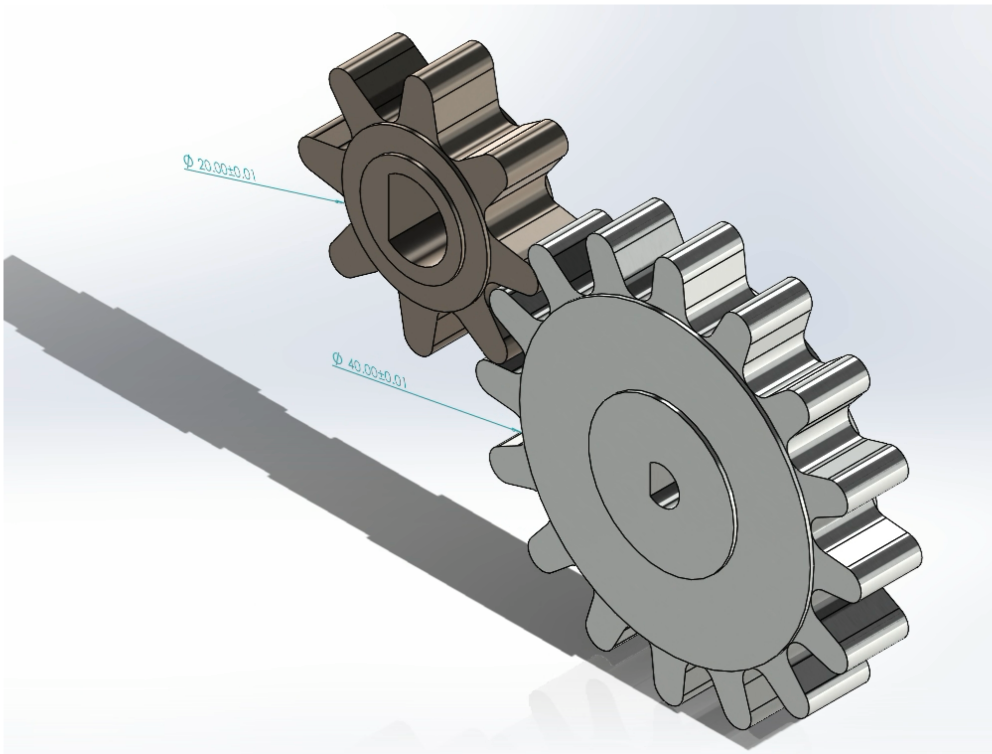


Figure 6: Internal Pinion and Gear

Mechanical advantage is generated between the pinion and intermediate gear due to gear ratio, by trading angular velocity (ω) for torque (τ). The forces present at the point of contact in the meshing of the pinion and intermediate gear are equal in magnitude and opposite in direction when rotating at a constant rate. Additionally, the velocities at the point of contact are equal. Note that during torque calculations, the gear radius is used as the moment arm from the gear's center of rotation to the contact point. Equation [7] shows the relationship between angular velocity (ω) and radius (r) of the pinion and intermediate gear when tangential velocities are equal at the the point of contact due to meshing. Equation [8] shows the sum of forces at the point of contact between the pinion and intermediate gear when rotating at a constant rate. Equation [9] is a combination of equations [7] and [8] where conservation of power is applied. The trade off between angular velocity and torque is shown. Note that torque is applied to the pinion and gear in opposite directions.

$$V_{contact} = \omega_{pinion} \cdot r_{pinion} = \omega_{intermediate\,gear} \cdot r_{intermediate\,gear} \tag{7}$$

$$\Sigma F_{contact} = 0 \quad \rightarrow \quad \frac{\tau_{pinion}}{r_{pinion}} = -\frac{\tau_{intermediate\,gear}}{r_{intermediate\,gear}} \quad (Steady\ State) \tag{8}$$

$$\tau_{pinion} \cdot \omega_{pinion} = \tau_{intermediate\,gear} \cdot \omega_{intermediate\,gear} \quad (Steady\ State) \tag{9}$$

The input shaft transmits 840 N-m of torque via the pinion rotating at 2 rad/s. The velocity at the contact between the pinion and intermediate gear is 0.2 m/s. With a radius of 20 cm, 1680 N-m of Torque is transmitted to the output shaft via intermediate gear rotating at 1 rad/s. Torque is magnified by a factor of 2.

2.5.3 Stage III - Output Shaft Assembly

The third and final stage is a single rigid body that includes the output shaft, intermediate gear and bull wheel sprocket.

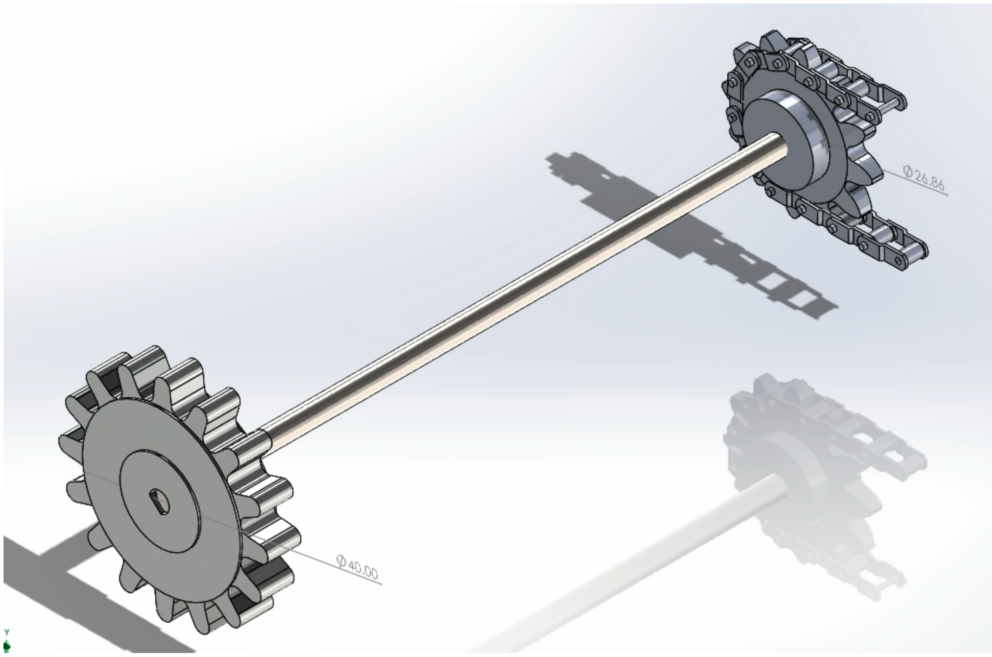


Figure 7: Output Shaft Assemble

This stage generates mechanical advantage using the same method used in stage I. By varying the radius of the intermediate gear and bull wheel sprocket, force and velocity can be traded due to conservation of power. Equation [10] is derived using the conservation of power, following the same approach as in Equation [6]. The trade off between tangential velocity and force is shown. Note that force is applied to the intermediate gear and bull wheel sprocket in opposite directions.

$$F_{intermediate\ gear} \cdot v_{intermediate\ gear} = -F_{bull\ wheel\ sprocket} \cdot v_{bull\ wheel\ sprocket} \quad (Steady\ State) \tag{10}$$

1680 N-m of Torque is transmitted to the output shaft via intermediate gear rotating at 1 rad/s. With a radius of 14.4 cm the bull wheel sprocket transmits 11667 N of force to the chain at a speed of 14.4 cm/s.

3 Design Components

3.1 Braking System

The braking system is a very important component of the roller coaster's mechanical system, responsible for absorbing shock, storing energy, and enabling smooth motion throughout operation. It plays a key role in managing dynamic forces such as sudden deceleration while maintaining structural integrity under repeated loading.

3.1.1 Requirements

The components required by the braking system are two brackets that lay at the end of the track and a compression spring between them. Additionally, as you will see in the next section, we address a friction based braking system to come right before the spring system.

To effectively absorb this energy without permanent deformation, the spring must be carefully designed with the appropriate stiffness and material. The braking system for the roller coaster must be engineered to safely decelerate a fully loaded car with a total mass of 1900 kg, traveling at an initial velocity of 10 m/s, within practical stopping distances. The system must maintain a deceleration limit of 5g (49 m/s^2) to ensure rider safety and comfort, while incorporating a safety factor of 2 to account for extra loads, material variability, and operational conditions. It should integrate friction-based braking mechanisms to dissipate 65% of the total kinetic energy, complemented by spring-based braking, specifically a spring design, which absorbs the remaining 35% with a controlled compression distance of less than 0.5 m.

Materials selected for the braking components must exhibit high tensile strength, fatigue resistance, efficient heat dissipation, and corrosion resistance, ensuring long-term durability in repeated use and varying environmental conditions. The system's design should balance efficiency, compactness, and reliability, allowing seamless integration into the coaster's track design while adhering to established safety standards and engineering best practices.

3.1.2 Choices and Options

Some choices that were considered were the types of braking and spring choices. Originally, the team was going to utilize just a simple spring system that the car comes to a stop with. It was determine through a series of calculations that with our design constraints mentioned above, the spring would have to compress nearly 11 meters. This was incredibly unrealistic for a number of reasons, the most significant being the high likelihood of the spring buckling under that much compression. Additionally, allocating 11 meters for a braking system in the track design was unrealistic. We determined that we would need to dissipate some of the kinetic energy from the roller coaster before the car comes to a complete stop via the spring. So, we decided to implement a friction based breaking system before the spring system. Another choice that had to be considered was the type of spring to use. we considered a regular, helical compression spring. This is the most common type of spring and provides a high level of reliability and durability. Additionally, we considered a conical spring, to account for both the limited space of the braking system and the potential of buckling. Ultimately, we decided to go with the combined spring and friction system, as well as the conical spring.

3.1.3 Why we Chose this Option

We opted for a combination of friction-based and spring-based braking to achieve a balanced, efficient, and space-conscious system while maintaining rider safety and minimizing structural deflection. Initially, relying solely on a spring brake resulted in excessive compression distances, which posed challenges in terms of system alignment and practicality. By distributing the braking forces, 65% through friction and 35% through a spring, we significantly reduced the required stopping distance while ensuring controlled deceleration. A conical spring was chosen over a traditional helical spring due to its superior ability to progressively stiffen during compression, providing smoother braking force application and reducing abrupt force transitions. Additionally, conical springs offer better lateral stability and a compact stacked form when fully compressed, making them ideal for space-limited designs in roller coaster braking systems. This combination ensures an optimized, reliable braking mechanism that meets performance requirements while maintaining durability and rider comfort.

