Appendix

1. Introduction

As mentioned in the thesis report, the main goal of the project was to design a mechanism for decoupling the energy storage and release in passive foot prostheses. Whereas the main goal was to decouple the energy storage and release capabilities, this research mostly focused on using the VSPA foot cam-based transmission [1] to implement the energy recycling concept [2]. This is depicted in figure 1 below.

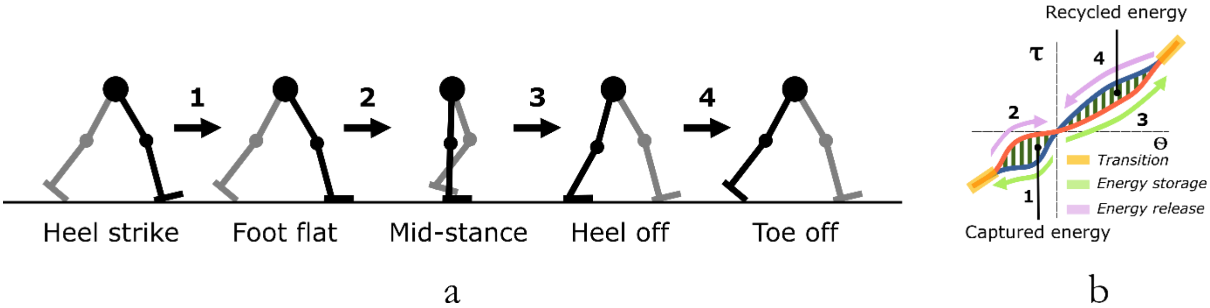


*Figure 1: **A)** Key phases during the stance phase. **B)** Desired torque angle curves for the stance phase*

2. Conceptual design

a. Requirements

Before the mechanical design of the Decoupled Energy Storage and Return (DESR) foot, certain design requirements were set up. This enabled the streamlining of the design process and set boundary conditions for both the functionalities of the device as well as the manufacturing and assembling processes. Below we can find a table of these design requirements.

Dual cam-follower transmission	The main goal of this project was to use multiple cam profiles in the VSPA-Foot cam-based transmission. This cam-follower transmission effectively decouples the ankle joint mechanics from the leaf spring mechanics, allowing the modulation of any arbitrary torque-angle curve at the ankle joint. By using two cam profiles the energy storage and energy release can be decoupled as well. The torque-angle path used to store the energy is not necessarily the same path required to release the energy, if the total energy stored or released is at most equal. This idea is depicted in Figure 1.
Continuous energy release	The use of multiple cam profiles requires the ability the switch the contact of the cam follower between the different cam profiles. This should switch should be non-noticeable and keep the path of energy storage and release gradual. Nor should it add additional energy losses that might diminish possible effects of decoupling the energy storage and release.
Quasi-passive	Despite the stiffness variability of the VSPA foot not being a key-phase of the dual cam-follower transmission, it is preferred to keep this in the DESR design. Not only does it allow for the stiffness variability in order to perform multiple tasks, it would also allow a fair comparison between using a single cam profile versus two cam profiles, in order to test the benefits of the energy storage and release decoupling concept.
Foot clearance	The main purpose of decoupling the energy storage and release was to enable the energy recycling concept in the VSPA-foot cam-based transmission. During

