Adapting a variable stability mechanism for a tilting tricycle from the delta to the tadpole wheel configuration

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Adapting a variable stability mechanism for a tilting tricycle from the delta to the tadpole wheel configuration
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Abstract:

We previously presented a narrow-track tilting tricycle with a variable stability mechanism integrated between the swing arms that support a pair of rear wheels, in the so-called "delta" configuration, and with recumbent seating. We now examine adopting that variable stability mechanism to work on a tricycle with a split-parallelogram linkage between a pair of front wheels, in the so-called "tadpole" configuration, and with upright seating.

It was fairly straightforward to allow for tilting by replacing the front wheel and fork with a split parallelogram comprising two paired A-arms and kingpins, controlling the motion of the two halves with a bell crank and two tie rods, and then varying the handling of the vehicle by moving the connection point of the tie rods on the bell crank, just as we did with the swing arms of the previous vehicle.

We have also separated the two tasks of positioning the tie rod ends on the bell crank and enforcing symmetry of the tie rods. The former does not require much force and can be easily implemented with Bowden cables, but the latter does require large forces and is better implemented with a local rigid-bar linkage.

Implementing decent Ackermann steering geometry, allowing for both large tilt and steer angles, and decoupling tilting from steering, however, proved to be quite a challenge, at least while we attempted to implement it with bar linkages. Fortunately, we discovered a 2006 paper by Prof Drstvenšek et al. describing a Bowden cable and cam system that looked promising.

Finally, the resulting vehicle handles very nicely. When in "full bicycle" mode, it handles quite similar to the original bicycle that we had converted into the tricycle. When in "rigid tricycle" mode, it keeps the rider upright when stationary or when riding at a walking pace. In between these two extremes, it handles even better than the original bicycle in a slalom course and when slowly following a straight line.

Keywords: Tilting Tricycle

Introduction

Our motivation for implementing a tilting tricycle at all is not only to provide the best of both worlds: hold the rider upright when stopped or moving slowly, as a rigid tricycle does, and lean into turns when moving faster than walking speeds to allow a narrow axle track and normal seat height without danger of roll-over accidents, as a bicycle does. We also seek to help the rider maintain balance when riding between these two extreme modes.

We have previously explored a promising mechanism for achieving these three goals with a vehicle that has recumbent seating, the so-called delta wheel configuration, with one wheel in front and two wheels at the back, and front-wheel drive by means of a so-called "moving bottom bracket" with the crank mounted to and moving with the front fork.

When we proposed pursuing this work further to Royal Dutch Gazelle, a supporter of this project, they advised us that recumbent seating is not very popular with European riders and that delta wheel configurations are thought to look antiquated and geriatric. Thus, our goal was to recreate the functionality of the variable geometry tilt mechanism in a vehicle with upright seating and tadpole wheel configuration.

Since rear-wheel steering is problematic at best for tilting vehicles, the two wheels in front must be steered, and since bicycle wheels are not designed to withstand large side loads, the two wheels in front should tilt with the vehicle to keep the load they bare mostly in the plane of the wheel.

Finally, a visit to a cargo bike festival, at which we were able to test ride several prototype and commercially-available tilting tricycles, highlighted that the situations that arise when encountering the limits on steering or tilting angles were awkward at best, requiring the rider to put a foot down, or dangerous at worst, leading to a roll-over accident. Therefore, we strove to implement the largest steer and tilt angles that we could.

Methodology

It was straightforward to retrofit the variable-width bell crank and tie rods to the split-parallelogram tilting mechanism supporting the pair of front wheels. By mounting the bell crank low and connecting the tie rods to the upper A-arms, we were able to keep the tie rods in tension and avoid buckling issues.

We also improved the control mechanism for the variable-width bell crank. In the first vehicle, the two functions of positioning of the tie rod ends and enforcing the symmetry of that positioning were both implemented with the same Bowden cable system. This has proven problematic, however, because the enforcement of symmetry requires large forces when the bell crank rotates to enable tilting of the vehicle. These forces are related to the weight of the rider and beyond the capacity of the bicycle brake cables originally used to counter them. Therefore, we separated these two functions, and developed a rigid-bar linkage to enforce symmetry, which then left the relatively low-force task of positioning the tie rod ends to perfectly sufficient Bowden cables.