3.1.4 Braking System - Detailed Calculations and Figures

The evaluation of the spring braking system began by determining a comfortable range of forces that will be experienced by the riders when they stop. Most humans can tolerate 4-6 G's of force. Since this ride is mostly meant for children, we will aim to hit a maximum of 3 G's to ensure the child's comfort and safety. The roller coaster cars will typically end the track with a velocity of about 10 m/s. To find the necessary braking forces and spring factors, we used conservation of energy. The car experiences a kinetic energy of 38 kJ as it reaches the braking point. The kinetic energy was found using equation 11.

$$KE = \frac{1}{2}mv^2 \tag{11}$$

This means that the entirety of the braking system must dissipate a total of 95 kJ of energy. Since we decided that the spring will take up 35% of braking responsibility and the friction system will take up 65% the spring system now must only dissipate 33.25 kJ of energy and the friction system the remaining 61.75 kJ. To find the stopping distance required by the friction system, we first must find total force experienced by the riders at the point of stopping. We do this by using equation [12], assuming a reasonable coefficient of friction of 0.4. This results in a total force of 7,455.6 N.

$$F_{friction} = m * a * \mu \tag{12}$$

Then, we can use equation 13 to find the friction system's required stopping distance. This comes out to 8.28 meters.

$$KE_{friction} = F_{friction} * d_{friction} \tag{13}$$

Next, since the spring system must dissipate the remaining 33.25 kJ, we can use equation 14.

$$E = \frac{1}{2}kx^2 \tag{14}$$

If we limit the compression distance to 0.5 meters, the k value comes out to be 266,000 N/m. Applying the safety factor of 2, the new k value is 532,000 N/m. We decided to make the spring system two spring in parallel to reduce the required k value. This leaves each spring with a k value of 266,000 N/m each experiencing a compression distance of 0.5 m. To ensure this fits within the desired constraint of the car only experiencing 3 G's of force, we must calculate the total acceleration of the entire braking system. We can use equation 15 to do so.

$$F_{rider} = ma_{dec} \tag{15}$$

3 G's of force means the car is decelrating at a rate of 29.43 m/s^2. The braking distance is governed by equation 16.

$$v_f^2 = v_i^2 + 2ad \tag{16}$$

Since our final velocity is 0, our initial velocity is 10 m/s and the deceleration is 29.43 m/s^2, the minimum braking distance comes out to be 1.7 m. Since our system takes up a distance of 8.28 meters for friction and 0.5 meters for the spring, we are well within the desired range. The overall braking distance of the system required is 8.78 meters.

To choose the material of the spring necessary, we will have to account for the fatigue that the spring experiences, being loaded multiple times a day. A high carbon spring steel such as SAE 9255 or AISI 1074/1075 would ensure high yield strength and excellent fatigue resistance with a tensile strength of about 1100 MPa. For the friction system, something like a copper-based brake pad would be ideal because it would offer high durability and heat dissipation. The team suggests that the spring and brake pads be changed every year to ensure proper functionality. The fatigue is accounted for in the safety factor.

Figure 8 shows a sketch of the conical springs that are used in the design. There will be two of these in the assembly, each with an effective k value of 266,000 N/m.

Figure 8: Conical Spring with effective k value 266,000 N/m.

Figure 14 shows an isometric view of the spring braking assembly. The car on the left comes to a stop by hitting the first plate in the series, which is allowed to move as the spring compresses. The right-most plate is stationary and attacked to the springs.

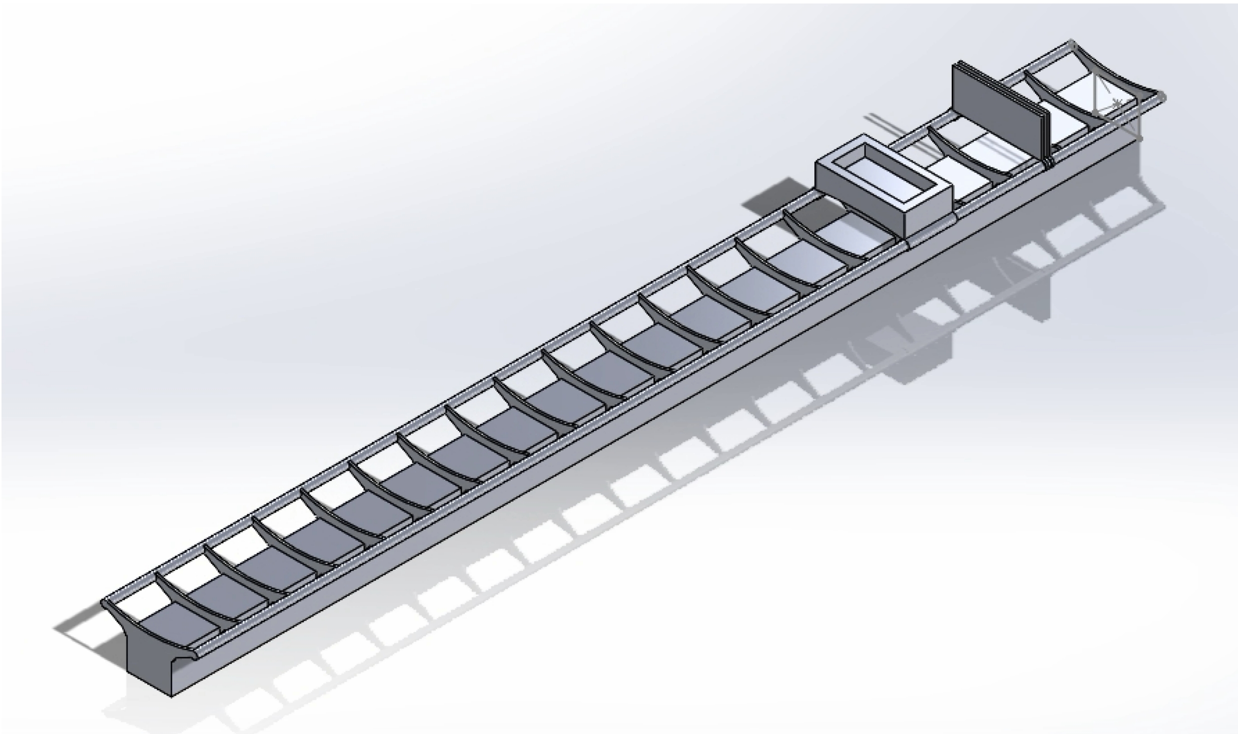


Figure 9: Isometric view of the spring assembly with 2 conical springs.

As you can see in figure 10, both springs are in parallel next to each other, attached to the stationary plate.

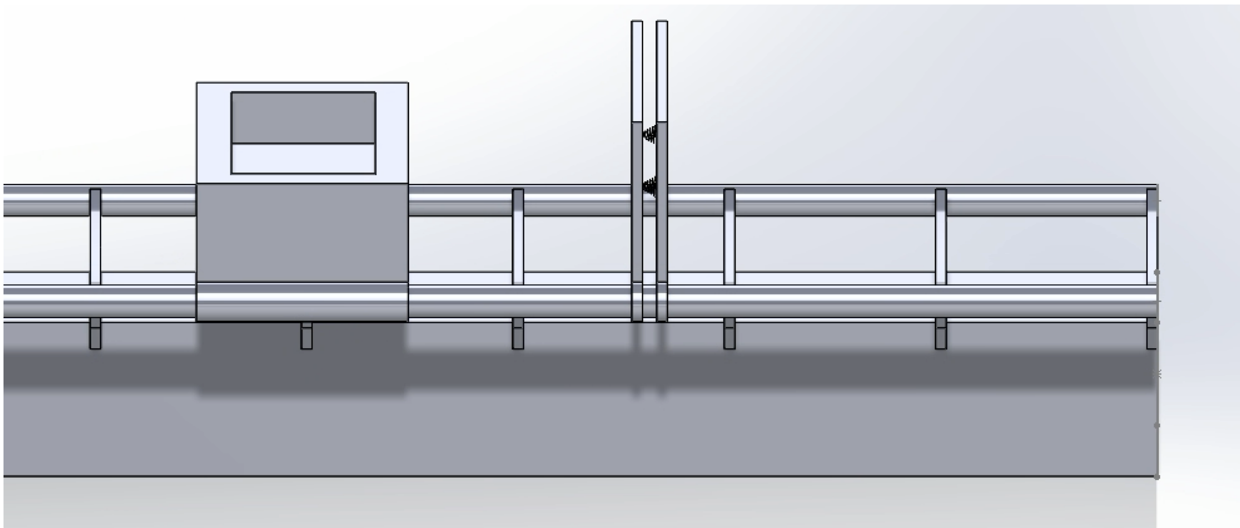


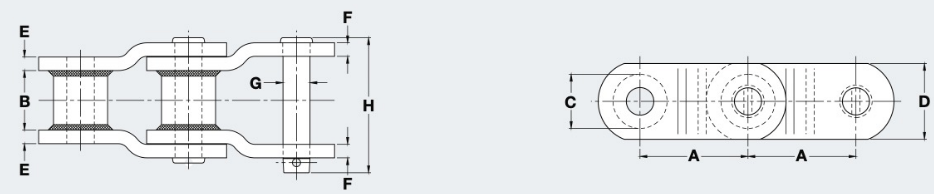
Figure 10: View 2 of the spring assembly.

3.2 Chain System

3.2.1 Requirements

The chain is a fundamental component of the roller coaster's lifting mechanism, designed to engage the roller coaster cart, effectively pulling it uphill against gravity. The WS82TP welded steel type lift chain, shown in figure 8, is ideally suited for this function, boasting a pitch of 3.075 inches, a barrel diameter of 1.219 inches, and a pin diameter of 0.563 inches. Its rated working load is 4,400 lbs (approximately 19,580 N), with an average ultimate tensile strength of 37,500 lbs (166.8 kN), making it structurally capable of handling the applied load of 19,718 N (based on a cart and passenger mass of 2,010 kg) with a conservative safety factor above 2.5. This chain component will be sourced from Renold, an industrial products company based in the United Kingdom.

Theme Park Chain



Welded Steel Type Lift Chain

Dimensions are in inches unless otherwise indicated.