	the early stance phase, more energy is stored than released by means of plantar flexing, meaning that spring will be in a bent position. This plantar flexion should be minimized in order to ensure that no compensatory gait is necessary. In the ideal case the second cam profile is designed such that it accounts for this energy storage and the deflection of the spring at mid stance.
Durability	The device should not undergo any plastic deformation. The mechanism for the dual cam should be designed such that it is strong enough to endure the loads during the desired torques.
Customizability	The new mechanism should have the ability to easily replace components in order to allow for patient specific customizability as well as achieve a modular design. A patient specific design can be achieved by either designing a primary torque angle curve around a patient's preferences, or by changing the slider stiffness around a pre-set primary slider position in order to achieve the preferred stiffness values of the amputee.
Modular design	Besides the befits of being able to switch out cam profiles for patient preferences, the easy of (dis)assembly also proves useful in terms of reproducibility of an experiment if a fellow researcher wants to reproduce the experiment or pick up on an ongoing project. Besides the reproducibility, a well thought out assembly sequence provides the ability to easily repair the device with repair parts, and it requires a low threshold learning curve for understanding and assembling the device.
Range of stiffness values	The range of stiffness values should be within the preferred range of the amputees. As patient specific preferences can vary, this range needs to account for a between-subject variance of preferred stiffness values.
Independent dorsiflexion and plantarflexion stiffness values	One of the reasons the VSPA-Foot cam-based transmission was used is the ability to generate any arbitrary torque-angle curve. This also means that the curve can differ significantly in the controlled plantar flexion and the controlled dorsiflexion. This is useful, as the torque-angle curve in able-bodied gait differs significantly during these two stances as well. For the DESR foot it is in useful to have different torque-angle curves in the plantar and dorsiflexion for two reasons: • The range of motion of maximum plantar flexion and dorsiflexion differ significantly. If the DESR foot is designed with multiple cam profiles, the transition in plantar flexion will occur at a different angle than at dorsi flexion. • The stiffness slopes differ for both the plantar flexion and the dorsiflexion during gait. In order to recycle energy, the energy storage during the early stance phase should equal the energy release during late stance phase. This should also account for the differences in the maximum range of motion as mentioned in the previous bullet point. Therefore, the curves in the bottom left quadrant of FIGURE 1b should have a steeper slope than the curves in the right top quadrant as both the absolute maximum torque values are limited as well as the maximum allowable range of motion in order to transition between the cam profiles.
Mass	If possible, the design should be as light as possible, with a maximum of 1,5 kg. Higher masses introduce asymmetry between the intact and residual leg, increase the balance related effort of amputees and overall increase the energetic cost due to a distally placed increased mass of the prosthesis.
Biomimicry	The placement of the ankle joint rotation point should be representative of the human ankle joint. By placing rubber plates under the DESR foot, damping can

	be introduced, along the ability to vary ankle joint height in order to line out with the intact leg ankle joint and the ability to adjust the roll-over shape of the foot. The torque-angle curves should be modelled in the range of the human joint ankle capabilities, preferably around the human ankle torque angle curves of controlled dorsi- and plantar flexion. The ability to vary around this stiffness slope can account for the between subject stiffness preference settings.
Ability to compare	The DESR foot already can switch between cam profiles to account for patient preferences. It should, however, also be able to operate with a single cam-profile, allowing the ability to compare the single cam profile versus two cam profiles. This might provide a good indication of the relevance of the decoupled energy storage and release concept of the DESR foot, whilst focusing on energy recycling.
Inverse model	Going from the desired torque-angle curve to the cam profile shape is the forward model, whereas, predicting the torque-angle curve from that fixed cam profile shape at a different stiffness value than the primary stiffness is the inverse model. The mathematical model of the DESR should have an inverse model for two reasons: • A change in the slider position under the spring doesn't merely change the stiffness slope values, but also the amount of captured energy to recycle throughout the gait. The inverse model helps with predicting the amount of energy that can be 'recycled' by adapting the slider position. • By implementing an inverse model, possible math errors in the forward model can be discovered. It acts as a validation to whether the forward model calculations are correct. This can be confirmed if the inverse model at the primary stiffness values outputs the same torque-angle curve as the initial input in the forward model.
Range of motion	The range of motion for level ground walking was discussed thus far, where a minimum 5 degrees plantar flexion and a minimum of 10 degrees dorsiflexion was required for transitioning between the different cam profiles. For performing different tasks such as stairs ascent and descent, a larger range of motion is required. This can be achieved by adjusting the slider position under the spring. But to ensure that decreasing the spring stiffness value does not require too much compensation by knee joint, hard stops of the ankle joint are set in place, with allowing a maximum range of motion of 20 degrees in the plantar and dorsiflexion.
Ensuring transition	In order to ensure the energy recycling concept during every stance phase of the stance gait, transitioning at both the plantar flexion and dorsiflexion should occur. The angles of 5 degrees plantar flexion and 10 degrees dorsiflexion were chosen by looking at experimental data of amputees [3] walking with the VSPA foot.
Off-the-shelf-components	In order to reduce the cost, designer work load, design complexity, design time and to increase the likelihood of success, it is desired to use off-the-shelf components in the design if possible. The bearings at the ankle joint center of rotation and the cam follower are examples of these off-the-shelf components that are high quality, yet low in price and have a short delivery time. The ability of using off-the-shelf components can also prove to be useful for the reproducibility of the experiment.
Smooth transition	The switch between cam profiles should be non-noticeable. A non-noticeable switch most likely won't exhibit any additional energy losses due to switching.