The rigid-bar linkage consists of two links and a slider on a rail. One end of each link is connected to one of the tie rod ends, and the other end of each link is connected to the slider, which is constrained to slide on a rail mounted orthogonally to the center of the bell crank. Thus, the tie rod ends are forced to maintain perfect symmetry about the center of the bell crank, and the rigid links easily handled the loads applied to them. These links, the slider, and the rail can all be seen mounted to the top edge of the bell crank in figures 7 and 8 below.

Finally, we needed a way to steer the two front wheels that would create correct Ackermann geometry, allow for large steer and tilt angles, and decouple steering from tilting to avoid bump steer. An investigation of steering systems implemented with rigid links, as on most automobiles, revealed that the goals of correct Ackermann and decoupled steering and tilting are incompatible. Plus, rigid links cannot provide a range of steering angles approaching 180° for danger of two links becoming colinear and thereby creating a mechanical singularity.

Fortunately, we discovered a 2006 paper by Prof. Drstvenšek et al. describing a Bowden cable and cam steering system for a human-powered vehicle that looked promising. We could not follow the published derivation of the kinematic relationship between steering angle and cam radius, so we derived our own, as shown below in Equation (1)

$$r_\ell = r_c\theta_c/\theta_\ell = r_c\theta_c/\mathrm{acot}(\cot\theta_c - \tfrac{1}{2}at/wb) \text{ and } \to r_r = r_c\theta_c/\theta_r = r_c\theta_c/\mathrm{acot}(\cot\theta_c + \tfrac{1}{2}at/wb) \tag{1}$$
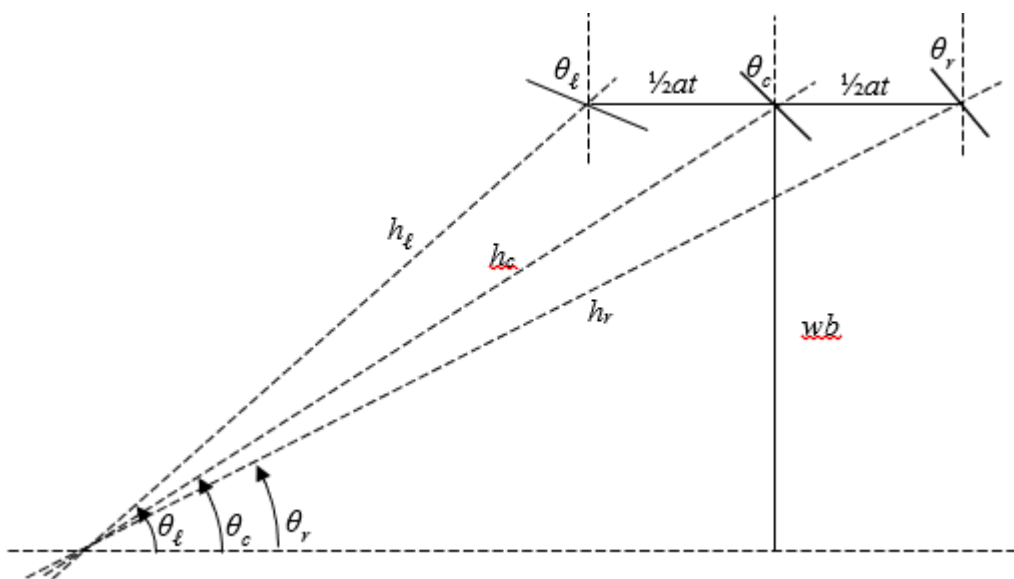
where

r_ℓ, r_c, and r_r are the radii of the left, center, and right cams, respectively.

θ_ℓ, θ_c, and θ_r are the rotation (steer) angles of the left, center, and right cams, respectively.

at is the distance between the two steered wheels, the "axle track"

wb is the distance between the front and rear axles, the "wheelbase"

These equations can be derived from Figure 1 as follows:

Figure 1. Ackermann steering geometry.