Chain No.	Pitch	Inner Width Max	Barrel Diam Max	Plate Height Max	Plate Thick Max	Pin Diam Max	Pin Length Max	Average Ultimate Strength	Rated Working Load	Weight
	A	B	C	D	E/F	G	H	Lbs	Lbs	Lbs/Ft
WS82TP*	3.075	1.750	1.219	1.250	0.250	0.563	3.219	37,500	4,400	5.60

Figure 11: Welded Steel Type Lift Chain

3.2.2 Choices and Options

When selecting the chain for the roller coaster's lift mechanism, several types and configurations were evaluated. ANSI standard roller chains were considered due to their standardization and availability but were ruled out due to limited load capacity for amusement-grade applications. Welded steel chains, specifically lift-type chains, were considered superior due to their ability to withstand high tension forces and their heavy-duty construction. Regarding pitch size, we considered options between 2.5 inches and 3.075 inches. A larger pitch increases the load-bearing capability but requires larger sprockets. The 3.075-inch pitch was particularly attractive due to its compatibility with heavy-duty lifting sprockets and increased tooth engagement area. Chain material options included plain carbon steel (low cost but prone to wear), stainless steel (corrosion-resistant but weaker), and alloy steels such as 4140 and 8620, which can be heat-treated for superior strength and wear resistance. Additional coatings such as zinc plating or galvanization were considered essential for corrosion resistance in outdoor environments.

3.2.3 Why we Chose this Option

We selected the WS82TP welded steel lift chain with a 3.075-inch pitch due to its high strength, durability, and compatibility with industrial roller coaster drive systems. The chain has a rated working load of approximately 4,400 lbs (19,570 N) and an ultimate tensile strength of 37,500 lbs (167 kN), which provides a safety factor greater than 3 for our design load of 19,718 N. The chain's 1.219-inch barrel diameter and 0.25-inch thick side plates offer excellent load distribution and durability. Heat-treated alloy steel construction, specifically case-hardened surfaces, ensures wear resistance during prolonged use. Zinc plating further enhances corrosion resistance in humid outdoor conditions. The selected chain pitch fits precisely with a 12-tooth sprocket, ensuring smooth engagement with a wrap angle over 120°, which maximizes tooth contact and minimizes slippage. This setup provides a robust and long-lasting solution for high-load, high-cycle applications like roller coaster lifts.

3.2.4 Chain System - Detailed calculations and figures

There are a number of forces that the chain must withstand to lift the roller coast cart and passsengers up the incline. The primarily force being the basic tensile force. In this design, the total system mass is 2,010 kg, which results in a force of about 19,718 N due to gravity. This is the minimum load the chain will carry in steady state and serves as a starting point for further strength and fatigue analysis. The equation for tensile force is shown below:

$$F_{chain} = (m)(g) \tag{17}$$

$$F_{chain} = (2010kg)(9.81m/s^2) = 19,718.1\ N \tag{18}$$

where m is the total mass of the roller coaster and passengers in (kg), and g is acceleration due to gravity.

To ensure safety, the chain must have an ultimate tensile strength exceeding the maximum working force by the previously stated factor of safety of 2.5. This accounts for dynamic loads, vibrations, misalignment, and material variability. This can be calculated using the following equation:

$$UTS_{required} = F_{chain} * SF \tag{19}$$

$$UTS_{required} = (19,718.1N)(2.5) = 49,295.25\,N = 50\,kN \tag{20}$$

The bearing stress at the contact surface between the chain pin and the inner link plate must also be considered as a part of the calculations. Too much stress here can cause plastic deformation or wear, leading to chain elongation or failure. It's crucial to size the pin diameter d and plate thickness t properly so this value stays well below the material's yield strength. The following equation help determine the bearing stress:

$$\sigma_{bearing} = \frac{F}{(d)(t)} \tag{21}$$

$$\sigma_{bearing} = \frac{(19,718.1)}{(0.008)(0.005)} = 493\,MPa \tag{22}$$

The value of 493 MPa is below the yield strength of the link plate material (4340 steel) yield strength of 850 MPa when it is heat treated, so it is acceptable.

Shear stress in the chain pin can also be considered. The result must be significantly lower than the shear yield strength of the chosen 4340 alloy steel. This ensures the pin won't fail during loading cycles, particularly during sudden loading or acceleration. The equation for shear stress in the pin is shown below:

$$\tau_{pin} = \frac{2F}{\pi d^2} \tag{23}$$

$$\tau_{pin} = \frac{2(10,718.1)}{\pi(0.008)^2} = 196\,MPa \tag{24}$$

Because the shear yield strength of 4340 steel is typically about 600-700 MPa, the shear stress in the chain pin of 196 MPa would be considered safe.

Some further considerations in the designing of the chain include that the contact zone between the chain and the gear teeth and chain barrel must be hardened to resist localized deformation or wear. The chain links are fabricated from heat-treated alloy steel, further enhanced by case hardening to optimize surface durability while maintaining a tough, fatigue-resistant core. Each pin, barrel, and plate interface is optimized for high-load transfer with reduced wear.

Lubrication is essential to reduce the coefficient of friction ($\mu = 0.1$) between the pin and barrel, minimizing power loss and wear. Regular lubrication, along with proper tensioning via an idler sprocket system, ensures minimal slack, preventing vibration and disengagement during ascent. At 5.60 lbs/ft, the chain adds substantial dead weight and requires careful tensioning calculation, factoring in both vertical lift length and frictional losses.

3.2.5 Chain Dog Design Requirements

The chain dog, or lift dog, is the mechanical interface between the roller coaster car and the lift chain. It must reliably engage the WS82TP chain to transfer pulling force from the chain to the car during ascent. To ensure precise and secure engagement, the dog is dimensioned to match the WS82TP chain's geometry — specifically, its barrel diameter of 1.219 inches (30.96 mm) and pitch of 3.075 inches (78.1 mm).

The hooked tip of the dog is formed as a semi-cylindrical cradle with an internal radius of approximately 16 mm, slightly oversized to allow for thermal expansion and manufacturing tolerance while preventing slip during engagement. The mouth of the dog has an opening width of 34 mm, allowing it to slide laterally onto the chain barrel with a clearance of 1.5 mm on either side. The cradle depth is set to 22 mm, allowing the barrel to nest securely below the centerline of the hook, reducing vertical disengagement risk under dynamic load.

The dog body is machined from 4140 steel, quenched and tempered to a yield strength of at least 950 MPa and surface-hardened to HRC 40–45. It has a total body thickness of 25 mm, and is reinforced with lateral ribs to resist torsional deflection. The hook is integrated into a pivoting arm, 150 mm in total length, mounted on a high-strength alloy steel pin.

3.2.6 Choices and Options

In designing the chain dog that connects the roller coaster cart to the lift chain, we analyzed both fixed and pivoting mechanisms. Fixed dogs are structurally simple but require precise alignment with the moving chain, making them vulnerable to jamming under load. Pivoting dogs offer greater alignment flexibility, which reduces wear and the likelihood of mechanical failure during engagement. Dog shapes included hook-style and flat-tab models, with the hook design offering more reliable chain engagement. Materials considered were aluminum (lightweight but not strong enough), stainless steel (corrosion-resistant but less strong), and hardened alloy steel (high strength and fatigue resistance). Various mounting methods were examined, including bolted and pinned designs. Bolted mounts allow for easy maintenance but must be locked securely to resist vibration. Pinned designs offer robustness but are more difficult to service. Engagement geometry was also considered, focusing on matching the dog to the chain's barrel and plate dimensions to distribute load evenly.

3.2.7 Why we Chose this Option

We ultimately chose a pivoting hook-style chain dog made from 4140 alloy steel, heat-treated to achieve a yield strength exceeding 950 MPa. This material choice offers excellent fatigue life and shock resistance, crucial in roller coaster environments. The cradle of the dog matches the chain's 1.219-inch barrel diameter for secure and stable engagement. The pivot is supported by a 16 mm diameter steel pin with integrated bronze bushings to reduce friction and wear. The dog's swing is limited to $\pm 5^\circ$ to allow alignment flexibility without risking unintended disengagement. Its plate thickness is 0.25 inches to align with the chain's side plates, ensuring full load transfer. The dog is bolted to the cart chassis using grade 8 fasteners, with thread-locking compounds and lock washers to resist vibration-induced loosening. This design balances strength, serviceability, and operational reliability under the high-frequency cycling of the roller coaster lift system.

3.2.8 Chain Dog - Detailed Calculations and Figures

The connection pin between the chain dog and the car's undercarriage passes through dual lug plates and is subject to both shear and bending. The chain dog transmits the lifting force from the chain to the roller coaster car via a steel pin connection. The shear force on the pin can be analyzed carefully using the following equation:

$$A_{shear} = \frac{(2)(3.1415)(d)^2}{4} \tag{25}$$

$$\tau = \frac{F}{A_{shear}} \tag{26}$$

$$A_{shear} = \frac{(2)(3.1415)(0.016)^2}{4} = 4.02 * 10^-4m^2 \tag{27}$$

$$\tau = \frac{19,718}{4.02 * 10^-4m^2} = 490\ MPa \tag{28}$$

4340 alloy steel (heat-treated) has a shear strength of 650 MPa, so this is well within limits with a factor of safety over 5.

Another factor that must be considered in the analysis of this component is when the chain dog is pulled by the chain, its mounting holes and surrounding slot structure are subject to bearing (contact) stress. This bearing stress can be calculated using the following equation:

$$\sigma_{bearing} = \frac{F}{(d)(t)} \tag{29}$$

$$\sigma_{bearing} = \frac{19,718}{(0.016)(0.008)} = 154\ MPa \tag{30}$$

This stress is below the yield strength of hardened steel (850 MPa), indicating the chain dog mount will not plastically deform under load.

To ensure safety, a 16 mm diameter pin is used, made from 4340 steel, surface hardened and mounted in oil-impregnated bronze bushings for wear resistance and smooth rotation.

The dog engages the chain at the base of the lift hill and disengages via a cam ramp mechanism at the crest. To accommodate vertical misalignment or slack in the chain, the dog is designed with $\pm 5^{\circ}$ angular float and 5 mm vertical compliance via a slot-and-spring system. The entire dog assembly weighs 4 kg and includes wear plates at contact zones, replaceable during maintenance.

3.2.9 Anti-Rollback Device Design Requirements

The anti-roll-back system is a fail-safe ratcheting mechanism located under the roller coaster car to prevent reverse motion if the chain fails during ascent. It consists of a steel ratchet rail – commonly referred to as the "click-clack" rail – mounted along the centerline of the track and a spring-loaded pawl on the underside of each car.