	Furthermore, it would not affect the comfort of walking with the prostheses, as a noticeable difference could decrease the comfort of walking with a prosthesis, and therefore affect the prosthesis acceptance.

b. Assumptions

Whilst designing the theoretical concept of the DESR foot, several assumptions were made. For the slope of the torque-angle curves, human walking data [4] was used to map and replicated the human ankle joint during the controlled dorsiflexion.

Using two torque-angle curves allowed for the ability to 'capture' energy in early stance and 'recycle' this in late stance in order to enhance the push-off. Datapoints around a linear stiffness slope, in both the plantar and dorsiflexion, were chosen and then the MATLAB spline function was used in order to get a smooth curve. Afterwards an iterative process was used to get equal areas for the energy captured and released. Here one of the spline points on the RED CURVE IN FIGURE 1b in the top right quadrant was moved upwards if the recycled energy was larger than the captured energy and move downwards if it was less. The difference between the captured energy and recycled energy after the iterative process depended on the step sizes taken for moving the spline point up or down.

For the forward model, the principles of virtual work were used to determine a cam profile shape that would deflect the leaf spring in such a manner, during an angular deflection of the ankle, that the desired torque-angle curve would be achieved at the ankle joint. The following assumptions were made in order to simplify the model:

- The spring acts as a linear rotary spring
- The system acts as a conservative system where no energy losses occur due to heat or friction
- The frame deflection is dependent on the desired ankle torque in the torque-angle curve
- All the energy during midstance is captured in the spring, as the torque during midstance at zero degrees is equal to zero Newton meter.
- The energy at zero degrees midstance is used to define a preload angle for the spring at midstance for the red curve in figure 1b
    - This preload angle affects the rest of the geometrical values as well.

Applying the principles of virtual work provides the necessary deflection of the spring element throughout gait. This deflection is for the center of the cam follower, but the cam follower has a finite radius. Considering that the point of contact between the cam profile and the cam follower is offset by the radius of the cam follower, the theorem of parallel curves can be applied to account for this offset.

c. Challenges

i. Self-intersection

After applying the theorem of parallel curves, the shape of the cam profiles would seem to self-intersect. This is illustrated in figure 2. The fixed goniometrical constraints in combination with the fixed spring mechanics output, a shape that could not be machined in a physical part. It appears that wanting to store more energy caused this self-intersection. As the plantarflexion phase stores the same amount of energy as the dorsiflexion phase, but during a smaller range of motion and with a lower absolute torque value, it was more susceptible to this self-intersection. An algorithm was designed to automatically detect this self-intersection, even if it was not perceivable by merely looking at the MATLAB plots.