First, use the wheelbase, the rear wheel axel axis, and the front wheel axel axes to define right triangles that meet the Ackermann condition: the axel axes all intersect at one spot.

Then, relate the left, center, and right steer angles, θ_ℓ, θ_c, θ_r, by introducing and them eliminating the hypotenuses h_ℓ, h_c, h_r.

$h_c\sin\theta_c = wb \to h_c = wb/\sin\theta_c$
$h_\ell\sin\theta_\ell = wb \to h_\ell = wb/\sin\theta_\ell$
$h_\ell\cos\theta_\ell + \tfrac{1}{2}at = h_c\cos\theta_c$
$wb\cos\theta_\ell/\sin\theta_\ell + \tfrac{1}{2}at = wb\cos\theta_c/\sin\theta_c$
$wb\cot\theta_\ell + \tfrac{1}{2}at = wb\cot\theta_c$
$wb\cot\theta_\ell = wb\cot\theta_c - \tfrac{1}{2}at$
$\cot\theta_\ell = \cot\theta_c - \tfrac{1}{2}at/wb$
$\theta_\ell = \mathrm{acot}(\cot\theta_c - \tfrac{1}{2}at/wb)$

$h_c\sin\theta_c = wb \to h_c = wb/\sin\theta_c$
$h_r\sin\theta_r = wb \to h_r = wb/\sin\theta_r$
$h_r\cos\theta_r - \tfrac{1}{2}at = h_c\cos\theta_c$
$wb\cos\theta_r/\sin\theta_r - \tfrac{1}{2}at = wb\cos\theta_c/\sin\theta_c$
$wb\cot\theta_r - \tfrac{1}{2}at = wb\cot\theta_c$
$wb\cot\theta_r = wb\cot\theta_c + \tfrac{1}{2}at$
$\cot\theta_r = \cot\theta_c + \tfrac{1}{2}at/wb$
$\theta_r = \mathrm{acot}(\cot\theta_c + \tfrac{1}{2}at/wb)$

Finally relate the motion of the center, driving pulley, to the left and right cams by their radii:

$$r_c\theta_c = r_\ell\theta_\ell = r_r\theta_r \to r_\ell = r_c\theta_c/\theta_\ell = r_c\theta_c/\mathrm{acot}(\cot\theta_c - \tfrac{1}{2}at/wb) \text{ and } \to r_r = r_c\theta_c/\theta_r = r_c\theta_c/\mathrm{acot}(\cot\theta_c + \tfrac{1}{2}at/wb), \text{ as above.}$$