3.2.10 Choices and Options

For the anti-roll-back mechanism, multiple configurations were considered to prevent the cart from sliding backward in the event of chain failure. Tooth profiles for the ratchet rail were evaluated, including rectangular, triangular, and sawtooth shapes. The sawtooth profile was chosen for its superior unidirectional locking capability, allowing forward movement while effectively halting backward motion. Pawl designs were assessed in terms of tip shape—flat tips provide solid contact, but curved tips reduce impact shock during engagement. Materials considered included tool steel, 4140 alloy steel, and stainless steel. Tool and alloy steels offer the necessary impact resistance and strength. The spring mechanism for biasing the pawl was also evaluated, with torsion springs favored over coil springs for consistent force application and space efficiency. Finally, additional features like polyurethane bumpers or dampers were considered to reduce engagement noise and prolong component life.

3.2.11 Why we Chose this Option

The final anti-roll-back system utilizes a sawtooth-profile ratchet rail with 2.5-inch spacing between teeth and a 15° engagement angle. This profile ensures that forward motion is unimpeded while providing immediate mechanical locking if reverse movement occurs. The pawl is constructed from 4340 steel, heat-treated to achieve a high surface hardness and impact resistance. The pawl has a curved tip, which helps it smoothly ride over the sawtooth profile and engage only when needed. It is 0.75 inches thick and 3 inches in length, capable of withstanding impact forces up to 23 kN, as determined by dynamic analysis of a loaded cart under rollback conditions. A torsion spring biases the pawl downward, ensuring reliable engagement at all times, while polyurethane dampers cushion the impact and reduce noise. This system is mechanically simple, fail-safe, and designed to last under the repetitive stress conditions of roller coaster operations.

Ratchet Rail: The rail is made of 4140 low-alloy steel, with teeth cut at a 15° approach angle and spaced every 2.5 inches (63.5 mm). The tooth profile is sawtooth-shaped, with a height of 20 mm, base thickness of 15 mm, and tip thickness of 4 mm. Each tooth is carburized to HRC 58–62 to resist high contact forces and impact spalling. The rail is bolted to the track via vibration-resistant hardware spaced every 500 mm, and is aligned such that the tooth centerline is 30 mm below the car's underside during lift.

Pawl Mechanism: The pawl is a hardened steel lever with a curved tip that slides over the ratchet teeth during forward motion and locks against them under backward motion. The contact tip has a radius of 6 mm and is angled at 15° to match the rail profile. The total length of the pawl is 180 mm, with a width of 30 mm and thickness of 12 mm. It pivots on a 20 mm diameter stainless steel shaft mounted in hardened steel bushings.

3.2.12 Anti-Rollback - Detailed Calculations and Figures

A torsion spring biases the pawl downward against the rail. The pawl is designed to tolerate up to 23 kN of impact force in the event of sudden rollback. Using a conservative impact deceleration assumption over 100 mm, and a cart mass of 2,010 kg:

$$F = \frac{mv^2}{2s} = \frac{2010(1.5)^2}{(2)(0.1)} = 22,650\,N \tag{31}$$

To handle this force without yielding, the pawl's cross section must resist bending. Taking a 100 mm arm length:

$$M = (F)(L) = (22,650)(0.1) = 2,265\,Nm \tag{32}$$

The bending stress in a rectangular cross-section:

$$\sigma_{bending} = \frac{6M}{bh^2} = \frac{(6)(2265)}{(0.03)(0.012)^2} = 1256\,MPa \tag{33}$$

Using 4340 steel ($yield\ strength > 1,300\ MPa$) ensures that the design remains within yield limits.

The engagement mechanism includes a shock-absorbing polyurethane bumper mounted just above the pawl to dampen rebound and reduce acoustic shock. To avoid false locking or bounce-back, the ratchet angle is under 15° and the pawl includes a detent notch to hold it in the downward-locked position under reverse load.

The entire ARB system is designed to allow inspection access every 2 meters, and wear indicators are laser-etched on each tooth. In total, the system adds 10 kg per car and is mounted using vibration-isolated bushings to prevent metal-on-metal resonance.

3.2.13 System Level Integration

The chain, chain dog, and ARB must operate in coordinated harmony. The chain path runs in a channel or trough up the lift hill, supported by guide rollers or wear plates to ensure alignment and minimize lateral sway. A tensioning mechanism—often an idler sprocket mounted on a counterweighted or hydraulic arm—is required to accommodate thermal expansion and chain stretch. For the WS82TP chain, total weight becomes significant: over a 60-meter lift hill (197 feet), the chain mass is:

$$(197\,FT) * (5.6\,lbs/ft) = 1,103\,lbs\ = 4,900\,N \tag{34}$$

This mass adds tension on the return side and must be factored into motor torque calculations.

3.2.14 Material and Safety Consideration

Safety and fatigue resistance are paramount, given the dynamic nature of rollercoaster operation. A safety factor of at least 5 accounts for acceleration, vibrations, and unexpected stresses. The chain must endure high-cycle fatigue, resisting wear over millions of loading cycles, which is achieved through material selection, heat treatment, and proper lubrication. Potential failure modes include chain stretch due to wear, monitored through regular tension checks, and tooth shear, mitigated by hardened gear teeth.

Material selection is guided by mechanical strength, fatigue life, corrosion resistance, and wear properties. Safety is governed by ASTM F2291 and EN 13814 standards for amusement rides. These recommend a minimum safety factor of 5 for critical components under direct passenger load, and minimum fatigue life of 10 million cycles. Redundant systems and daily visual inspections are mandatory.

3.2.15 Maintenance and Manufacturing

Manufacturing tolerances on the chain must be tight—pitch deviation must be 0.5% to avoid misalignment. Barrels are machined then welded, with pins press-fit and hardened. Chain sections use master links or removable pins for field service. Chain dogs are forged or CNC-machined, with replaceable bushings at pivot joints.

Daily maintenance includes visual inspection for wear, cracks, and lubrication status. Weekly tension checks ensure elongation does not exceed 1.5% of nominal length. The ARB pawls are checked for free motion and tip wear. All high-load areas are inspected using dye penetrant or magnetic particle NDT.

3.3 Output Shaft

The output shaft is a key component in the roller coaster's drivetrain, responsible for transferring high torque from the internal gear assembly to the external drive mechanism. Though its role is mechanically straightforward, the design of the output shaft must account for a range of structural, spatial, and performance considerations to ensure safe and efficient operation.

At the core of the design process is the need to withstand a maximum applied torque of 1680 N·m without yielding. This requires careful material selection, stress analysis, and geometric optimization. The shaft must remain within allowable shear limits while maintaining a radius smaller than that of the surrounding gears for proper integration. A solid circular cross-section was ultimately chosen to balance strength and simplicity in manufacturing.

In addition to torsional strength, the shaft must span a fixed distance of 1.5 meters to accommodate the layout of the internal system and provide physical separation from the drive components below the track. This length introduces potential concerns related to deflection, buckling, and bending all of which are addressed through the strategic placement of radial support bearings.

Beyond structural requirements, practical concerns such as alignment, fatigue, and corrosion resistance were also considered. While fatigue may be introduced through cyclic loading, the shaft's longevity will be managed through consistent maintenance and scheduled replacement rather than relying solely on theoretical fatigue life. The final design includes an integrated notch-based connection at each end to ensure proper engagement with the gear system.

This section details the requirements, constraints, and engineering decisions that led to the finalized output shaft design.

3.3.1 Requirements

The output shaft serves a critical role in the drivetrain system, transmitting a torque of 1680 N·m from the internal gear assembly to the car's drive gear or sprocket. Among the mechanical components, it has one of the most clearly defined functions. As such, the primary structural requirement is that it must resist shear stress due to torsional loading without yielding. The design should ensure that the maximum shear stress remains well below the material's shear yield strength, accounting for an appropriate factor of safety.

The shaft must also span a minimum length of 1.5 meters. This dimensional requirement is driven by the layout constraints of the roller coaster design, where sufficient spacing is needed to isolate the internal gear system from the drive mechanism beneath the track. The extended shaft length creates a physical buffer zone that simplifies maintenance access and prevents mechanical interference. However, such a span introduces concerns related to shaft deflection, critical buckling, and natural frequencies. To mitigate these, the shaft must be supported at multiple points using well-aligned radial bearings to control both bending and lateral displacement under load.

A further design constraint is imposed on the shaft radius, which must remain smaller than the radii of the mating gear and sprocket. This ensures that the shaft does not physically interfere with the gear teeth or rotational motion of adjacent components. While alternative coupling designs can sometimes accommodate larger shaft diameters, minimizing the shaft radius within structural limits offers improved mechanical clearance and reduces rotational inertia.

3.3.2 Choices and Options

The first choice to make in the design process is not any geometric or load-bearing considerations yet. Rather its the material selection where the process begins. The shaft can be constructed from a number of different materials with sufficient yield strength, fatigue resistance, and toughness. While we have already decide to build design with steel there are still many options. Choices include medium to high carbon steels, such as AISI 1045 or 4140, or structural steels like A36. If operating in dynamic or cyclic loading environments, fatigue life analysis is necessary.

The next major design decision involved the general geometry of the shaft, specifically the choice between a solid or hollow cross-section. Both configurations can be engineered to withstand the applied torque and bending moments, provided their geometric parameters are appropriately selected. While a hollow shaft requires a larger outer radius to match the strength of a solid shaft, it offers advantages in terms of reduced mass and potentially improved torsional stiffness-to-weight ratio. This decision is tightly coupled with economic and performance trade-offs and directly influences the subsequent step in the process.

The final design choice is the shaft radius. More than any other parameter, the selection of radius is dependent on prior material and geometric choices. Numerous radii could satisfy the strength and deflection requirements of the shaft under the applied loading conditions. While selecting a radius at the upper limit would ensure mechanical robustness, it would also lead to excess weight and material usage, resulting in an over-engineered solution. Instead, it was more appropriate to select a reasonable initial radius and verify, through stress and deflection analysis, that the design remains within allowable limits for the chosen material and loading conditions.

3.3.3 Why we Chose this Option

For the shaft, we chose to use A36 steel. This decision was driven primarily by economic considerations A36 is one of the most cost-effective steels available while still offering sufficient strength for our application. Additionally, because our design does not place significant emphasis on the strength-to-weight ratio, we prioritized strength alone. The widespread availability and common usage of A36 also enhance its feasibility and ease of procurement.

For the geometric design, we opted for a solid shaft with a radius of 2.5 cm. Output shafts are typically designed as solid or thick-walled, and we determined that, from a manufacturing and feasibility standpoint, a solid shaft would be the most straightforward choice. Although a hollow shaft with a larger radius was considered especially since our design primarily involves shear stress these configurations often resulted in extremely thin walls. For instance, a hollow shaft with a 10 cm outer radius would require a wall thickness of well under a quarter centimeter to meet the same stress requirements. Such thin walls could be vulnerable to secondary or tertiary failure modes, such as buckling or fatigue, which would compromise reliability. Thus, designing a solid shaft with a smaller radius provided a safer and more robust solution, especially since additional bearings can be incorporated as needed to address weight and support concerns.

