Ackermann Steering Geometry Applied to a Skateboard Truck

By

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Abstract

Conventional skateboard trucks are currently unable to meet the challenges of the modern enthusiast. They are lacking in key performance metrics such as handling, stability, and traction. Longboard enthusiasts, whom rely heavily on handling performance, are hungry for new and innovative technology to help bring the sport to the next level. The aim of this project was to solve these problems by applying specific aspects of automotive steering geometry and best engineering practices. Three successive prototypes were designed, with the first two prototype sets being manufactured and extensively tested. The first prototype served as a proof of concept, but suffered from design and manufacturing complexities and would have been too expensive to be mass produced. Positive and negative feedback was obtained from enthusiasts which was used to design the second prototype set. More testing was done and while the second prototype set showed major improvement across all key metrics, problems still existed. A third prototype design was developed to solve the remaining problems and is currently being manufactured. Overall, the result of the project is a longboard truck system that is superior to current products in terms of stability and handling. Its simple design and ease of manufacture allow for a potentially very competitive price point. Furthermore, the new technology will be a basis for future developments and refinement much like the roller skate truck has been since the 1940s.

Introduction

Skateboard truck innovation has progressed very little since skateboarding was invented. The trucks that riders use today are essentially a modified roller skate design. The most common complaint that longboard enthusiasts have is that skateboards are inherently unstable, and that handling and traction are sacrificed for added stability. Longboarding is a form of skateboarding that places an emphasis on handling performance. Longboard truck manufacturers are aware of these issues as evident in the large number of new products continually being released. These new products, however, are merely refinements of the original truck design and little new innovation has occurred. Some of the main problems include:


Figure 1: Roller Skates [19]

- Speed wobbles. This is an uncontrollable oscillation of the steered wheels at about 4-10 hertz. If the rider is unable to dampen them, a crash is inevitable.
- Lack of precision. The turn to lean ratio (see Key Terms Defined) is not identical for every turn because of natural slop in the system. This greatly decreases handling and control.
- Bump steer (see Key Terms Defined). Since the wheels are so far away from the central kingpin, hitting a bump will generally cause the board to self-steer.

- Scrubbing (See Key Terms Defined). Scrubbing is one of the main problems of turn table steering. During a turn, the wheels tend to scrub, taking away cornering traction and making “slide outs” more prevalent.

- Wheel bite. Since the wheels swing an arc about the centerline of the kingpin, there is a higher risk of the wheels making contact with the board, causing a crash. Therefore, if a rider wants to be able to turn sharper, he or she must place risers in between the trucks and the board which may increase instability due to the raised center of gravity.

This project will serve to solve many of these common issues regarding longboard truck performance by applying advanced vehicle dynamics and engineering principals. The objectives of the project are to:

1. Determine what design characteristics can be applied to a skateboard truck to increase stability and handling performance.
2. Develop conceptual designs of the mechanisms required.
3. Develop detail designs of the prototype.
4. Manufacture a working prototype set.
5. Collect feedback from riders.
6. Optimize the design to better meet established objectives.

The remainder of this report will outline the solution to this problem in detail, with the intent of having a design that is ready to be marketed to the public.

Background

The first portion of this section describes what a skateboard truck is and how it works. Furthermore, brief descriptions of some alternatives to conventional technology are also included. The second portion serves to define some key terms that are used extensively throughout the remainder of this report but are not explicitly explained.

Current Technology Explained

A skateboard truck is a device that is mechanically attached to the underside of a skateboard or longboard deck that allows the rider to control the direction of travel. Typically two of these trucks are used, one in the front and one in the back. They are mirror images of each other such that in a turn the back truck will steer in the opposite direction of the front. All skateboard trucks are lean activated such that leaning the board in one direction will cause the board to track in that direction. The truck achieves this by converting a percentage of the lean angle into turn angle. All conventional trucks have a single and central pivot axis, commonly referred to as the kingpin. There are two basic components, the hanger and the base-plate. The base-plate is mounted to the underside of the deck and serves as the mount for the hanger. The hanger, which is the carrier for the axle, is located on the base-

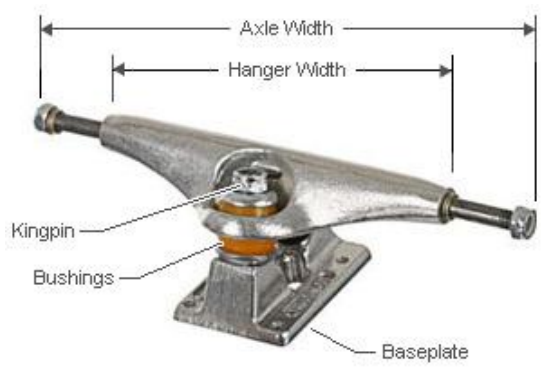

Figure 2: Conventional Truck [2]

plate by two points, a pivot cup and bushing cup. If an imaginary line is drawn between these points and referenced by the ground, the angle of the truck is defined. This angle is what determines the turn sensitivity. The smaller the angle is, the smaller the turn to lean ratio. Alternatives to the conventional truck have been designed, manufactured, and marketed. One is the reverse kingpin truck. The reverse kingpin truck is notable because the pivot axis and kingpin are 90 degrees to one another. This allows for less binding on the kingpin and a linear turn to lean ratio. Typically RKP trucks are suited for long boards where increased stability and decreased turn sensitivity are desired. The first and most common reverse kingpin truck is the Randal. A second alternative is the mechanics employed by Seismic Trucks, developed by Dan Gesmer. While similar to a reverse kingpin truck, the kingpin and pivot axis are collinear. Furthermore the truck utilizes coil springs as opposed to polyurethane bushings to return the truck to center. Benefits of this configuration are increased rebound and decreased stress on the kingpin. A third alternative is the Stroker truck developed by John S. Solimine as detailed in US patent number 4054297. Solimine's design utilizes two vertical kingpins positioned slightly inboard of the wheels. Spindles, which house the wheel axles, rotate about the kingpins. Connected to the spindles are steering arms. The base plate rotates about a point slightly below the axle height. Two links with universal joints at each end

Figure 3: Reverse Kingpin Truck [14]

Figure 4: Seismic Truck [15]

Figure 5: Stroker Trucks [17]

tie the base plate to the left and right steering arms. By leaning, the links transfer the force which causes the spindles, and thus the wheels, to rotate about the kingpins (patent #4054297). This system can be compared roughly to typical automotive steering setups. Stroker trucks, however, were designed with turning ability, not high speed stability, in mind. Therefore the track width is very narrow and the steering sensitivity is very high. Furthermore, they were not successful due to their design complexity and frequent mechanical failures of the linkage system.

Key Terms Defined

Ackermann:

For steering systems that utilize two kingpins, the inside wheel must turn with a smaller radius than the outside wheel. A steering linkage that approaches this condition is said to have Ackermann geometry. [See Appendix]

Scrub:

If a steering system does not achieve Ackermann, one of the steered wheels must lose traction in order to complete the turn. In simple terms, the wheel must scrub against the ground instead of merely rolling.

Turn to Lean Ratio:

The rider must lean the skateboard deck either right or left in order to initiate a turn. For a given amount of deck lean, the skateboard will turn a corresponding amount.

Turntable Steering:

Both steered wheels are rigidly fixed to a solid axle which is able to pivot about a central kingpin. [See Appendix]

Bump-Steer:

If a bump in the road surface causes a vehicle to steer without the user’s input, the vehicle is said to have suffered from bump-steer.

Literature Review

The literature review briefly describes the research that was completed to effectively solve the problems laid out in the introduction. The first few subsections deal with vehicle dynamics theory of skateboards, bicycles, and automobiles. The last few subsections highlight key information regarding product and fixture design as well as high speed machining practices.