These equations produce discontinuities, of course, when the argument to arccotangent() changes sign, but we handle this in the MATLAB implementation to generate the following smooth curves, as shown in Figure 2.
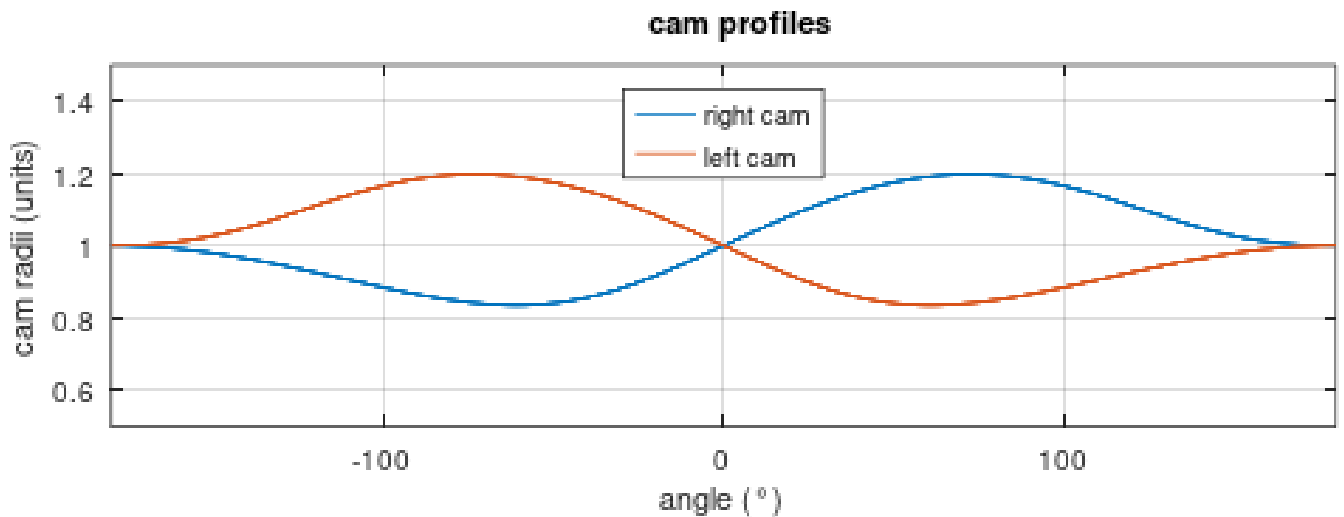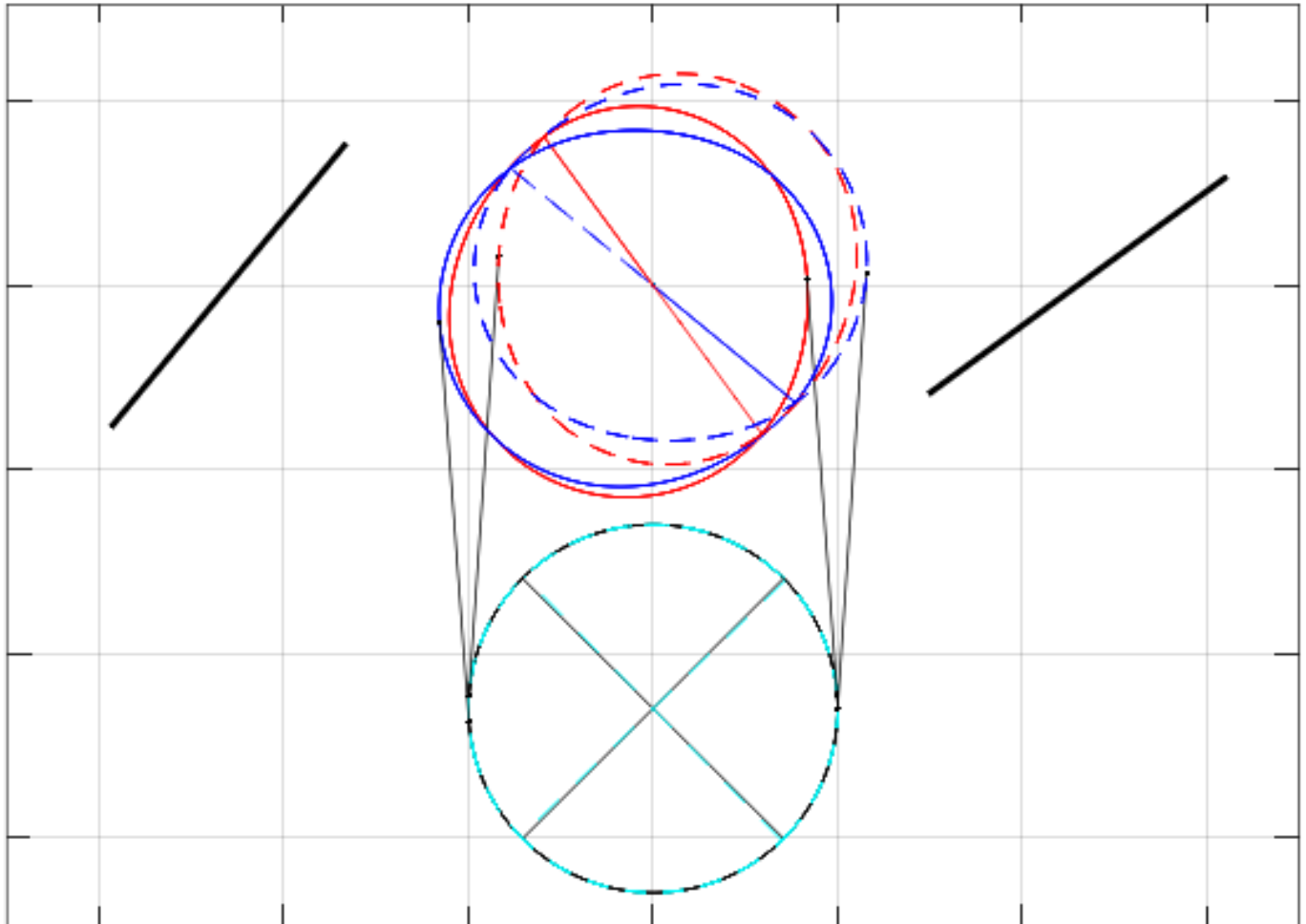

Figure 2. Steering cam profiles.