3.3.4 Output Shafts - Detailed Calculations and Figures

Pictured below is a SolidWorks rendering of the output shaft. Units are in centimeters and the model includes the integrated connection that will slide into the gears on either end of the shaft.

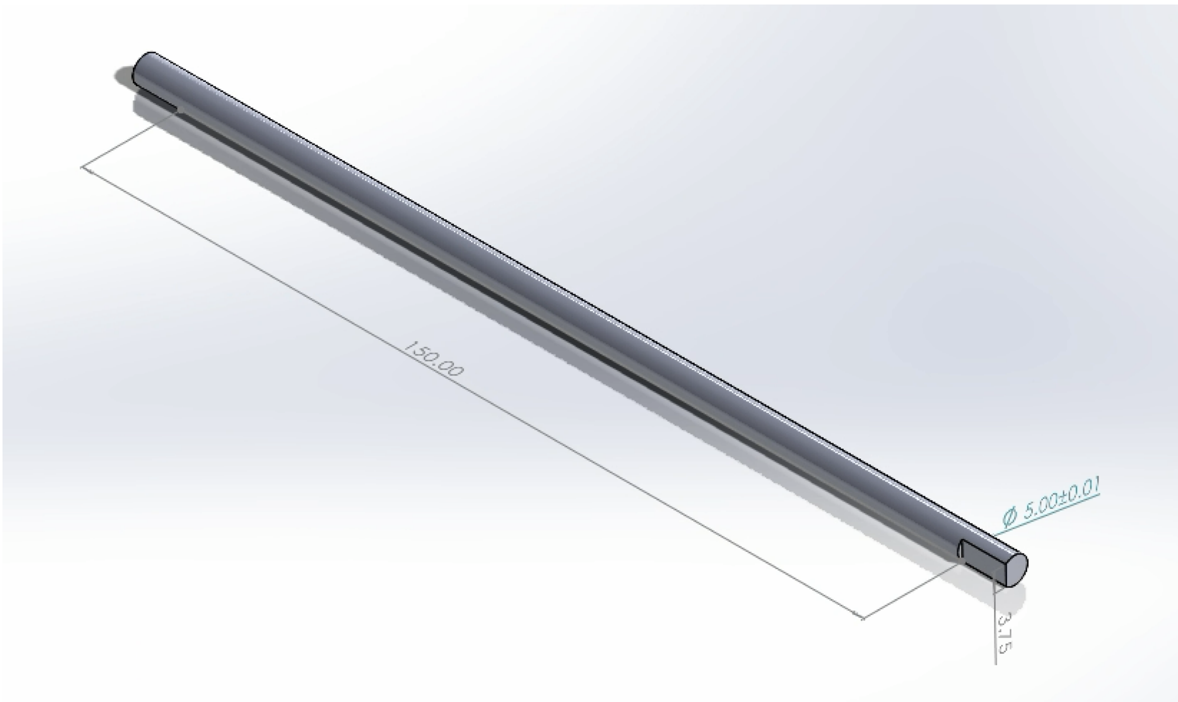


Figure 12: Isometric view of output shaft

A36 steel has a yield strength of approximately 250 MPa. However, this value applies to normal (tensile) yield strength, and since we are designing for shear, it must be converted accordingly. To obtain the shear yield strength, we divide the tensile yield strength by the square root of three:

$$\tau_y = \frac{\theta_y}{\sqrt{3}} \rightarrow \frac{250(MPa)}{\sqrt{3}} \rightarrow 144.3375(MPa) \tag{35}$$

Before we begin designing the output shaft to stay within acceptable shear limits, we must apply a safety factor to the yield strength. Because we are enforcing a weight limit on the ride and keeping the overall weight as consistent as possible, the variability in loading will be minimal. Additionally, failure of the shaft would not pose safety risks to riders but would simply stall the ride and prevent its operation. Given these considerations, a relatively low safety factor of 2 is reasonable for the output shaft:

$$\tau_{max,allowable} = \frac{\tau_y}{S.F.} \rightarrow \frac{144.3375(MPa)}{2} \rightarrow 72.168 \approx 72(MPa) \tag{36}$$

We now know that the shaft must not exceed a shear stress of 72 MPa; this value has been rounded down slightly for an added margin of safety. With this limit in mind, we can begin evaluating the performance of a steel rod using the following equation:

$$\tau = \frac{Tr}{J} \tag{37}$$

We will test a solid shaft with a radius of 25 mm. First, we calculate the polar moment of inertia, J:

$$J = \frac{\pi}{32}D^4 \rightarrow \frac{\pi}{2}r^4 \rightarrow (1.57)(25)^4 \rightarrow 613,592.36\ (mm^4) \tag{38}$$

Next, we substitute the polar moment of inertia, the radius of 25 mm, and the applied torque of 1,680,000 N·m into the shear stress equation. First, we convert the torque into consistent units:

$$T = 1680\ N \cdot m \times \frac{1000\ mm}{1\ m} \rightarrow 1,680,000\ (N \cdot mm) \tag{39}$$

Now we substitute all values into the shear stress formula:

$$\tau_{torque} = \frac{Tr}{J} \rightarrow \frac{1,680,000\ (N \cdot mm) \times 25\ (mm)}{613,592.36\ (mm^4)} \rightarrow 68.45\ (MPa) \tag{40}$$

The shear stress experienced by a solid shaft with a 25 mm radius carrying 1680 N·m of torque is 68.45 MPa. We compare this with the allowable shear strength of A36 steel:

$$\tau_{max,allowable} \approx 72\ (MPa) \geq 68.45\ (MPa) = \tau_{torque} \tag{41}$$

Since the induced shear stress is less than the maximum allowable shear stress for A36 steel, we conclude that a solid shaft with a 25 mm (2.5 cm) radius is adequately designed for this application.

There are a few additional tests we need to run to ensure that the shaft is suitable. One of these is torsional deflection, or the angle of twist:

$$\theta = \frac{TL}{JG} \tag{42}$$

We know all variables except the shear modulus, G, which is a standardized property of A36 steel. For A36, G = 79 GPa. First, we convert G and L into consistent units:

$$G = 79\ (\text{GPa}) \times \frac{1{,}000\ \text{N} \cdot \text{mm}}{1\ \text{GPa}} \rightarrow 79{,}000\ \text{N/mm}^2 \tag{43}$$

$$L = 1.5\ (\text{m}) \times \frac{1{,}000\ \text{mm}}{1\ \text{m}} \rightarrow 1{,}500\ \text{mm} \tag{44}$$

Now we can plug these values into the equation for torsional deflection:

$$\theta = \frac{TL}{JG} \rightarrow \frac{[1{,}680{,}000\ (\text{N} \cdot \text{mm})] \times [1{,}500\ (\text{mm})]}{[613{,}592\ (\text{mm}^4)] \times [79{,}000\ (\text{N/mm}^2)]} \rightarrow 0.052\ \text{radians} \tag{45}$$

This is a small deflection, only about three degrees, so we can conclude that the shaft performs adequately in terms of torsional deflection.

Other potential failure modes, such as bending and buckling, were also considered. However, the shaft will be supported by multiple bearings positioned, significantly reducing any bending moments and effectively eliminating the risk of buckling. As a result, these are not primary concerns in the current design.

Fatigue was also taken into account, given the possibility of cyclic loading during operation. Instead of relying solely on theoretical fatigue life calculations, our approach emphasizes regular maintenance and timely replacement of the shaft to ensure continued reliability over time.

One of the primary concerns with the shaft, aside from its strength, is the connection to the gears. We chose to build the connection directly into the shaft to simplify the design process and ensure proper alignment. The image below shows the integrated connection.

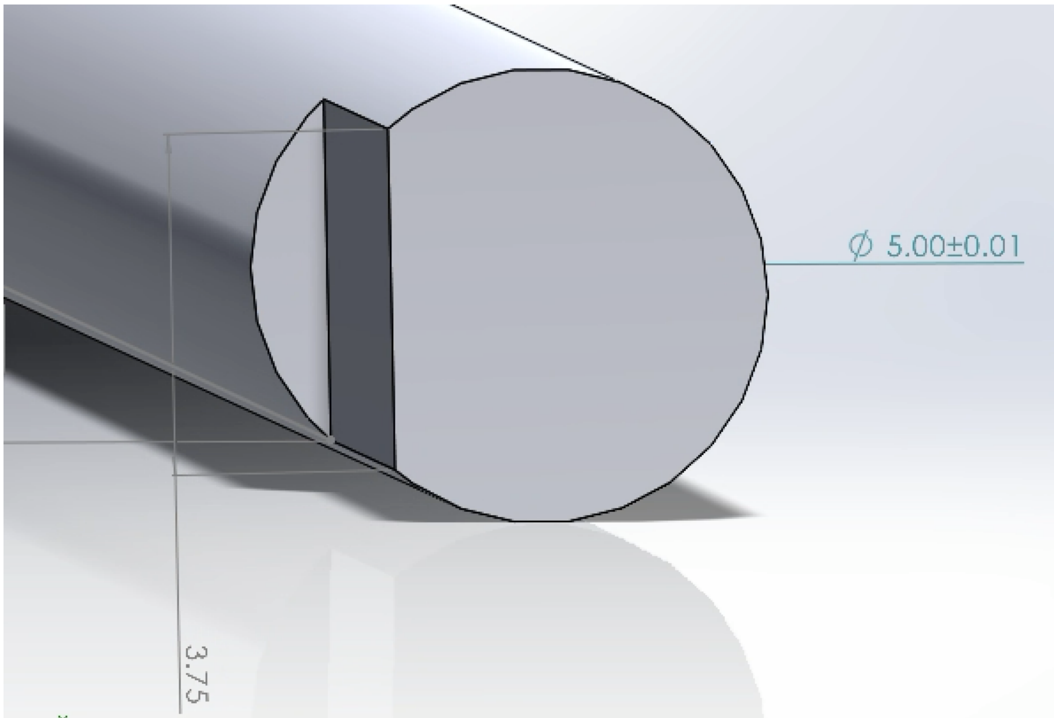


Figure 13: Shaft notched to create connection

The connection is a simple extension of the shaft—an additional 8.38 cm in length. The shaft is notched so that the gears can either drive or be driven by the shaft. The notch length is three-quarters of the shaft diameter, making it 3.75 cm. This design allows the shaft to transfer torque effectively without compromising the strength of the connection.