Increasing the spring-stiffness or height of the ankle joint seemed to diminish the self-intersection occurrence. Both these solutions were suboptimal though. Increasing the spring-stiffness reduced the height difference between the two cam profiles at their equilibrium positions to a smaller distance. Therefore, the imperfections due to machining tolerances would have a larger effect on the error in captured and recycled energy due to an increased ratio of the machining tolerance divided by the height difference between the cam profiles. Increasing the ankle joint height would prove contradictory to the requirements mentioned above.

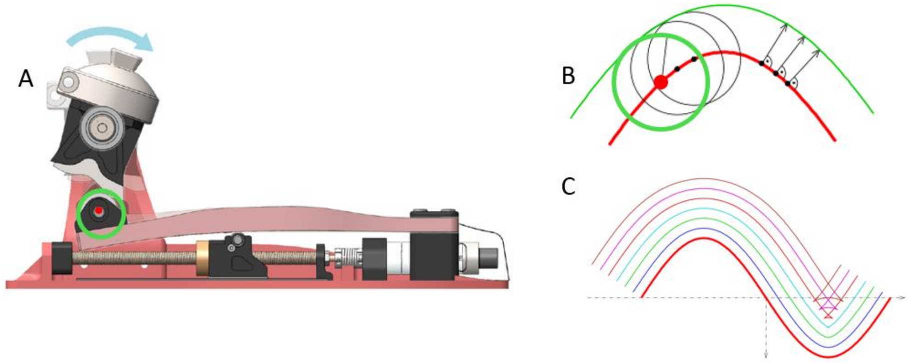


*Figure 2: **A)** The red dot represents the position of the center of rotation of the follower, the green circle represents the outer radius of the bearing that will contact the cam profile. **B)** The theory of parallel curves is illustrated. If the red dot should undergo a certain deflection, the shape to the cam profile should account for the radius of the bearing around the center of rotation, as this point is not infinitely small. Due to this finite geometry of the follower, the cam profile will take the shape of the green curve (adapted from [5]). **C)** Here it is illustrated that a certain deflection profile of the red dot has a maximum allowable bearing size, before self-intersection occur (adapted from [5])s.*

For achieving certain torque angle profile, the virtual work calculations give a deflection profile for the center of rotation of the follower that is connected to the spring through an axis. The point of contact with the cam profile however is the outer radius of the follower. To achieve the actual physical form of the cam profile, the theory of parallel curves must be applied, where the contact of the outer radius of the follower causes a desired movement of follower's center of rotation. This way the movement of the follower is 'linked' to the deflection of the spring. This is illustrated in figure 2.

Choosing a larger radius of the follower can cause a self-intersection after the parallel curve calculations. From a machining point of view, the self-intersected parts are unable to exist in the physical realm for two-dimensional machining, as the machining tool simply follows the cam profile curves that were input for machining. The maximum value of the follower radius is therefore limited.

The differing torque angle curves allow for different ankle response torques for the same ankle angle. That also means the spring must undergo different deflections. The physical interpretation of the self-intersecting cam profile curves is that the deflection path necessary to achieve the desired torque angle curves is not possible due to physical constraints of the spring and follower, such as the inability to change the spring stiffness during the stance phase and the cam follower having a constant finite radius that can interact with more of it's surrounding than just the point of contact.

Besides the follower radius, there are four more factors that affect, and can prevent, the self-intersection of these cam profile curves. 1) Simply capturing less energy between the two curves is less prone to self-intersection, as the similarity of both torque angle curves requires similar shaped cam profiles. 2) Getting a spring with higher stiffness values. To achieve the same amount of energy storage, the spring must be deflected less. Therefore, it is less likely to encounter the physical constraints mentioned above, as the deflection value can be changed throughout the gait within a large range, whereas the spring stiffness is fixed. 3) Increasing the height of the ankle joint of rotation. The same ankle angle, will lead to a longer cam profile as the height is increased, making the cam profile 'smoother' as the changes between torque angle curves can be dealt with gradually. An analogy to explain this would be the difference between a stair step and a ramp with the same height. If you roll an object off either, they will undergo the same height difference. But as the ramp does this gradually, it will seem smoother. One of the 'subjective' design parameters was also the smoothness of the cam profile, as the person might find a sudden change in the profile uncomfortable. 4) Displacing the axis of the follower from the spring in the vertical direction. This affects the way the spring is deflected and affects the virtual work calculations. Placing the follower higher than the spring can also allow the ability to use a smaller radius cam follower, which is beneficial as mentioned above. This is because the cam profiles then are less likely to clash with the spring, as the vertical distance between the cam profiles and the spring increases.