These cam radii generate the desired Ackermann steering angles, as shown in Figure 3, at least in theory.
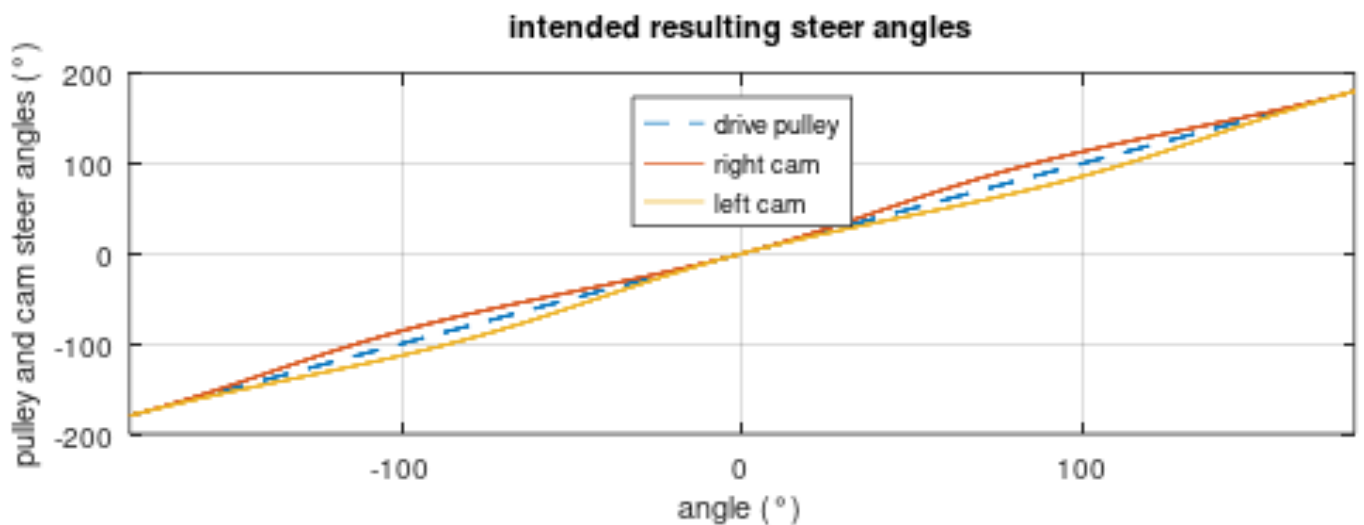
Figure 3. Resulting steer angles.

Figure 4, below, shows how the pulleys, cams, and cables may be arranged. In the actual, physical implementation on the protype tricycle, the left and right cams are separated and attached directly to the left and right kingpins, respectively, and the cables are routed through flexible Bowden cable housing from the centrally located drive and idler pulleys to the left and right wheels.

**Figure 4.** Schematic diagram of drive and idler pulleys at bottom, left and right input and output cams at top center, right steered wheel at top right, and left steered wheel at top left, steered to a 45° right turn.

An issue with this whole design, however, is that it ignores how the point at which the cable contacts the cam moves slightly around the cam as the cam radius changes, and so the cam at these nearby points does not have exactly the intended radius. This additional detail causes small errors in the resulting steer angles which we extract from a numerical simulation, as shown in Figure 5. It can also be seen that the differences are not perfectly symmetrical and that the left side follows a slightly different track when returning from the right-hand turn than it did as the turn was made. Thus, the blue line appears double between 25° and 50°.