Speed Wobbles on a Skateboard

"High speed wobbles or shimmies are an extraordinarily complex phenomenon that even leading professors of Mechanical Engineering can't fully explain. They are simply a fact of life for any type of flow motion sports gear that incorporates moving joints of any kind" (Gesmer). Speed wobble can be characterized by an uncontrollable weave that continues to increase in magnitude. Speed wobble can occur on a skateboard when a vibration is created that matches the natural frequency of the system. This vibration can be initiated by movements of the rider's center of gravity, a bumpy road surface, or a combination of the two. A skateboard/rider system may have a number of natural frequencies. In 1979, Stanford University conducted a study which compared the oscillatory behavior of a skateboard to that of an aircraft. They found that, like aircraft, skateboards go through various zones of oscillatory stability and instability as a function of speed. Gesmer goes on to comment that the geometry of the truck plays a significant role in how able the skateboard is in absorbing vibrations. He concludes that while no truck currently incorporates it, positive trail and caster could significantly reduce the chance of speed induced wobble because gravity would serve to return the board to center (Gesmer, 2000). An analogy to aircraft may also be of value. Aircraft dynamic stability can be quantified and falls into two categories, positive and negative. Negative dynamics stability occurs when displacements from the desired trajectory continue to increase until failure of the aircraft. In positive dynamic stability, the forces and moments acting on an aircraft allow it to return to a steady state after an initial displacement. This phenomenon can be compared to a ball bearing placed on a curved surface. If the surface is concave, the ball will also be forced to the center due to the effects of weight. If the surface is convex, the ball will only be at equilibrium (stable) when the ball is at the apex. Therefore, any displacement will cause the ball to be unstable (Pallett, 1987).

Stability Enhancing Characteristics of Bicycles and Automobiles

Available literature regarding skateboard dynamics is extremely limited. Therefore the steering mechanism of a bicycle may be compared and contrasted to the steering mechanism of a skateboard in that both operate with a single steering axis and both rely heavily on the balance of the rider. Furthermore, my design, a truck closely resembling rack and pinion geometry may be compared and contrasted with the stability enhancing characteristics of an automobile.

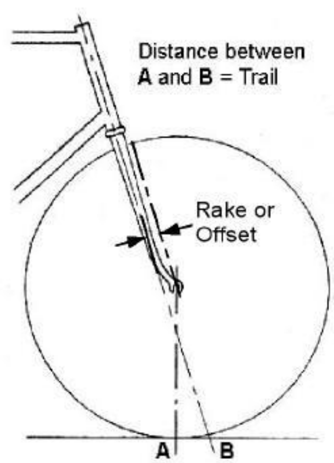

Figure 6: Bicycle Trail [1]

There are a number of factors which affect the stability of a bicycle, but the geometry of the front fork is what is relevant to this discussion. The main contributing factor to the stability of a bicycle is trail. Trail is defined by the linear distance between the contact patch of the front wheel and the imaginary intersection between the steering axis and the ground. Stability is increased when the distance the front wheel trails is increased. The reason this is stable is because the wheel wants to naturally find this position, much like the caster wheel of a grocery cart wants to naturally fall behind the caster axis when being pushed. This is precisely why all bicycle forks are angled back. If the fork were angled forward, negative trail would be induced and decrease the stability of the bicycle. Furthermore, positive trail is what causes the bicycle to self-correct in order to stay upright. If the fork had zero or negative trail, the rider would be responsible for sensing that he or she is falling over and manually correct by adjusting the angle of the handlebar. The reason that infinite trail is not used (a completely horizontal steering fork) is because the bicycle would not be able to turn. A compromise must be made between turning ability and stability. A common misconception is that the gyroscopic effect of the wheels is what makes a bicycle stable. In reality, its effects are largely overshadowed by the stabilizing effects of trail (Jones, 1970).

Automobiles utilize a number of design characteristics to increase stability at speed. The first vehicle steering system is commonly referred to as “turntable steering” [See Appendix]. Much like a skateboard truck, it uses two wheels fixed on a common axle with a central steering axis. A split steering axis system was first invented by Erasmus Darwin. Due to the relatively long distance between the steering axis and wheel, the diameter of the wheel must be sufficiently small so that it can swing under the frame of the carriage. Furthermore, Darwin comments on the stability of the old steering design. He states that the “basis on which the carriage rests changes from a rectangle to a triangle if the turn is severe”. With his invention he was able to eliminate both the problem of having small wheels as well as the mitigating the stability decrease in a turn. Darwin's steering invention is essentially what all automobile's use today (King-Hele, 2002).

There are a number of design characteristics which can serve to increase or decrease the overall stability of the vehicle (Dixon, 1996). The ones that are pertinent to this project include:

1. **Ackermann condition**. When an automobile makes a turn, there is a specific instantaneous center of rotation that all points, particularly the wheels, rotate about. Therefore, the wheels on the inside must turn a smaller radius than the wheels on the outside. An automobile that can achieve this is said to have perfect Ackermann geometry. In reality, perfect Ackerman is impossible to obtain from simple linkages. The idea of having a perfect steering geometry is somewhat outdated, mostly required by old coaches and buggies so that the process of steering would not upset the gravel roads due to tire scrub. (Jazar, 2008) Modern automobiles almost exclusively approach a parallel Ackerman geometry where the inner and outer wheels follow nearly the same radius. This may increase the steering torque required at low speeds but at high speed the steering torque is decreased. For this reason, most race cars employ slightly negative Ackerman where the outer wheel turns a sharper radius than the inner wheel (Gillespie, 1992). Reasons for this include decreased tire temperature and decreased slip angle induced drag. Furthermore, the need for Ackerman

geometry becomes increasingly insignificant as the track width of the vehicle decreases as well as when large turning radii are common. (Milliken, 1997). [See Appendix]

2. **Steering Ratio.** This is defined by the rotation of the steering wheel divided by the corresponding steering axis angle. Typical race cars have anywhere from a 20:1 steering ratio to 1:1. The cars that typically must run in an accurate straight line have higher ratios while cars which must be nimble use lower ratios (Milliken, 1997).
3. **Caster angle and Trail.** Both of these characteristics are what cause an automobile steering geometry to self-center. Caster angle works more for low speed while trail is more important for high speed. Caster angle is defined as the angle of the kingpin in a side view. Typical angle values can range anywhere from 0 to 10 degrees, depending on requirements. This causes the center of gravity of the vehicle to rise, or “jack” in a turn. Since the system always wants to find the point of minimum potential energy, the wheels will center when pressure from the steering wheel is removed. This is only effective at low speeds where steering angles are high. For high speed stability, trail is employed and is identical to trail for a bicycle. It is defined by the linear distance between the contact point of the tire and the imaginary intersection between the ground and the kingpin axis. It should be noted that positive caster nearly always increases trail (Milliken, 1997).
4. **Kingpin Inclination Angle.** This is identical to caster but viewed from the front. This is important for determining the scrub radius.

5. **Scrub Radius.** This is defined by the distance between an imaginary intersection between the kingpin and the ground and a true vertical line that passes through the center of the wheel. Scrub radius is important when looking at the effects of braking. The greater the scrub radius, the larger the moment about the steering axis. The force of braking can cause this torque. If the scrub radius is zero, there will be no moment. (Gillespie, 1992). The road surface may also be a factor, albeit to a lesser extent when compared to braking torque. Typically, these driving forces are not typically large enough to permit drastically reducing the scrub radius (Milliken, 1997).
6. **Four Wheel Steering.** Four wheel steering is the ability of both the front and rear axles to have steering capability. A positive four wheel steering (4WS) system, where front wheels turn one direction and the rear wheels turn the other, is beneficial for decreasing turning radius at low speeds. A Negative 4WS system, where both front and rear wheels turn the same direction, is beneficial for high speed stability and yaw minimization but it increases the turning radius. A Positive 4WS is typically not recommended for high speed applications because it lends itself to over-steer (Jazar, 2008). By being able to over-steer, a positive 4WS system suffers from a critical speed. Critical speed is the theoretical maximum speed a vehicle can reach before any slight change in steered direction will cause the vehicle to lose control (Jazar, 2008).

Product and Fixture Design, Minimum Constraints, and Prototyping

Minimum constraint design is defined as supporting and guiding each body or component only at points and at as few points as possible to get the result and functionality that is required. By doing this, the chances of binding and slop between moving parts is greatly reduced. Furthermore, manufacturing tolerances can potentially be increased. (Kamm, 1990) This is particularly important for this project, where redundant constraints can easily become a problem from by both a design and a manufacturing perspective. Because the solution to eliminating the inherent instability of a skateboard truck will require a somewhat complicated mechanical device, a design which reduces the number of redundant constraints is preferred.

A redundant constraint may be preferred if it is used in a supporting role. One example is a three jaw lathe chuck. Two of the jaws are enough to locate a circular part, yet the third one is required for clamping purposes. A redundant constraint must only be added if absolutely vital to functionality (Kamm, 1990).