The theory of parallel curves uses a differentiation process. The areas of the captured and recycled energy area determined iteratively by moving the cubic spline points. The resolution of these areas should be quite precise. If the areas differ too much, it will cause a jump/discrepancy whilst plotting the cam profile in MATLAB.

d. Resolution

A finite amount of data points is used to plot the splined torque-angle curves, and to get equal areas of the captured and recycled energy. This causes a difference between a nano- and a picometer, depending on the plantar or dorsiflexion side of the cam profile. Despite this difference being negligible, and the inability to see this due to machining capabilities, it is still noticeable when loading the MATLAB generated curves into SolidWorks. The curves must be split up and only one curve should be used in the transitioning zone.

The room for design of the DESR mechanism is limited. Whilst keeping the geometry of the DESR foot as close to the VSPA foot for comparison during testing, there was limited freedom for design. One of the main challenges was to combine all the above-mentioned design requirements, whilst not increasing the foot size.

3. Mechanical design

a. Morphological overview

In order to design multiple cam profiles that allow for utilizing the dual cam path transition, a morphological overview was made to map the possible solutions. Figure 3 shows the coordinate system in order to better understand the morphological overview in table 1.

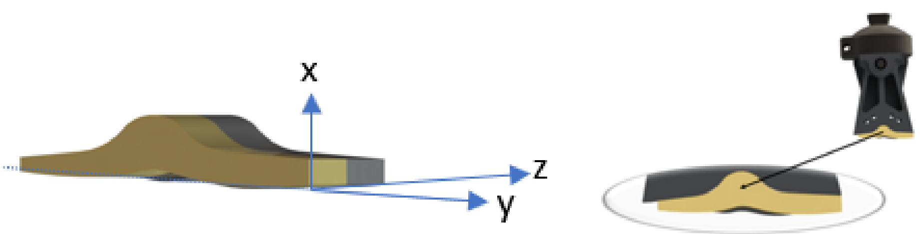


Figure 3: Two cam profiles with their three-dimensional coordinate system that was used to design the cam profiles in MATLAB.

Direction of movement / Moving part	X-axis (vertical)	Y-axis (posterior-anterior)	Z-axis (medio-lateral)
Cam profile displacement	Tough to implement in small mechanism due to machining capabilities whilst maintaining structural integrity	Would clash with follower	Possible by splitting up cam profiles in multiple parts along the y-axis
Cam follower displacement	Two followers needed. Or non-spherical followers. Due to the fixed cam follower axis, it would prove difficult to disengage one of the followers	Calculations of virtual work are based on a fixed cam follower axis	Possible, introduces friction

Table 1: Morphological chart. For the transitioning, both the cam follower and cam profiles can translate in a three-dimensional space. This table show the different possibilities, where the red options cannot be used due to geometrical constraints and the constraints of the virtual work calculations. The purple option was used to work out two concepts, but the frictional losses prevented the transitioning between torque angle curve. The final design is the green option, and the blue option was not explored.