Methods that we tried to correct for these small errors include: simply using the error calculated for the theoretically ideal cam profiles as a correction factor to reshape them, and forcing the cables to contact the cams at the 3 o'clock and 9 o'clock positions. Neither of these made a noticeable improvement, and we suspect that the issues are that the first does not take into account where the cable actually contacts the cam, and the second artificially distorts the length of the cable because it is no longer tangent to the cam at the point of contact.
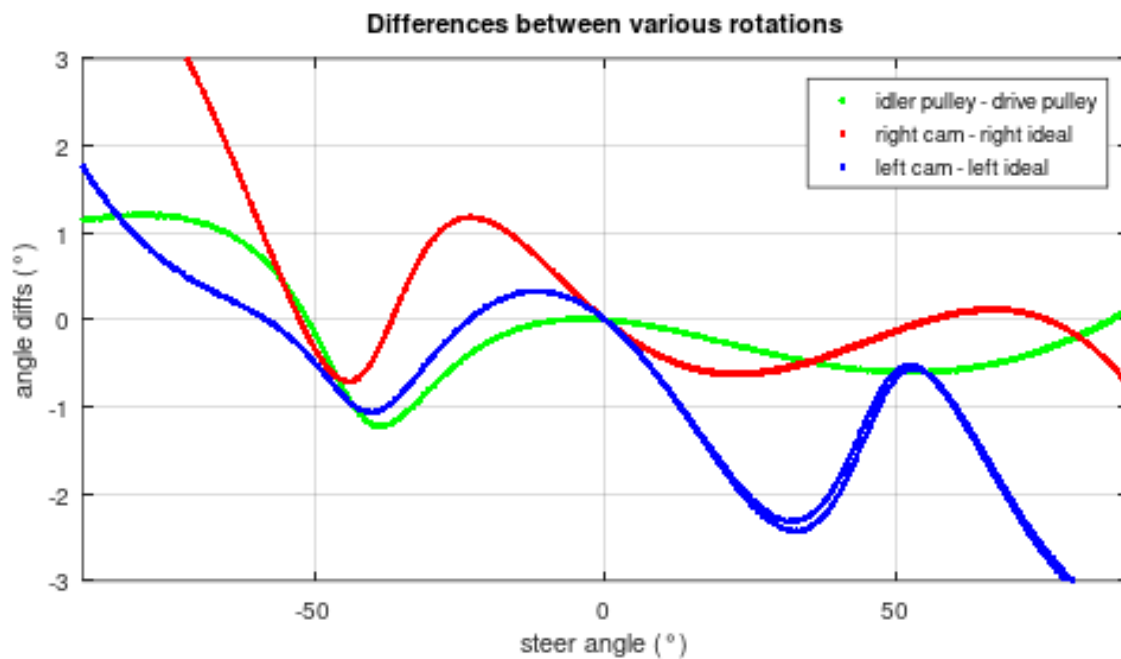
Figure 5. Differences between intended and actual angles, according to numerical simulation.

After trying to correct this error by several methods, without success, we were eventually forced to accept it due to the time-limited nature of the project and its funding.