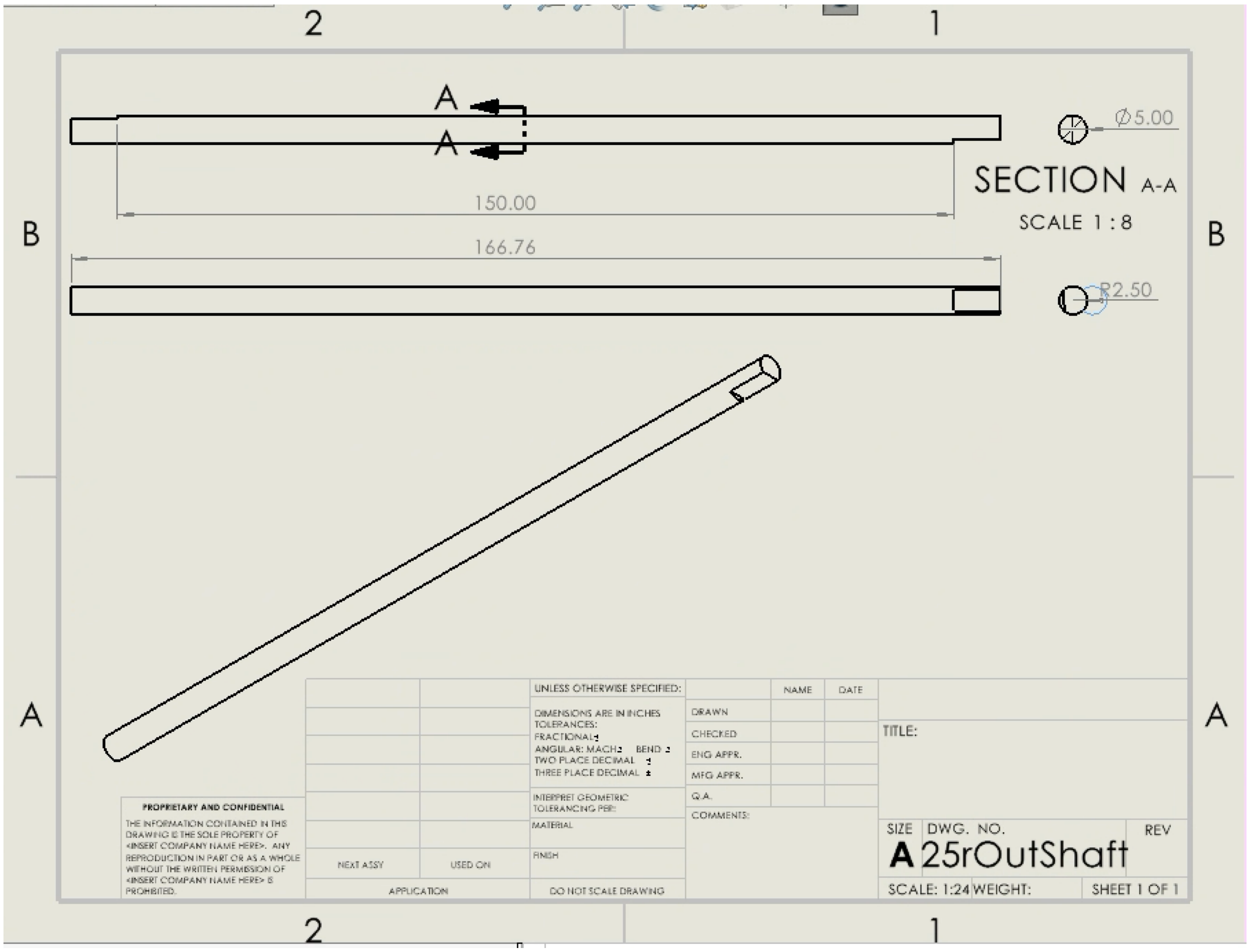


Figure 14: Shaft Drawing

3.4 Bull Wheel Sprocket

The final component of the rollercoaster that we focus on is the bull wheel sprocket. This is a sproket that rotates with the resultant torque from the internal mechanical advantage system and pulls the chain that lifts the carts up the coaster's initial incline. It serves as the critical connection between the internal gear mechanism and the actual motion of lifting the cart, making its functionality and safety essential to the overall system.

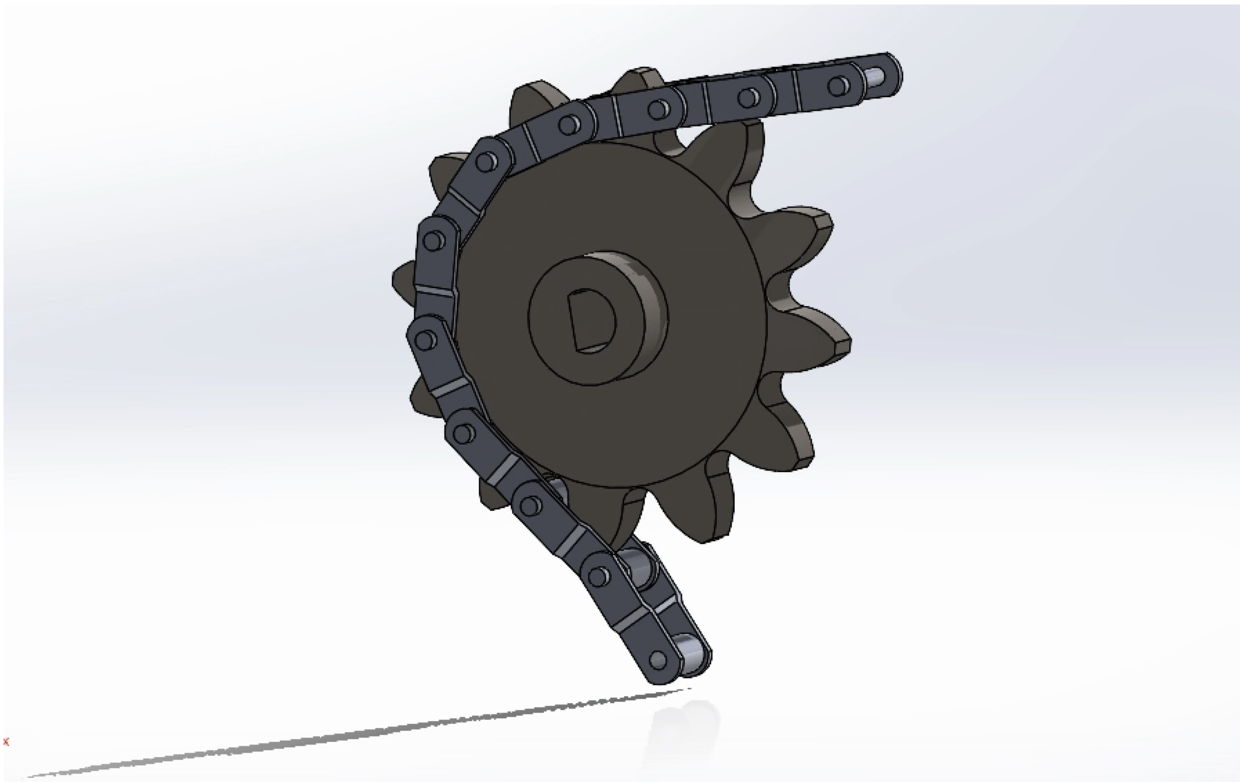


Figure 15: Sproket and Chain CAD Image

3.4.1 Requirements

There are several key considerations when designing the bull wheel sprocket. First, the gear must be strong enough to withstand the forces and stresses required to transmit torque into a pulling force on the chain. When the chain is in progress of lifting the coaster the forces on the sprocket will stay pretty constant. There will be an increased force in the beginning when the carts latch onto the chain. The coaster cars need to accelerate in the beginning to get up to speed. This acceleration force is accounted for by using a cart and people weight of 2000 kg when the carts won't be more than 1900 kg. The sprockets width, radius and material selection are all factors that can be adjusted to meet strength needs. This gear will experience 19,620 Newtons of force while lifting the cart and people.

Another critical design aspect is the sprocket's compatibility with the chain. This includes selecting an appropriate pitch diameter, number of teeth, chain wrap angle and matching pitch so the chain and sprocket can have proper connection. The sprocket teeth will experience concentrated stresses, especially at the points of contact with the chain. These stresses should be analyzed using appropriate safety factors to ensure the gear can perform reliably under repeated loading. Fatigue resistance is also an important factor, particularly since the gear will undergo continuous cycles of loading and unloading. Finally, the sprocket must have a shaft connection point, e.g keyway, that is supported.

3.4.2 Choices and Options

Initially, our team looked to outsource the sprocket for the roller coaster. Sprocket and chains are the most popular way for carts to be raised up the roller coaster. However, after much searching a sprocket that matched the dimensions we need couldn't be found. The decision was made to manufacture out own sprockets so strength and measurements could be controlled in house. The decision to make the sprocket also allow us to match the dimensions of the chain we already chose to buy from another company. When designing the sprocket many of our choices were confined to the chain dimensions and minimum diameter of output sprocket. The major decision being made was the material we would make the component out of. The decision to use A36 Steel for the shaft was made before hand and it seemed a good idea to interface the same metals together and start there.

3.4.3 Why we chose this option

Many reason A36 Steel is great is depicted in the Output Shaft section. A36 Steel is a cheap low carbon steel. A36 steel is great for a sprocket because of its ease in being welded, machined and formed. This steel is different than most as it is designated by material properties rather than chemical composition. As a result, A36 Steel must meet material property grades like yield strength. A36 Steel has an Ultimate strength of 400 - 550 MPa and yield tensile strength of 250 MPa making it a great option for lifting carts via chain.

3.4.4 Bull Wheel Sprocket - Detailed calculations and figures

It is important to confirm that the sprocket can hold the weight on the chain without breaking. To do this FEA analysis was performed on the sprocket becasue specific calculation would be too complicated. Weighting a sproket is complicated as it is distributed differently across all the teeth. An analysis with the cahin was attempted but the program would fail. A simplified loading method was taken to try and replicate what the actual sprocket would feel. The methodology was force would be applied to the upper teeth and slots more and lessened as the imaginary chain would travel down the sprocket. The total force would be 40,000 N which is a safety factor slightly above 2. The recommended safety factor for components like this being between 2 and 3.5.