The transitioning between torque-angle curves can be achieved by translating either the cam profile or cam follower in one of the three directions seen in figure 3. The red options in table 1 are not possible due to either geometric constraints or the assumptions made for the principles of virtual work. The purple option was explored but did not prove useful in practical terms. The green option is was used for the final design and the blue option was not explored.

b. Failed concepts

i. Concept 1

In order to switch between the cam profiles, a new cam and follower transmission is devised. Figure 4 illustrates that the follower is now spherical, so it can roll along the cam profiles as they rotate in the sagittal plane, but also transition in the medio-lateral direction between the cam profiles due to a translational degree of freedom in the frontal plane. This additional degree of freedom is provided by the geometry of the housing in which the follower is situated and the geometry of the cam profiles at the transition points.

Grooves are machined in the cam profiles to aid with keeping the follower in place and guiding it along the cam profiles during gait. The extremities of the dual cam profile allow for a transition between the separate cam profiles. The internal geometry is described by figure 4 and the locations of these transition points on the cam profiles are illustrated in figure 5.

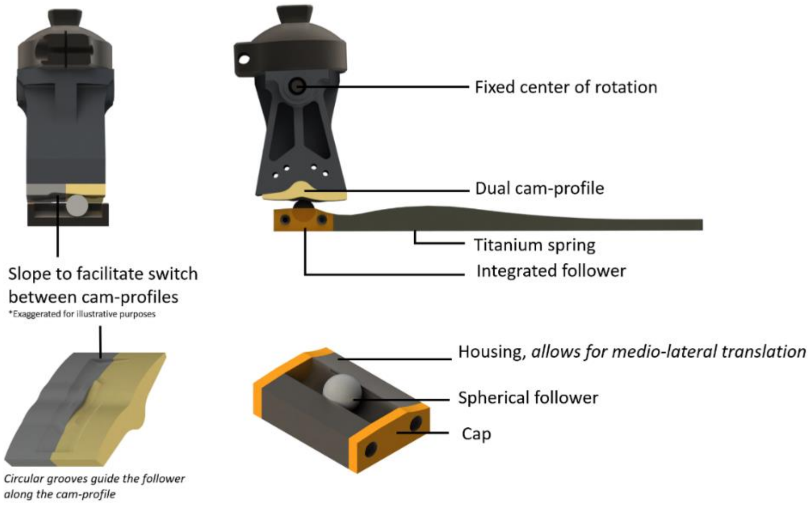
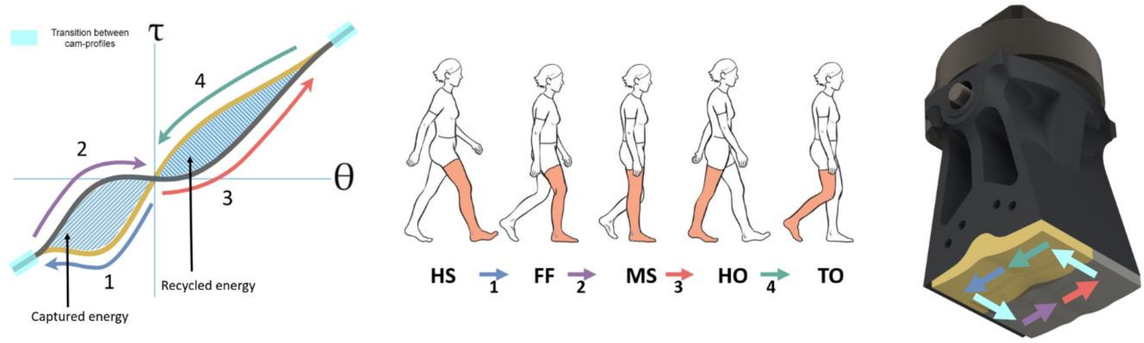


Figure 4: Assembly of the first concept with important features denoted.