There exist two types of prototypes, analytical and physical. An analytical prototype typically precedes a physical prototype because specific constraints and parameters are much easier and less expensive to alter. Typically the analytical prototype is used to create a feasible range of parameters. The physical prototype may be then used to confirm the design. A physical prototype may be very beneficial in detecting and correcting unanticipated phenomena. One of the problems with analytical prototypes is that they are not controlled by the laws of physics, only by certain equations that are approximations. Therefore they can never reveal phenomena that are not addressed directly in the model. A prototype may also be useful in that it can expedite other developmental steps (Ulrich, Eppinger, 1995).

High Speed Machining

Because this project will most likely require machined parts, high speed machining may be of value. From a manufacturing standpoint, high speed production of machined parts is a conglomeration of ultra-flexible tooling, fast throughput, minimum number of machining passes, predictable machining results, and a good part design. High speed machining is increasingly important when the medium to be machined is a non ferrous material, particularly aluminum. And with new developments in cutter materials such as polycrystalline diamond, high speed machining can greatly reduce cycle times (Erdel, 1993).

Where high speed machining is different from conventional machining, excluding advancements in machine tool technology is the economic approach. Previous economic models were expressed in terms of speed, feed, depth, tool life, tool change time, setup time, overhead, and tool costs. New economic models go beyond this and look at it from a much more general standpoint. For example, they might consider transportation time, penalty costs, and waiting costs (King, 1985). By looking at it from a different perspective, increasing the material removal rate while simultaneously decreasing tool life considerably may be cost effective.

Design

The design phase consisted of conceptual and detail designs as well as manufacture of two separate prototypes. Furthermore, a third prototype design phase was also initiated. Each consecutive prototype was refined based on testing feedback from the previous prototype.

Conceptual Design of Prototype 1

The conceptual design began with the identification of certain requirements in order to maximize a longboard's handling and stability. An outline of characteristics was developed that would be used to generate the detail designs later on. They included:

1. **Track width:** The distance between the centerline of each wheel. Typically wider trucks are more stable but if the truck is too wide the likelihood of the pushing foot making contact with the back wheel is high. Therefore, a width between 210 and 220mm was determined to be suitable for this project.

2. **Ride height:** This is defined by the distance between the top of the baseplate and the ground. A lower ride height will increase stability but too low would cause issues with the deck hitting the ground during a sharp turn. The lowest possible truck was calculated based on factors such as board lean, board width, and roll axis height.

3. **Roll Axis:** The roll axis is defined by the imaginary centerline that the board rotates about during a turn. While a low roll axis is typically beneficial for handling, a higher roll

axis is better for stability. This is sometimes referred to as the “hammock effect”. If the roll axis is above the surface of the deck that the rider has his or her feet on, then their weight will be forcing the board to center itself. This makes it very stable, but handling is compromised because the rider’s center of gravity must rise which takes energy. Therefore, a roll axis that was positioned between 1.0” and .5” below the rider’s feet was selected.

4. **Turn to lean ratio:** This ratio is calculated by dividing the steering angle of each wheel by the lean angle of the baseplate. A truck with a high ratio will tend to be better at slalom style riding- quick left to right carves in succession around cones. This is because the board does not have to cycle through as much angular displacement. A truck with a lower ratio will tend to be much more stable but handling is sacrificed. Therefore, a very stable ratio of .6 was selected which is extremely low for a longboard truck. However, the lean angle was set very high, at 25 degrees, so a relatively small turning radius was still possible. With this information, the roll axis to tie rod distance was set at .54 inches and the steering arm length to .75 inches.

5. **Linkage Design:** This is the mechanism that is used to connect the two knuckles together and is the single most important and most complicated design aspect. The first task was to decide on the basic linkage system that would be used. After weighing the positives and negatives of a number of different linkages, a solid tie rod design was chosen. A solid bar, known as a tie rod, would be passed through the hanger and located by a transverse bore that is both parallel to the ground and perpendicular to the direction of

travel. The tie rod would be able to move back and forth along the axis of the bore. At each end of the tie rod would be an intermediate link that would connect to the steering arms. Therefore, when the tie rod slides back and forth, it would case the knuckles to rotate. This is essentially how rack and pinion steering works but the need for universal joints in place of the intermediate links is not a requirement because there is no suspension. This design is superior for a number of reasons. First, it is very strong and the tie rod itself acts as a structural member for the truck. Second, Ackermann angles can be precisely controlled because the linkage always remains in a plane. Third, since the tie rod is located by the bore, flex is greatly minimized. And fourth, this design would be very conducive to an innovative return to center mechanism.

6. **Return to Center Mechanism:** This is the system that causes the deck to return to a position of zero steering. The tie rod would play a key role in the development of the return to center assembly inside of the hanger. The basic concept is that when the tie rod displaces either left or right of the center point, it causes a spring to compress. Below is a simple hand drawn sketch of this design:

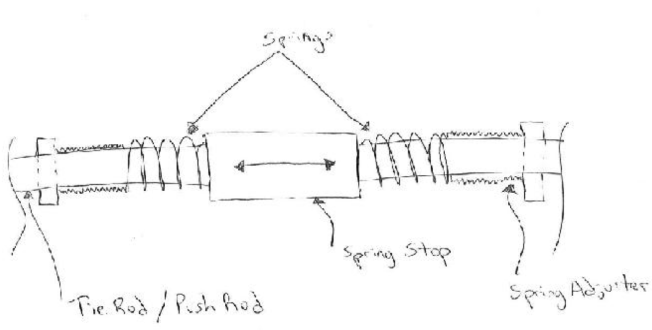

Figure 7: Conceptual Design, Return to Center Mechanism

At the center point, when the board is tracking in a straight line trajectory, the two rebound spring rates cancel each other out. The static spring preload can be adjusted using the two spring adjusters. The spring adjusters are essentially bolts that have a hole in them allowing the tie rod to pass through. If the rider wants to increase the rebound force, he or she need only screw in the spring adjusters.

The spring rate of the rebound system is especially critical because that is what dictates the “feel” of the board for the rider. A force of 50 pounds (riders weight bias on the edge of the deck during a turn) was used to calculate spring rates and it was determined that 9/16” square wire die spring could meet this demand.

7. **Steering Actuation:** This deals with how leaning motion will be connected to steering motion. To do this, the rotational motion of the baseplate must be converted to linear motion acting on the tie rod. Furthermore, a lean to the left must steer the front wheels counterclockwise and the back wheels clockwise. To do this, a slot and pin mechanism would be employed. The pin would be fixed to the block that the springs act upon. The pin would have two roles: to secure the spring stop to the tie rod and to be the component that interacts with the slot in the baseplate. Below is a sketch of the mechanism:

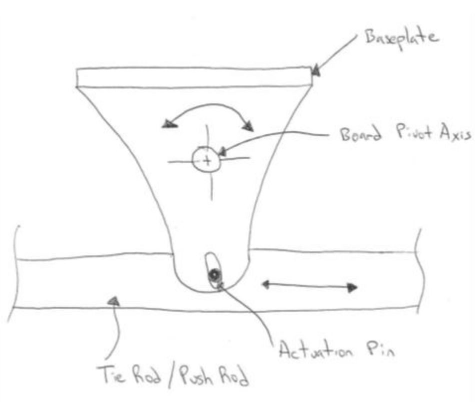

Figure 8: Conceptual Design, Steering Actuation

As shown, the actuation pin is moved back and forth by the rotation action of the tie rod. The benefits of this design are that it is extremely simple and easy to manufacture. The drawbacks are that wear is increased, and the pin is in single shear and very highly stressed.

Detail Design of Prototype 1

Once the conceptual design was completed the basic mechanisms that truck would be comprised of were identified, the detail component design phase was started. The first component to be designed was the main body of the truck, called the hanger. The hanger has a number of different functions and is the part that allows the mechanisms to interact with each other. Using the data gathered during the conceptual design phase, the hanger was designed using Pro- Engineer modeling software. Aside from function, the hanger is a highly stressed member so material selection was important. 6061-T6 was decided upon because of its high strength to weight ratio, affordability, and machinability. Basic mechanical stress analysis was performed on the hanger to ensure that it was strong enough.