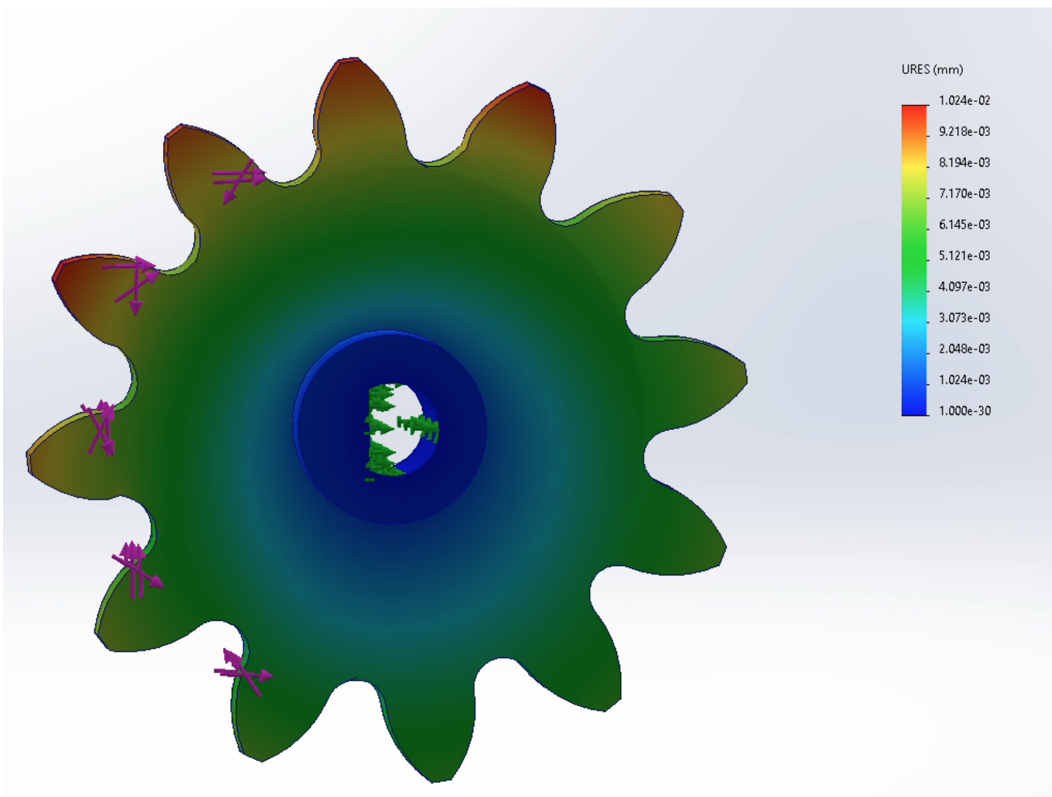


Figure 16: Displacement of Sprocket under forces

The image directly above is a displacement analysis. Though it predicts there may be slight displacement this is nothing to worry about as there is some ductility to A36. Looking at the future analysis' will show this sprocket can withstand the load.

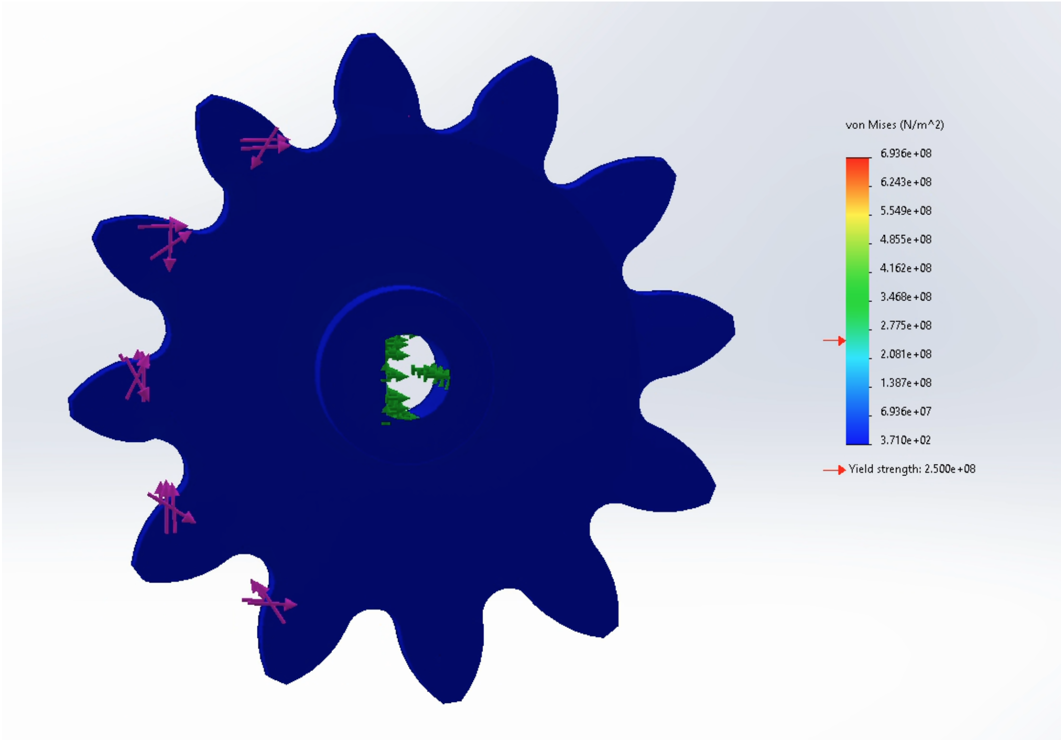


Figure 17: Von Mises Stresses in Sprocket under forces

It can be seen that there is minimal stress within the sprocket due to the forces on the teeth. This means the sprocket should not yield when it is loaded. Due to low stresses, fatigue should not be much of an issue and this sprocket will have a long lifetime before it needs to be replaced.

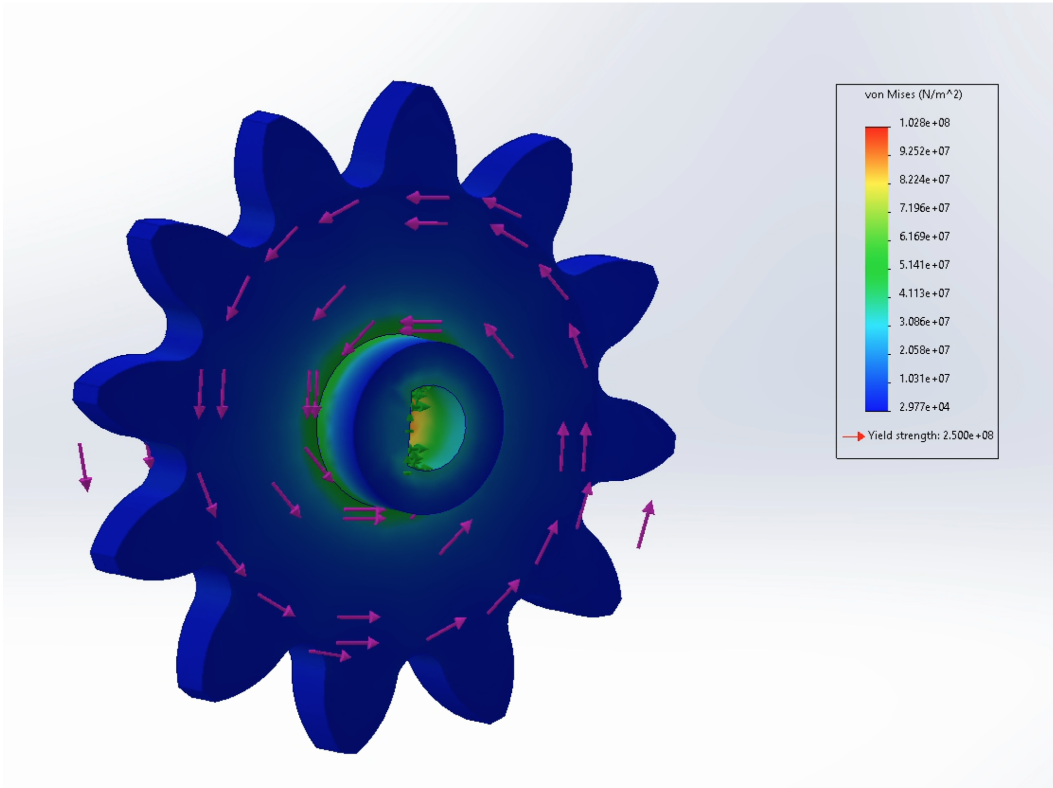


Figure 18: Torque on Sproket showing Von Mises Stresses

Another analysis performed on the sprocket was Torque. This was look at the body of the sprocket to see if there is any potential failure point. As you can see there is a bit more stress within the opening. However, when the shaft is in place this stress should lessen as the opening will be no more.

Designing Sprocket:

When designing the sprocket, the chain was the sole basis for dimensions. The chain was chosen from an outside company that manufactures roller coaster chains. The chain company uses inches for its units, however our team is using metric for parts measurements. This is fine as the parts need to fit together but not perfectly. Using the pitch (p) given in the chain table of 3.075 inches and a minimum pitch circle diameter (PD) of 28.8 cm, the number of teeth was able to be calculated. Resulting in a number with a decimal it was rounded up to 12 teeth. Equation 46 was then used again with pitch and number of teeth to find a pitch diameter of 30.177 cm.

$$PD = pcsc(\frac{180}{n}) = (3.075in)csc(\frac{180}{12}) = 30.177cm \tag{46}$$

For the chain to connect with the sprocket, the diameter of the slot the chain barrel sits in must be slightly larger. Equation 47 is used to find the diameter of the slot needed in the sprocket. Given a barrel diameter of 1.219 in, the diameter of the sprocket needs to be 1.228 in or 3.119 cm.

$$D_s = 1.005D_r + 0.003in = 1.005(1.219in) + 0.003 = 1.228in \tag{47}$$

Below in Figure 19 is a CAD drawing of the Bull Wheel Sprocket with all the dimensions. This is to allow for its replication.