A B C

*Figure 5: **A)** Torque-angle profiles for two distinct cam profiles. **B)** The key stance-phases corresponding to the torque-angle profiles, where HS = Heel strike, FF = Foot flat, MS = Mid-stance, HO = Heel-off, TO = Toe-off (adapted from [6]). **C)** Follower path across the two cam profiles.*

Some form of energy must be used to transition the follower between cam profiles. A slope between the grooves of the two cam profiles is introduced to guide the follower from one profile to the other. Because pressure is applied on both sides of the follower, the slope will assist the follower in transitioning between cam profiles. Since the follower is transitioning across a slope, some energy is lost in the transition period. For determining the slope there is a trade-off between the efficacy and efficiency of the design. This slope should preferably be as shallow as possible to ensure minimum energy loss; however, it should be steep enough to ensure the transitioning of the follower during gait.

Another downside of this concept was that the follower would experience sliding. During the rotation of the ankle in the sagittal plane, the spherical follower and the cam profile would have a rolling contact (which is desired), whereas the spherical follower and the spring housing would have a sliding contact. This sliding contact is undesired, as it will add additional friction that the system must overcome and will cause wear of both the spherical follower and the housing over time.

ii. Concept 2

1. Cam follower

The cam follower had to be adapted in order to switch between the two cam profiles, resulting from the previously mentioned two torque-angle curves. The mediolateral side-by- side placement of the cam profiles required the cam follower to have additional rotational degree of freedom (DOF). A standard spherical bearing has two additional rotational DOFs, to a certain extent, due to a spherical inner ring. The difference in rotational degrees of freedom between a roller bearing and a spherical bearing can be seen in figure 6. Here the spherical shape of the inner ring in figure 6C allows for two additional rotational DOFs. These additional DOF are limited as an axis will run through the inner diameter of the inner ring.

The problem with commercial spherical bearings, however, is that the distance from the most outer part of the bearing to the center of rotation depends on the orientation of the bearing. This would prove troublesome, as the cam-based transmission is based on the principles of virtual work where the spring deflection should be controlled. The calculations could become significantly more complex if the effective radius of the follower was dependent of the orientation, as this is hard to predict. The outer ring of the custom-made bearing, the disk, was machined to have a spherical shape in order to always have the same effective radius. The orientation of the bearing would no longer influence the center of rotation; therefore, the cam follower could tilt from one cam profile into the other without the loss of any potential energy stored in the spring. The internal geometry and the custom-made spherical follower are illustrated in figure 7.

The effective radius is the distance from the point of contact between the cam follower and cam profile till the center of rotation of the follower. By using a spherical shape as the outer diameter of the outer ring, the effective radius is constant for every possible tilt and rotation of the bearing.

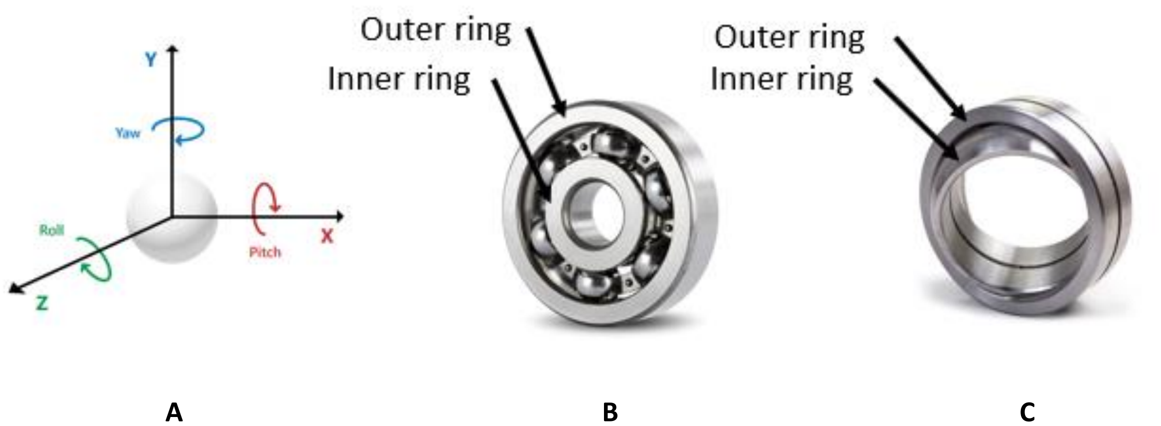


*Figure 6: **A)** The three rotational degrees of freedom possible in a three-dimensional space. **B)** A roller bearing with a single 'roll' degree of freedom (adapted from [7]). **C)** A spherical bearing where the spherical shape of the inner ring partially allows for a 'yaw' and 'pitch' along with the 'roll' rotational degree of freedom as well (adapted from [8]).*