After the hanger was designed, the next set of components to be designed was the knuckles. The knuckles are the components that the wheels mount to through the use of axles. Aside from that, they also incorporate the steering arms. The steering arms connect to the intermediate links which connect to the tie rod. The angle and length of the steering arm play a significant role in determining the handling characteristics. Parallel steering arm geometry was decided upon. This means that the wheels will be parallel to each other at all times even during turns. The theory behind this is that even though the scrub will occur during turns and decrease traction, it has a centering effect. The steering arms also play a role in the determination of the steering ratio. Recall from the conceptual design phase that a steering ratio of .6 was ideal. To obtain this, a steering arm length of .75 coupled with a board pivot axis to tie rod axis distance of .54, was determined. The amount of spring compression of the return to center mechanism was also taken into consideration to determine these numbers. The knuckles are very highly stressed because the must take the full weight of the rider as well as withstand continual dynamic shock loading. As a result, low carbon steel was decided upon as the material because of its attractive cost. Since the geometry is fairly complicated, the knuckles are actually a welded assembly of three components: the kingpin bearing, the steering arm, and the axle mount

The tie rod is the last main component that makes up the steering linkage system. It is the component that ties the two steering arms of each knuckle together and slides back and forth through the spring adjusters that are threaded into the hanger. The most critical aspect of the tie rod is the center to center distance between the two end holes. These holes house one of the two pins of each of the intermediate links. The distance is essentially determined by what will make the two wheels parallel to each other during straight trajectory, not by selecting it. Furthermore, the tie rod has the function of being what is shuttled back and forth. There is a hole at the center which a steel pin is able to pass through. That pin also fastens the spring stop to the tie rod. The pin sits in a slot milled into the baseplate.

The baseplate is the last main component of the design. It is the component which directly mounts to the board. When the rider leans, it causes the baseplate to rotate about the board pivot axis. This action forces the tie rod to shuttle in the opposite direction of lean via the actuation pin. The slot in the baseplate not only experiences high force, but there is also sliding between the pin and the slot. For this reason, steel was selected as the material because it has good wear resistance qualities. Furthermore, since steel is fairly hard to machine and the part is essentially a deep channel with thin walls, a three piece weldment was decided upon. Each plate is 3/16" thick which is a good compromise between strength and weight. The plate that the board physically connects to has two shallow channels milled into it that aide in locating the other two plates prior to welding.

Below are two renderings of the final prototype design. The first is a frontal view and the second is a rear view.

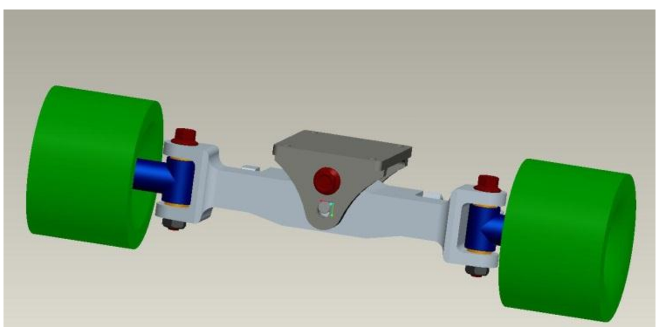

Figure 10: Front View, Prototype 1

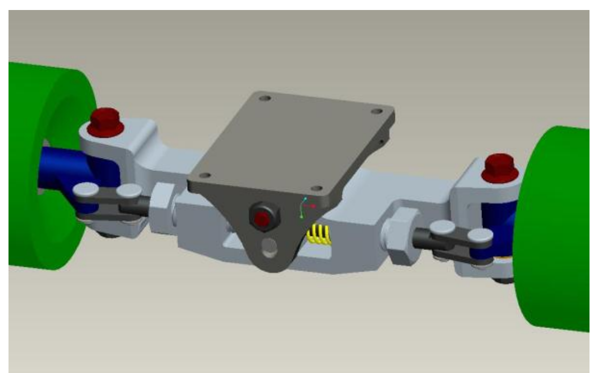

Figure 9: Rear View, Prototype 1

Design of Prototype 2

During the design phases of prototype 2, there were two main concerns: to fix all of the issues that were realized during the testing of the first prototypes, and to design a truck that could be potentially marketed. Test data and detailed explanations of the problems with the first prototype design are located in the Methodology section of this report. The remainder of this section will outline the design process of prototype 2 and how the negative feedback was addressed.

The most obvious and dominating problem with prototype 1 was the return to center mechanism and the first problem that was addressed when designing prototype 2. The first problem with the return to center mechanism was that it was simply not stiff enough. When calculating spring rates, it was wrongly assumed that a leaning force of 50 pounds was an accurate estimation. In reality, the required force would need to be a least doubled for an average weight rider. Furthermore, there was not enough finite adjustment for the rider. Also, for very small lean angles the mechanism provides negligible resistance. At the center, the two spring forces are equal and opposite. If the spring stop is pushed slightly to the left, the spring on the left will compress and the spring force will increase. However, the spring on the right side is still causing the spring stop to move to the left, but at a decreasing rate. In mathematical terms, the force causing the spring stop to move back to the right is:

$$Centering\ Force = FRightx + dx - FLeftx - dx$$

Eventually, when the displacement is sufficiently large enough, the second term essentially drops out of the equation. But for corrective movements, such as during downhill, the rebound feels non-existent. To remedy this problem, I designed a mechanism that would enable the springs to have extremely positive centering ability. Included on the following page is a simple sketch of the design:

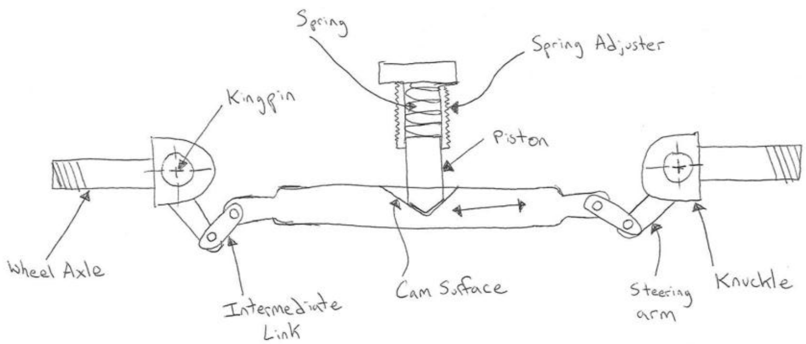

Figure 11: Spring Detent Return to Center Mechanism

The mechanism basically operates similar to a spring detent. A v-shaped cam profile is milled into the tie rod. A piston is able to sit in the v-shaped groove and can slide in and out of a bore milled into the component labeled "spring adjuster". Inside the spring adjuster is a set of belleville disc springs. When the tie rod moves back and forth within its bore, it causes the piston to compress the belleville disc springs. At the center point, the springs are already preloaded so that a powerful centering force is possible even at small displacements of the tie rod. Furthermore, the Belleville springs are extremely versatile. They typically possess high spring rates at relatively low displacements and are the most compact spring available. The disc springs can also be stacked in a large number of combinations making the spring rates highly conducive to tuning.

In order to determine the return to center mechanism design parameters such as spring rate, spring displacement, and cam profile angle, an Excel model was developed[See Appendix]. From the calculator, it was determined that with a cam profile angle of 30 degrees, two springs sets of fifteen ½" belleville disc springs would supply the required spring force.

Recall from the first prototype design that the Ackermann angles were set such that the steered wheels of each truck always remained parallel to each other. While this set up increased stability, handling and traction were sacrificed. For the second prototype design, the Ackermann angles were refined such that perfect Ackermann geometry was closely approached. To do this, a parametric sketch was created using Pro-Engineer where the steering arm angles and intermediate link angles could be finitely adjusted. These adjustments could be tested by sweeping through its full turning motion to see if the intersection circle was of sufficiently small diameter. Special consideration was applied to high steering angles because the negative effects of scrubbing are magnified during tight turns. Below is a screenshot of the sketch; note the intersection point circled in red:

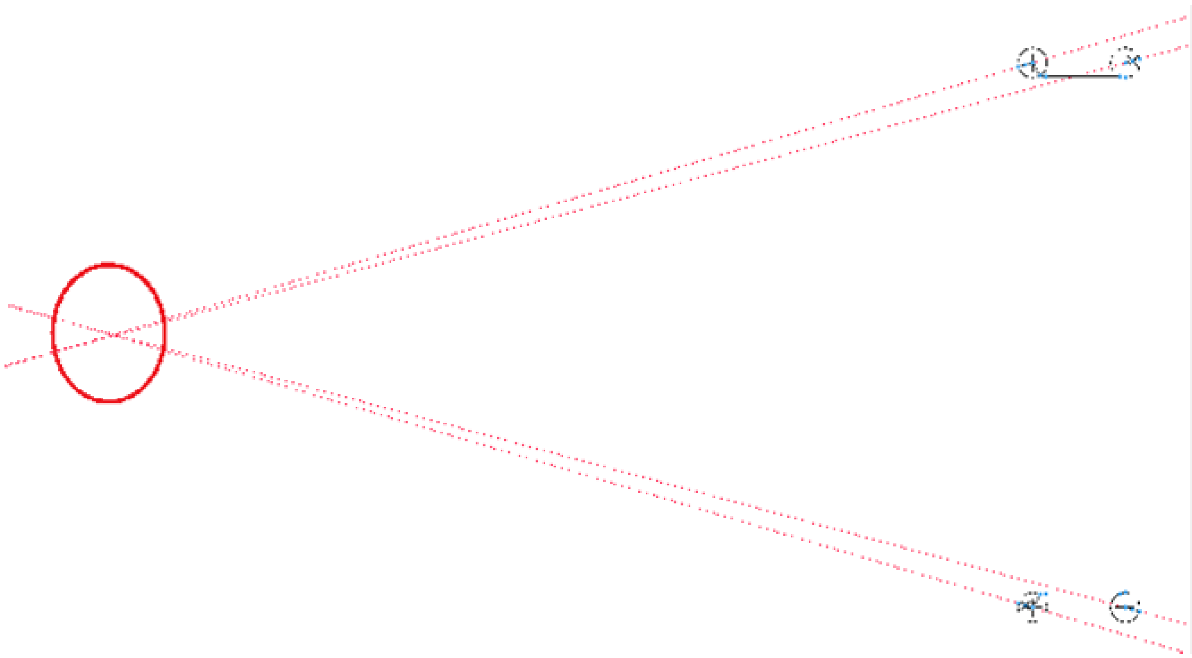

Figure 12: Ackermann Steering Achieved

In this state, the skateboard is in a severe left hand turn, but all of four wheels are steering about virtually the same point. This will translate to maximum traction during turns while speed is not excessively reduced.

Another main concern that was revealed after initial testing was the excessive wear occurring between the baseplate slot and the actuation pin. Different ideas were brainstormed on how to shuttle the tie rod back and forth that would significantly reduce the problems with the slot and pin design. A ball and socket mechanism was decided on for two reasons; one is that the effective bearing surface area is greatly increased, and two, a ball and socket allows for easy grease lubrication.

Below is a simple sketch of the mechanism:

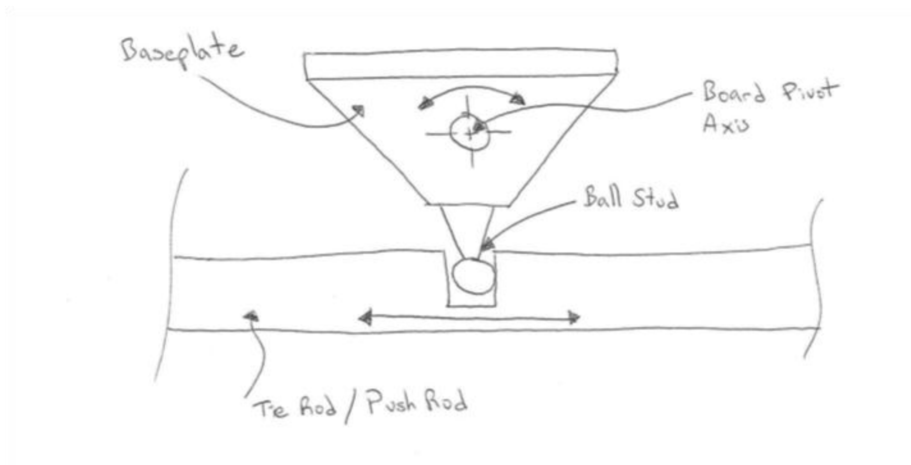

Figure 13: Ball Stud Steering Actuation

Aside from the general advantages of the ball and socket mechanism, it allowed the use of a very inexpensive component to be purchased instead of manufactured. For this application, a standard gas spring ball stud which costs about $.85 each was used. Furthermore, the ball stud would allow the baseplate to be constructed of aluminum, thereby decreasing the overall weight of the trucks.

As far as the detail design was concerned, the biggest priority was design for manufacturability and assembly because the critical concepts had already been proven. This was broken into two key metrics; number of make parts, and number of fixtures required. Overall, the number of make parts was reduced from 23 to 11 and the number of fixtures required was reduced from 6 to 1. Below are two brief tables identifying the DFM and DFA improvements for the main components of the truck:

Table 1: DFM Improvements

COMPONENT	DFM IMPROVEMENTS
Hanger	• Soft jaws no longer a requirement • # of setups reduced from five to four
Knuckles	• Consolidated from a three component weldment to a single machined part • # of setups reduced from six to two • # of fixtures reduced from two to zero • Cycle time and cost significantly reduced due to removal of welding process
Baseplate	• Consolidated from a three component weldment to a single machined part • Aluminum construction for easy machining • Manufacturing accuracy significantly increased
Tie rod mechanism	• # of parts required for tie rod mechanism reduced from three to one

	• # of make parts reduced from
Intermediate Links	• Intermediate links for new design are ANSI #35 connecting links, a purchased part

Table 2: DFA Improvements

COMPONENT/ Sub-Assembly	DFA IMPROVEMENTS
Return To Center Mechanism	• Each of the return to center mechanisms was designed as a sub-assembly that is easily attached to the hanger • Springs are preloaded using spring adjuster bolts and are much easier to install
Knuckles	• Bronze bushings no longer required, reduced assembly time
Intermediate Links	• Intermediate links are ANSI #35 connecting links and come pre-assembled. • Since the pins are precision ground, they permit low friction insertion into the steering arms and tie rods
Steering Actuation	• The steering actuation sub-assembly was reduced from 3 to 2 parts, simplifying the assembly process

Below are two renderings of the final design which incorporate the improvements previously described:

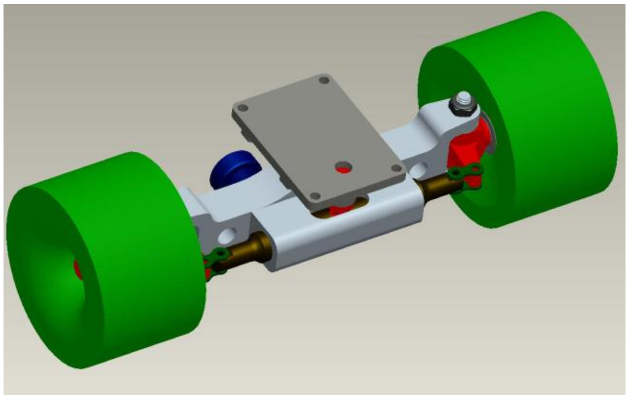

Figure 15: Rear View, Prototype 2

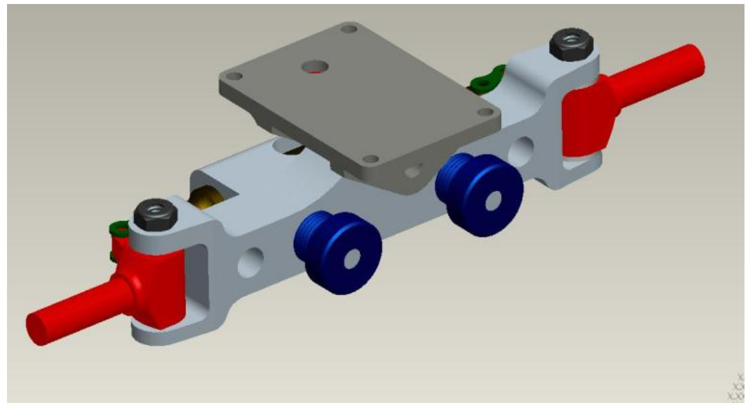

Figure 14: Front View, Prototype 2

These improvements have allowed a design to be developed which has serious potential for marketability at an attractive price point for the consumer. The cost of the trucks will be further explored in the next sub-section.

Cost Analysis

After testing of the first prototype set, the high marketability potential was realized but manufacturing costs would have been too expensive. Since the potential consumer of this product is relatively young and of limited means, the need for low manufacturing costs was magnified.