We 3D printed prototype cams, as shown in Figure 6, to confirm that the system would work as designed and that the cables would generate sufficiently low friction, and it took us two tries to meet this goal. Our first attempt used generic bicycle brake cables and housing, which generated far too much friction. When we switched to high-end, coated cables and lined housing to eliminate metal-on-metal contact, the situation was greatly improved.
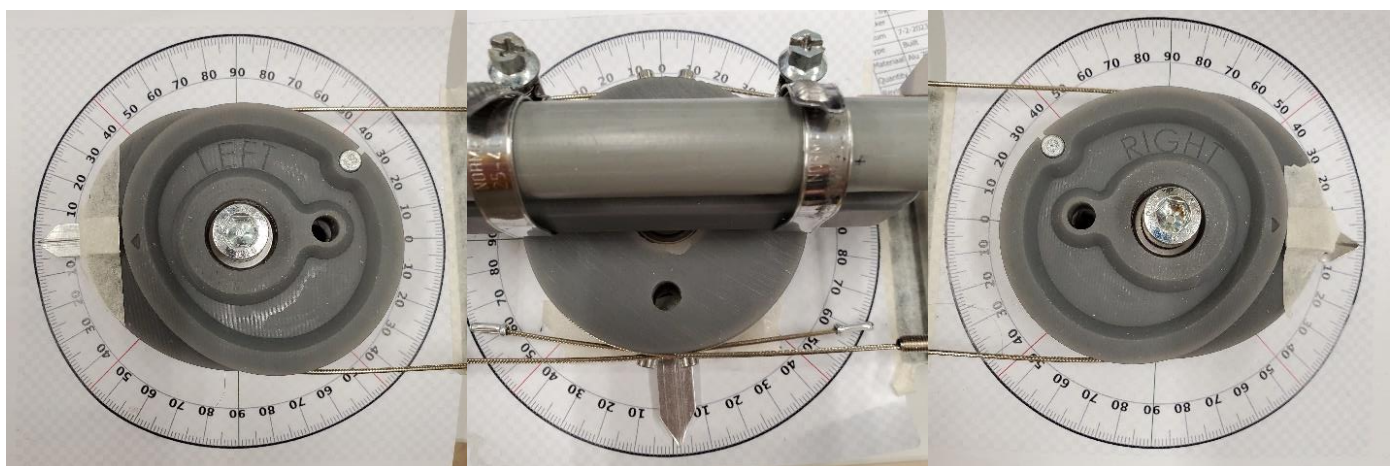

Figure 6. Images of the cable and cam steering system 3D-printed prototype.

These images of the 3D printed prototype cams also illustrate the error described above. If all the cams had constant radii, then the cable and the radial line at which it contacts the cam would be orthogonal. If the cams do not have constant radii, and the cam radius

is changing rapidly enough in the vicinity of where the cable contacts the cam, the cable will contact the cam at a point to which the radial line is no longer orthogonal to the cable.

Nevertheless, we forged ahead to keep on our tight schedule, and the bicycle we chose to convert into a tilting tricycle is a Gazelle Ami C7, a modern take on the traditional Dutch city bike.