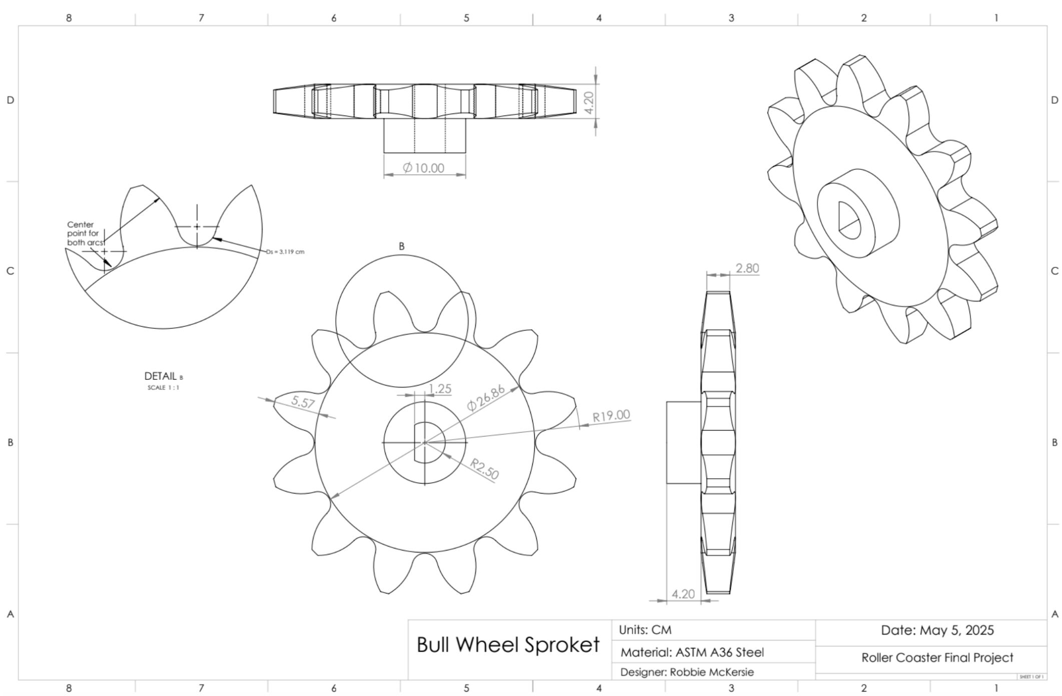


Figure 19: Sproket Drawing

4 Safety, Ethical, Environmental, Legal, and Humanitarian Concerns

Our self-sustaining, human-powered roller coaster concept inherently raises several ethical, environmental, legal, and humanitarian considerations that must be addressed to ensure the ride is not only functional and exciting but also responsible and inclusive.

4.1 Safety Factors

To ensure safety for all passengers, this rollercoaster will have an absolute maximum total weigh for passengers. No more than 800 kg will be allowed to ride the rollercoaster. The portion of the tracks next to the loading dock will have an integrated spring scale to determine if the train and passengers weigh over 2000 kg. If the passenger exceed the 800 kg limit some passengers will be asked to get off the train. This spring scale will need to be calibrated regularly to ensure accuracy.

Fatigue occurs due to repeated stress cycles, which can lead to microcracks in structural components over time. Roller coasters experience high-cycle fatigue, especially in steel structures subjected to dynamic loading. Load distribution is another key factor. Roller coasters endure varying forces, including gravitational, inertial, and impact loads. Proper material selection and structural reinforcement help mitigate stress concentrations, ensuring components can withstand repeated forces without failure. Impact forces arise during sudden deceleration, braking, and transitions between track elements. These forces must be carefully managed through shock-absorbing materials and controlled braking mechanisms to prevent excessive stress on the ride's framework. For these reasons, the design will include several safety factors as required by ASTM F2291-24, which outlines guidelines for amusement park design. Some general guidelines that were considered were as follows:

- Structural components like load bearing structures require safety factors ranging from 3.0 to 5.0.
- Fasteners and welds require safety factors between 4.0 and 6.0.

- Braking systems require safety factors between 2.0 and 4.0.
- Gear Systems, like those for chain lifts and drive mechanisms, safety factors of 2.0 to 3.5 are common to prevent excessive wear and failure.

4.2 Ethical

One of the primary ethical concerns revolves around the physical effort required by riders. While the concept adds interactivity and sustainability to the amusement park experience, it is crucial to ensure that no participant feels coerced into physical activity or excluded due to physical limitations. Therefore, our design must allow for opt-in participation, and we must provide alternative sources of power such as substituting extra people in. This ensures accessibility for individuals with disabilities, children, the elderly, or anyone who is unable or unwilling to contribute physical power.

Safety is another vital ethical responsibility. Riders generating power must not be placed in positions of risk, physically straining, slipping, or getting caught in moving components. We plan to mitigate this with carefully designed ergonomic grips, guardrails, non-slip flooring, and accessible emergency breaks. Additionally, there will be clear signage and instructions for participants, as well as emergency egress routes for all stages of the queue experience.

4.3 Environmental Consideration

Our ride design aims to significantly reduce energy consumption by eliminating traditional electric lift motors for most operating conditions. By utilizing the gravitational potential energy from human-powered elevators and tug-of-war stations, this roller coaster design minimizes reliance on grid electricity, thereby lowering carbon emissions and energy costs.

Material selection also plays a critical role in reducing environmental impact. Wherever feasible, we will use recyclable or sustainably sourced materials such as powder-coated steel for structure and recycled plastics for ergonomic handles and safety components. Moreover, we will consider corrosion-resistant components that reduce the need for replacements and minimize maintenance-related waste over the system's lifetime.

Lastly, our use of human energy contributes positively by raising awareness about sustainability in entertainment spaces, encouraging active participation in green design. We hope that our design will then influence future designs to incorporate similar values, adding to the sustainable impact our project will have.

4.4 Legal Considerations

Any amusement ride must comply with strict regulatory and safety standards. Our design must meet ASTM International standards for amusement rides and devices, most notably the ASTM F2291 – Standard Practice for Design of Amusement Rides, which establishes the criteria for the design of amusement parks. This includes ensuring structural safety, mechanical integrity, emergency systems, and thorough documentation for inspection and certification.

We will also address liability by designing redundant safety mechanisms, clear user agreements, and limitations on participation to avoid legal risks associated with injury, overexertion, or mis operation. All components—especially those involved in load transmission and lifting—will be validated through detailed engineering analysis and modeled using finite element analysis (FEA) to meet appropriate safety factors.

4.5 Humanitarian Considerations

Beyond safety and accessibility, this design fosters a sense of community, collaboration, and mutual support; riders power the ride for others, and vice versa. This creates a shared experience that mirrors humanitarian values of collective benefit and effort.

To further ensure inclusivity, the ride will incorporate multilingual instructions and signage, intuitive visual interfaces, and staff trained to accommodate a diverse range of guests. In low-rider situations or off-peak hours, a hired staff will be available to support customers in enjoying the ride.

Our goal is to design an experience that is fun, energy-efficient, and aligned with modern social values,without sacrificing the core principles of fairness, inclusion, and safety.

References

[1] S. A. Keim and A. Adesman, "Injuries Associated With Amusement Rides," *Clinical Pediatrics*, vol. 58, no. 1, pp. 38–44, 2019. doi: 10.1177/0009922818812602. [Online]. Available: https://pubmed.ncbi.nlm.nih.gov/30514155/

[2] IE Park, "Rolly Roller Coaster," 2024. [Online]. Available: https://www.iepark.com/portfolio-articoli/rolly/#toggle-id-1. Accessed: May 5, 2025.

[3] ASTM International, "F2291-24: Standard Practice for Design of Amusement Rides and Devices," 2024. [Online]. Available: https://store.astm.org/f2291-24.html. Accessed: May 5, 2025.

[4] Engineering Stack Exchange, "How to calculate forces on bearings in a chain drive?" 2018. [Online]. Available: https://engineering.stackexchange.com/questions/51299/how-to-calculate-forces-on-bearings-in-a-chain-drive. Accessed: May 5, 2025.

[5] Tsubaki of Canada, "Sprocket Number US-3075C9," 2024. [Online]. Available: https://catalog.tsubaki.ca/item/drive-chain-sprockets/sprocket-number-us-3075/us-3075c9. Accessed: May 5, 2025.

[6] Linn Gear, "1030C12 Sprocket," 2024. [Online]. Available: https://www.linngear.com/part/1030c12/. Accessed: May 5, 2025.

[7] K. Carlson et al., "Pull strength at overhead locations during reaching tasks," *International Journal of Occupational Safety and Ergonomics*, vol. 10, no. 1, pp. 1–9, 2004. [Online]. Available: https://www.tandfonline.com/doi/pdf/10.1080/10803548.2004.11076594

[8] Roy Mech, "Human Strength Tables," 2024. [Online]. Available: https://roymech.org/Useful_Tables/Human/Human_strength.html. Accessed: May 5, 2025.

[9] Treetools, "How much force can you apply with one hand?" 2023. [Online]. Available: https://treetools.co.nz/Blog/How_much_force_can_you_apply_with_one_hand_200N_apparently. Accessed: May 5, 2025.

[10] IE Park, "Roller Coaster Product Brochure," 2019. [Online]. Available: https://www.iepark.com/wp-content/uploads/2019/11/ROLLER-COASTER.pdf. Accessed: May 5, 2025.

[11] Mr. Wayne's Class, "Roller Coaster Design Interactive Tool," 2008. [Online]. Available: https://www.mrwaynesclass.com/apResources/coaster/web/index08.html. Accessed: May 5, 2025.

[12] Metal Supermarkets, "A36 Steel Grade Guide," 2024. [Online]. Available: https://www.metalsupermarkets.com/grade-guide-a36-steel/. Accessed: May 5, 2025.

[13] S. Brownlee, "Tug of War - History, Rules & Rope Selection Guide," Ravenox, 2022. [Online]. Available: https://www.ravenox.com/blogs/news/tug-of-war-with-our-soft-but-strong-twisted-cotton-rope. Accessed: May 5, 2025.

[14] Btiernay, "Grade Guide: A36 Steel," Metal Supermarkets, 2021. [Online]. Available: https://www.metalsupermarkets.com/grade-guide-a36-steel/. Accessed: May 5, 2025.

[15] ClayTownR, "Sprocket Design Tutorial - CAD - Chief Delphi," 2020. [Online]. Available: https://www.chiefdelphi.com/t/sprocket-design-tutorial/387449. Accessed: May 5, 2025.

[16] "Correlation of Height and Maximum Overhead Reach," 2025. [Online]. Available: https://journals.sagepub.com/doi/pdf/10.1177/0025802418816283. Accessed: Apr. 16, 2025.

[17] IE Park, "Gravity Roller Coaster Launch Coaster," 2019. [Online]. Available: https://www.iepark.com/wp-content/uploads/2019/11/ROLLER-COASTER.pdf. Accessed: Apr. 17, 2025.

[18] "Standard Practice for Design of Amusement Rides and Devices." [Online]. Available: https://store.astm.org/f2291-24.html. Accessed: May 4, 2025.