During the design phase, a number of considerations were made to minimize the costs. Since the market size is low, limited setup times and quick cycle times are crucial so that current demands can be met dynamically and a pull system can be approached. To obtain this, the trucks were designed such that specialized tooling is not required and the level of precision necessary is minimal. Below is a manufacturing cost breakdown for the second prototype design:

Table 3: Manufacturing Costs

Component	# per set	Make/buy/ modify	Estimate/ actual	Unit cost (n=250)	Cost Per Truck (n=250)
hanger	1	Make	actual	$8.52	$8.52
tie rod	1	Make	estimate	$3.75	$3.75
knuckle	2	Make	estimate	$4.00	$4.00
piston	1	Make	estimate	$2.00	$2.00
spring adjuster	1	modify	estimate	$1.25	$1.25
baseplate	1	make	estimate	$7.00	$7.00
belleville disc springs	12	buy	actual	$0.25	$3.00
ball stud	1	buy	actual	$0.85	$0.85
nylon thrust washers	4	buy	actual	$0.40	$1.60
kingpin, grade 8	2	buy	actual	$0.17	$0.34
kingpin nut	2	buy	actual	$0.15	$0.30
ANSI #35 connecting link	2	buy	actual	$0.40	$0.80
baseplate axle	1	buy	actual	$0.45	$0.45
baseplate bushing	2	buy	actual	$0.41	$0.82
assembly cost			estimate	$3.00	$3.00
				Total	**$37.68**

In order to obtain data for the machining costs of the make parts, I created an engineering drawing of the hanger and sent an RFQ to multiple foreign and domestic machine shops. With a production volume of 250, one Chinese shop quoted $8.52. While it was not the lowest quote, the company has a good reputation among customers and their quality is high. Taking into account material costs, number of setups, and machine time, costs of each of the make parts was estimated. The buy part costs are actual quotes from vendors. Overall, the manufacturing cost for one truck is $37.68. This included the assembly cost as well. This cost was estimated based on an experimental assembly time of 3.0 minutes. Using an average job shop hourly rate of $60/hour, the cost to assembly one unit was estimated to be $3.00. Since trucks are sold as a set, the unit cost with a production volume of 125 sets is $75.36. The main competitors of these trucks are referred to as *precision trucks* and are priced between $300 and $700. Therefore, the price to cost ratio may be potentially very high.

The production volume of 125 sets was chosen because the market size is relatively small and would most likely only consist of enthusiasts who purchase very expensive upper tier products. If this assumption is underestimated, however, it is important to discuss the scenario of increased demand. As the production volume increases, price breaks are awarded by both job shops and vendors, but at a diminishing rate. There exists a point where other manufacturing methods such as casting or forging become more cost effective. While that type of analysis has not been completed, it has been a factor in the direction of the design. While prototype 2 is not welcoming to net shape manufacturing processes, the 3rd prototype design is [see Future Plans].

Methodology

This section of the report will outline the fabrication and assembly process of the first and second prototype sets. Furthermore, the testing methods will be described and the data collected will be explained.

Manufacture and Assembly of Prototype 1

The manufacturing procedures to construct the first prototype set were fairly straight forward. A combination of CNC and manual mills and lathes was required for each component. Furthermore, six separate fixtures were designed and fabricated to support the manufacture of the hanger, baseplate, knuckles, and tie rod. The hanger required five milling operations. For the first three, a custom set of vice jaws was designed and machined. The last remaining two operations were reserved for drilling and tapping the holes for the spring adjusters. Since the two tapped holes could not have been done on the same setup, it required a fixture that would allow the part to be flipped with a very high degree of accuracy. To do this, an

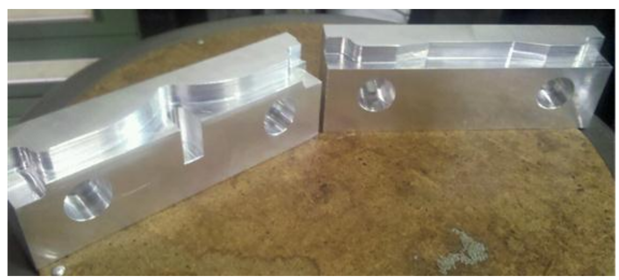

Figure 17: Soft Jaws for Hanger Machining

Figure 16: Vertical Fixture

aluminum plate was faced and drilled in order to press fit four ½ inch precision ground pins into it. A toggle clamp was then fastened to the plate in order to rigidly fasten the hanger to the fixture. This assembly was then clamped to a precision ground cast iron angle block which was bolted and squared up to the machining table. This fixture allowed me to drill and tap the two holes on two separate setups with enough accuracy to ensure that the tie rod could pass through the spring adjusters without binding.

The fabrication of the knuckle was the most time consuming process of the manufacturing phase. It required machining six components, two fixtures, and one welding procedure. First, the kingpin bushing and the axle mount were machined using a standard manual mill and a manual lathe. For the steering arms, a plate fixture was required to secure the steering arms. The fixture allowed four steering arms to be machined at once. Once the components were machined, the aluminum fixture necessary to accurately weld the components together was fabricated. The fixture design allowed for all of the key features to be machined on one setup, thus utilizing the machines accuracy. The axle holder was located by a v-groove while the steering arm and kingpin bearing were located by two precision ground dowel pins. Each

Figure 19: Steering Arms, Post-Machining

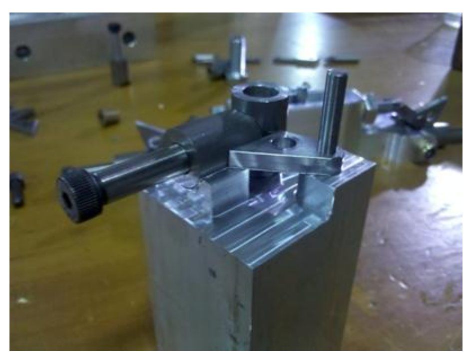

Figure 18: Knuckle Welding Fixture

of the four knuckles was then assembled on the fixture and TIG welded together, ensuring that the part never got hot enough to warp or bind onto the fixture.

The baseplate was machined next. The baseplate is comprised of three components machined out of 3/16” thick steel plate. A plate fixture was built to clamp the stock to and all of the components to make one baseplate were machined on one setup. Since the parallelism of the two side plates was crucial, an aluminum slug with a hole thru it was turned such that it could be placed in between the two plates and fastened with the pivot axle bolt. Once the components were mocked up, the assembly was welding using a TIG process.

The last few components to be machined were the actuation pins, spring stops, intermediate link plates, and tie rods. The actuation pins are essentially a short length of ¼” steel bar stock that necks down to a 3/16” diameter and is threaded. All of its features were machined on a manual lathe. The spring stop is a rectangular block with two holes in it, one being threaded for the actuation pin. Both spring stops were machined on a manual mill. All eight intermediate link plates were machined on a manual mill and then saw cut and chamfered using a belt sander. The fabrication of the tie rods required a specialized fixture such that the holes for the intermediate links and the hole for the actuation pin could be machined at a very accurate degree of perpendicularity. To do this, a 3” length of ¾” square bar stuck was cut and drilled radially on center with a drill .001” greater than the diameter of the tie rod. Then at the center of the block and on one of the sides, a small hole of the same diameter as the actuation pin was drilled. Perpendicular to that surface two small holes were drilled for set screws. In order to machine the tie rod, a length of 3/16” diameter steel round stock was cut and placed in the fixture, using the set screws to secure it in place. At that point, the small hole for the actuation pin was drilled using the drill bushing hole machined into the fixture. Then the fixture was placed in the vice of a manual mill so that the holes for the intermediate link pins could be drilled.

Once all of the components were made, they were assembled in order to have two working trucks. First, the bronze bushings were pressed into the knuckles and installed into the hanger using the kingpins. Then the spring stop was placed in the hanger's slot along with the two die springs. The spring adjusters were then screwed into their respective holes until a light preload was experienced by the springs. Finally, the tie rod was then able to be passed through the adjusters. Once the tie rod was in place, the intermediate links were used to connect it to the left and right steering arms of the knuckles and cotter pins were used to secure them in place. At this point the truck was fully assembled but missing the baseplate. The bushings for the baseplate pivot axle were pressed into the hanger and then the baseplate assembled such that the pivot axle bolt could be passed through and secured with a lock washer and nut. This process was completed for the second prototyped truck and both trucks were fastened to a standard longboard deck. Below is a picture of the assembled board taken during testing:

Figure 20: Assembled Longboard, Prototype 1

Testing of Prototype 1

Once the skateboard was assembled, the testing phase began. Initially, this phase involved extensive riding by myself but was later expanded to allowing my friends ride them and provide feedback. Testing spanned a wide variety of different conditions including:

- **Downhill:** This is the act of riding down steep hills with the intent of going as fast as possible. Light sweeping carves are used to slow down if necessary. Stability is critical for this type of riding.
- **Carving:** This is a test of the trucks handling ability. It can be done at slow speeds or down hills, but typically consists of making quick back and forth carves at elevated turning forces. Traction is critical for this type of riding
- **Flatland pushing**: Flatland pushing is the least performance intensive form of longboarding but also the most common. It consists of commuting at slow speeds, riding on sidewalks, in and out of pedestrians, and over rough terrain. Good durability, handling, and low weight are critical for this type of riding.