With the key objectives and components selected, the rest of the design consisted of many compromises to minimize axle track while maximizing tilt and steer angles. Figure 7 shows the resulting solid model, and Figure 8 shows the resulting physical vehicle, which has an axle track of 750 mm, a maximum tilt angle of 48°, and a maximum steer angle of 45°.
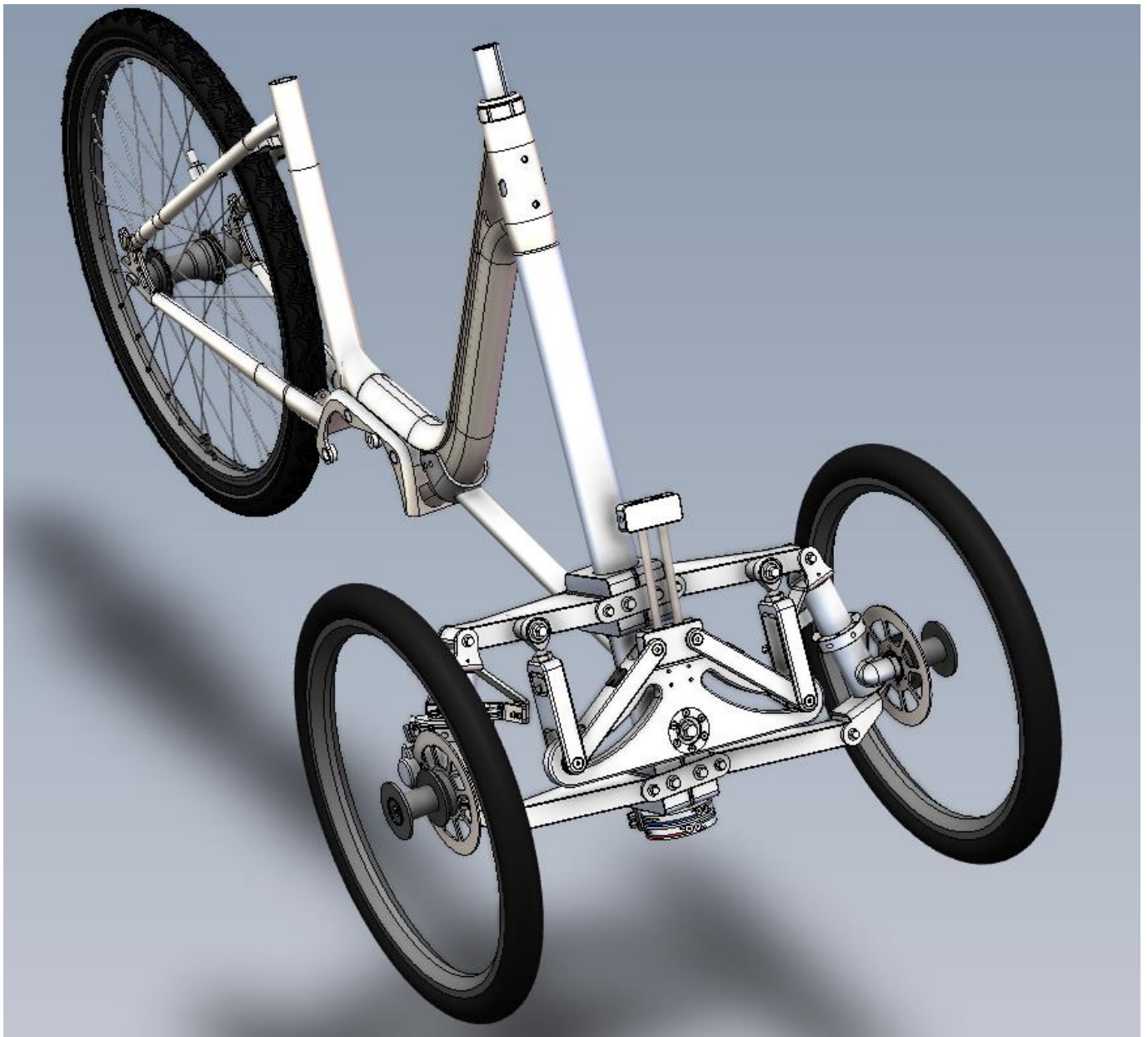
Figure 7. Final design solid model.

The resulting physical prototype, when in the "bicycle" configuration, handles nearly as well as the original bicycle, and, when in the "rigid tricycle" configuration, keeps the rider upright without need for feet on the ground. When configured halfway between these two extremes, it handles better, based on a handling metric developed in a parallel project, than any other bicycle or tricycle we tested.

The handling metric mentioned above was developed by Floris van Willigen as part of his master's thesis at TU Delft, and he finalized this metric before this tilting tricycle was available for testing. He focused specifically on commuting and so, from his review of the literature, he chose two riding tests to perform:

1. a slalom course ridden at a "moderate" speed

2. and following a straight line at a "slow" speed.

During these tests, IMUs rigidly attached to the front and rear frame record yaw and roll rates which he then integrated to calculate yaw and roll angles. From those values, Mr. van Willigen calculated three handling metrics:

1. maximum yaw factor, which is the yaw rate divided by the steer angle, a measure of the steer response of the vehicle,

2. the mean absolute steer angle, a measure of the low-speed balance of the vehicle,

3. and the time delay between the roll rate and steer rate, a measure of the responsiveness of the vehicle to rider inputs.

He collected data from multiple trials, rejected outliers, and performed a careful analysis to ensure that the results are statistically significant. Finally, he scaled the values and combined them into a single score.

He performed these tests on four bicycles and three other tilting tricycles, and when the tilting tricycle created for this project was configured halfway between a bicycle and a rigid tricycle, it produced the best score.

Conclusion

A tilting mechanism with variable geometry, which had previously been demonstrated on a tricycle with recumbent seating and delta wheel configuration, has been successfully adopted to a tricycle with upright seating and tadpole wheel configuration.

A steer-by-cable mechanism has been implemented to provide good Ackermann steering geometry, large tilt and steer angles, and decoupled tilting and steering to avoid bump steer.

The resulting physical prototype handles as hoped and scored well on a handling test that was developed before the physical prototype became available.

Acknowledgements

Figure 8. Final physical prototype.

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