Total test time exceeded 20 hours and was discontinued once enough feedback was generated. That feedback, both positive and negative, is located in the Results section of this report.

Fabrication and Assembly of Prototype 2

The manufacturing process started with the main body of the truck, referred to as the hanger. Four setups were required; the first three using a standard machining vice and the fourth using the same plate fixture that was made for manufacturing prototype 1. The only highly critical operation was boring the hole for the tie rod to pass through. Since there would be radial load of nearly 400 pounds on the tie rod, it was crucial that the bore be made only a couple ten-thousandths of an inch oversize. Furthermore, the bore needed to be machined straight within a couple of tenths as well. To do this, the hanger was placed on the vertical fixture and rigidly secured. A drill with a diameter of 95% of the final feature size was then used to make the pre-hole. A reamer with a diameter of .0004" greater than the diameter of the tie rod was used to finish the hole. This yielded excellent results with no noticeable slop between the tie rod and the hanger bore.

Figure 21: Finished Hangers

The knuckles are the most complicated component of the whole design, both in function and in geometry. By spending a lot of time on the knuckle design, a design was created that could be machined in only two setups with no special fixtures; a significant improvement over the first design in both time and cost. The first operation was a turning operation out of 1018 steel round stock. During this operation, the axles were accurately turned and threaded. For the second operation, the axle was placed into a 5C collet 4th axis dividing head. All other features, including the steering arm and the kingpin bearing, were machined on that setup. One of the significant benefits of that process design was that since the axle was located by the collet's axis, all of the critical features were referenced very accurately. This translated to increased quality of the final product and less variation.

The tie rod was the next component to be machined. The design for manufacturability permitted this part to be entirely machined on a single set up (aside from center-drilling for use with a tail-stock) using the 4th axis dividing head in the CNC mill. The first sequence turned down the ends to a diameter of .3125” and also milled the flats and drilled the holes for the intermediate link pins. A reamer with a diameter of .0001” oversize of the precision ground intermediate link pins was used. The most critical sequence was the creation of the socket for the ball stud because if the diameter is too large, the rider would be able to sense excessive slop and the wear rate would be drastically increased. Therefore, great care was taken to ensure that the diameter was machined to no more than .001” larger than the ball diameter of the ball stud. At this accuracy, zero noticeable slop existed between the ball stud and the tie rod socket. Another key aspect of the tie rod fabrication was creating the cam surfaces for the return to center mechanism. Since the only critical aspect of the cam surfaces is smooth surface finish, a 6 flute finishing end mill operated at an inflated surface speed was utilized on the finishing pass.

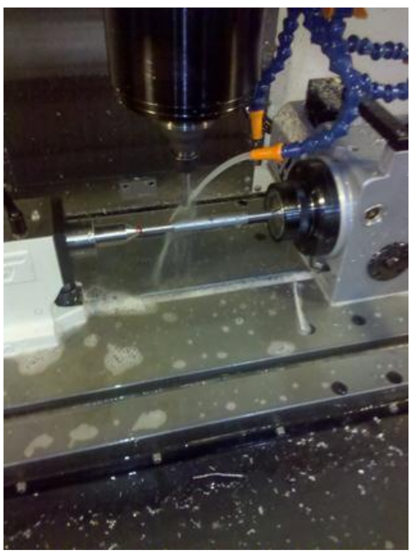

Figure 22: Tie Rod Milling

Once the tie rod was finished, the basic components of the steering linkage were assembled to check for fit and finish. First, the pivot axle bushing was pressed into the hanger using an arbor press. The knuckles were then assembled utilizing grade 8 kingpins and nylon thrust bearings. Then the tie rod was sufficiently lubricated using general purpose automotive wheel bearing grease, and slid into the cross-bore in the hanger. The intermediate links, simply roller chain connecting links, were lightly greased and assembled to the tie rod and both knuckles. At this point, the basic steering linkage was assembled.

The next step in the build process was to manufacture the remaining components of the return to center mechanism. This included machining both the spring adjusters and the pistons which follow the cam surface milled into the tie rod. Since the piston needed to slide in and out of the bore of the spring adjuster very accurately, the decision was made to bore the adjuster first. This was because it was easier to fine tune the diameter of the adjuster than the spring adjuster bore. The spring adjusters were machined out of standard 5/8-18 x 1.0" grade 5 bolts. The first task was to turn the head of the bolt down until completely round. This allowed clamping in the chuck in preparation for boring. A pre-hole was drilled through the adjuster, and followed up with a standard carbide insert boring bar. Surface finish was critical so speeds and feeds were adjusted accordingly for the finish pass. Next, each piston was turned out of bearing bronze and custom fitted to their respective adjuster. Once turned, each adjuster was placed in a 5c collet dividing head so that the end could be machined to match the tie rod cam profile, thereby maximizing surface area. Once completed, fifteen belleville disc springs were placed in each adjuster. Each piston was greased and placed into the adjusters prior to the adjusters being threaded into their respective holes. At this point, the trucks were completely assembled less the baseplates.

The baseplates were the final components to be machined for each truck, and with the least amount of critical features. They required three setups each, all done using a standard vice. Once machined, thread sealant was applied to the ball studs and then screwed into each of the baseplates. Then the baseplates were positioned over the pivot axle bushing previously pressed into the hanger. The pivot axle was then passed through the pivot axle bushing and screwed into the baseplate. At this point, both trucks were completely assembled and ready to be bolted to the deck for testing. Below is a photo of the finished trucks assembled to the skateboard deck.

Figure 23: Finished Trucks, Assembled to Deck

Testing of Prototype 2

In order to determine whether prototype 2 was successful at meeting the established performance goals of improved handling and stability, riders of various skill levels were allowed to ride the trucks and provide feedback. Since the success of the project is measured subjectively it was important to have a large enough sample of riders. Since the trucks had still not been patented at the testing stage, however, testing was limited to only friends that could be trusted to withhold specific information about the technology. Furthermore, riding was limited to areas with low pedestrian traffic to further protect the technology.

Figure 24: Downhill Testing

Figure 25: Carving Testing

Aside from determining whether the trucks were more stable and provided better handling, the purpose of testing the second prototype set was also to determine what, if any, improvements had been made over the original design. The results of the testing stage are outlined in the Results/Discussion section of this report.

Results and Discussion

After extensive field testing, described in the Methodology section, the following results were obtained for both prototype designs. The first section highlights some positive characteristics and attributes while the second section includes issues that need to be addressed for future development.

Positive feedback and Discussion, Prototype 1:

- The trucks handle very well and traction is definitely increased. In the event of a slide, the wheels maintain just enough traction so that the board remains in control.
- The trucks are very low which makes pushing extremely easy because the rider does not have to step down as far to make contact with the ground. This also noticeably increases the stability.
- The trucks are insensitive to lean, boosting confidence in downhill. The ability to lean at 25 degrees, though, means that maneuverability is not sacrificed. Furthermore, the rider's ankles are kept closer to a 90 degree angle with their lower legs making toe side and heel side maneuvers much easier on the rider's ankles.
- Wheel bite is a non-issue because the wheels never swing underneath the deck.

Negative Feedback and discussion, Prototype 1:

- The trucks are too heavy overall. Pushing from a stand-still is noticeably more difficult than a standard longboard and carrying the board takes more effort.
- There is too much slop in the linkage system. For very small changes in deck angle the wheels do not steer. This can be unnerving in downhill situations where very small adjustments are required.
- The return to center mechanism is completely un-satisfactory. Many riders stated that the spring rate was too light and that the board felt “dead”. This means that it is hard to make quick carves back and forth because the rebound force is not large enough. Furthermore, the adjuster screws had little effect on the board’s feel.

Once enough feedback was collected, the trucks were disassembled so that all of the parts could be inspected for wear. All of the components were fine except for the actuation pin and the slot in the baseplate. Those measured at about .005” clearance, explaining, in part, the issue with slop. As far as manufacturing is concerned, the trucks have too many components and too many fixtures required and simply not marketable in their current design. All of this information was compiled and used to improve the design of Prototype 2.

Positive Feedback and Discussion, Prototype 2:

- There was a noticeable increase in traction, both over current longboard technology as well as prototype 1. This is most likely attributed to the more precise steering linkage angles, causing less scrub to occur
- Weight reduction. Pushing effort was noticeably better in comparison to the first set of prototypes. In fact, each truck weighed 0.3 pounds lighter. However, they still remain nearly 0.5 pounds heavier than a standard set of longboard trucks.
- Stability was comparable to prototype 1 and slightly better than conventional trucks. High speed instability still seems to occur, but some riders said that the added control and the precision of the trucks allowed for them to deal with it much more confidently. Positive caster may be introduced by wedging both of the trucks. The likelihood of this increasing stability is high, but remains untested.
- Return to center mechanism was greatly improved. Riders noted that quick carving and slalom style riding was made very easy because the rebound force helps the rider to make the transition between turning directions with less effort. One rider stated that the trucks felt more “alive” than the first set.
- Improved aesthetics. Many riders noted that the trucks looked less “clunky” and more high performance oriented than the first set.

Negative Feedback and Discussion, Prototype 2:

- Trucks still too loose for high speed downhill situations. This is not a severe problem, though, because stiffer springs can be purchased and easily installed.

Furthermore, not much experimentation was done with the spring orientation which can be used to adjust the spring rate as well.

- Twitchy maneuverability at speeds greater than 25 miles per hour. Some riders stated that the trucks exhibited a certain tendency to weave at elevated speeds. Some riders explained that it was most likely due to the looseness of the trucks- an easily correctable problem.

After about 20 hours of total test time, the trucks were disassembled and the components were checked and measured for wear. All of the critical components displayed zero noticeable wear, but one worrisome observation was made. The greased ball and socket mechanism was infused with dirt, sand, and other debris which could lead to unwanted and accelerating wear during extended use.

Other Considerations

From an engineering perspective, one of the main flaws of the design is that during turns, the ball stud experiences very high stresses. This in turn increases the likelihood of wear between the ball stud and the socket. Furthermore, having the spring loaded pistons acting on the tie rod means that the bore must be very accurately machined and that grease is a necessity. In hindsight, the better approach would have been to reposition the return to center mechanism such that it acts on the baseplate. This would have greatly reduced the stress on the steering linkage while simultaneously increasing the service life of the truck.

Conclusion

This section is an outline the future plans of this project, including a brief discussion of the current detail design of prototype 3. Following that, a summary of the project is given.

Future Plans

The immediate plan for the future consists of first filing for provisional patent status so that a broader range of riders may be able to test the trucks. This will generate new feedback which will be used to further develop the truck as a marketable product. A third prototype is currently being designed which addresses the issues that were revealed from testing prototype 2. For example, the return to center mechanism was redesigned so that the spring operates against a cam surface milled into the baseplate as opposed to the tie rod. This should drastically decrease the stress on the ball stud as well as eliminate or greatly reduce wear related issues. Some other innovations and improvements include:

- Zero steer rear truck capability. In downhill situations, the rider has the option to make the rear truck not steer- only lean. Therefore, the steering sensitivity will effectively be decreased by 50 percent. Furthermore, based on ground vehicle dynamics theory, front wheel steering is much more stable than four wheel steering above 20 miles per hour.

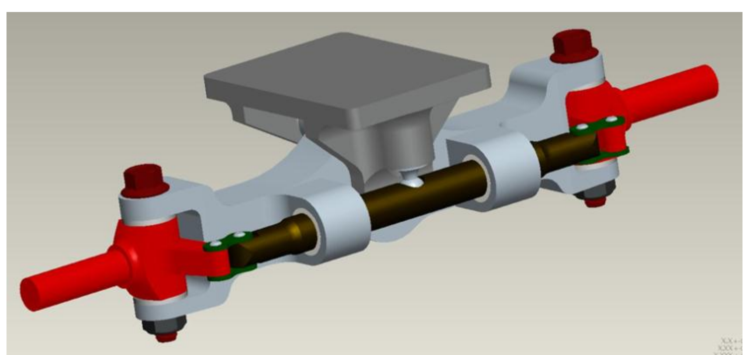

Figure 26: Prototype 3, Initial Design

- Steering ratio adjustability. The rider will have the option of adjusting the steering ratio. For carving, a higher turn to lean ratio is better. This opposite is true for downhill.
- Option to cast the baseplate and Hanger. This would potential lower the manufacturing cost for elevated production volumes [See Cost Analysis].

While some problems still exist, the initial goal of improving the skateboard handling and stability characteristics by applying automotive steering principles was successful. The trucks provide the rider with unparalleled performance when compared to standard solid axle trucks. Second, the work started during the project is an excellent platform for future development and progression.

Project Summary

The goal of this project was to develop a longboard truck that exhibited superior handling and stability characteristics in comparison to products that are currently available. In order to do this, performance enhancing characteristics employed in automobile steering were applied to the design of the trucks. The trucks were designed, built, and extensively tested to determine their benefits and drawbacks and comparisons were made to current technology. Using the feedback generated during testing, the trucks were redesigned to resolve the issues. Furthermore, DFM, DFA, and manufacturing cost reduction played a significant role because the potential for marketability was realized.

The second prototype set was extensively tested by a number of riders under various conditions. Mostly positive feedback was generated and the trucks performance was clearly superior to currently available products. The main benefits of the truck included unparalleled traction, quick carving ability, and predictable handling at elevated speeds. Problems, while small, were still pointed out. These problems, however, were mostly design and manufacturing related problems that did not have a direct effect on the performance of the trucks. Because of this, the project is considered to be successful. The problems with current technology were identified and a corresponding solution to those issues was created.

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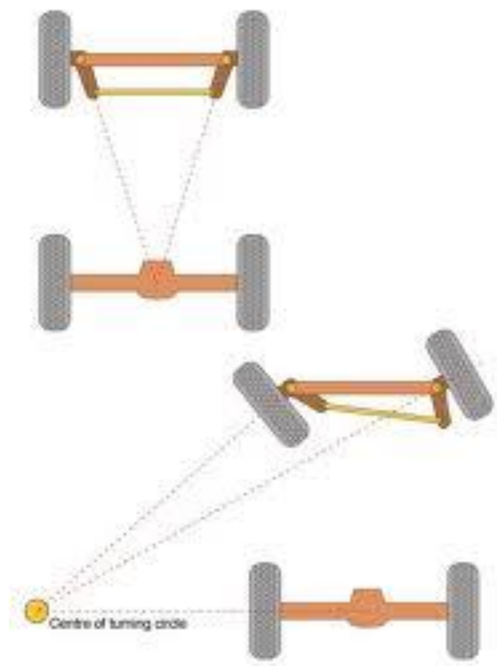
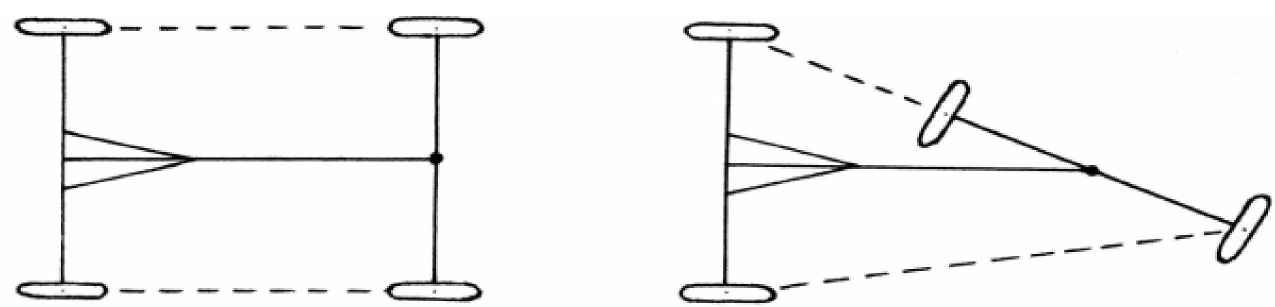
Appendix



Ackermann Steering Condition [3]



Turntable Steering [3]

W	theta	phi	Fb	D2	Fsp	T1	L	Fr	D3	X
10	0.3	1.13	30	0.75	341.36	0.7	1.14	200	0.2	0.53
	17	65			170.67					
										30.37

W	Board Width	inches
Theta	Knuckle Rotation	radians
Phi	Board Rotation	radians
Fb	Lean Force per Spring	pounds
Fsp	Spring Force	pounds
Tl	pivot to TR dist.	inches
X	cam surface angle	radians
D3	spring displacement	inches
Fr	Force on tie rod	pounds

Excel Model Used to Determine Design Parameters for Prototype 2