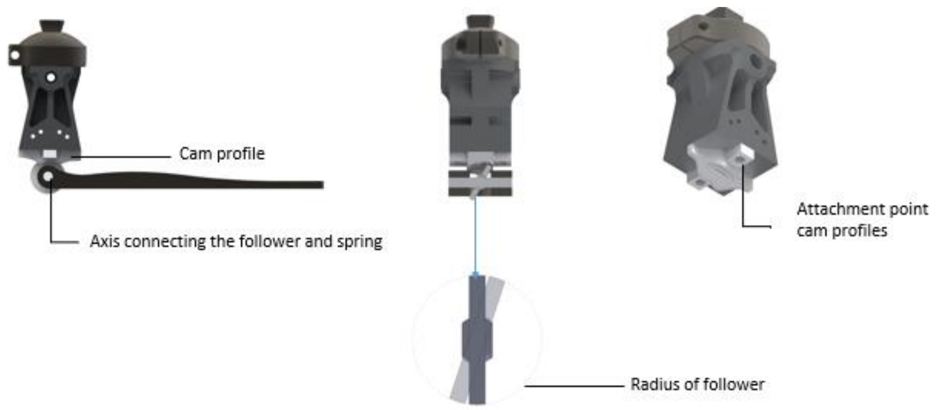


Figure 7: Assembly of the second concept with important features denoted.

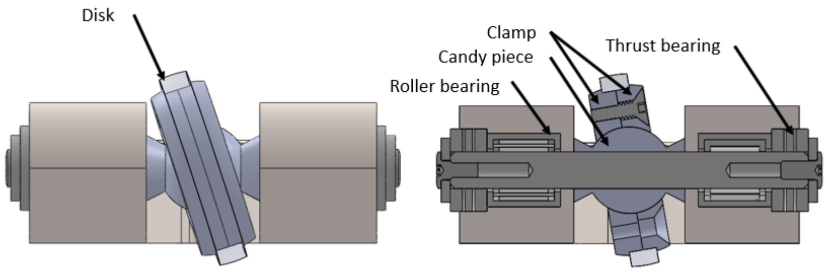


Figure 8: Frontal plane view of the spring with a tilted cam follower. The disk, clamps and candy piece are pointed out. The internal geometry of the spring is also visible.

The outer ring of the follower, in this case the spherical disk, needs a spherical inner ring, just as the spherical follower uses. The tilt of the disk can inflict mediolateral forces on the inner ring. In order to keep this inner ring from sliding across the axis, it is shaped like a ’candy piece’, where the spring acts as a mechanical hard stop. This candy piece is illustrated in figure 8.

As the inner part of the disk can have a smaller effective radius than the inner ring, the disk cannot be manufactured as one part. Therefore, two parts are clamped from both sides of the inner ring and bolted in order to for a disk that can rotate in multiple directions. The clamping of the disk is illustrated in figure 8.

2. Dual cam profile

In order to accommodate this new type of bearing, the contact surface area of the cam profiles needs to change. The line contact between the new cam follower and profile should now take a circular shape, where the deflection of the spring is not affected by the tilt of the follower. This is illustrated in figure 7. In order to prevent the follower from losing contact with the cam profile during gait, or switching cam profiles before the transition zone, walls are added to the cam profile. The walls act as boundaries during the tilt of the follower throughout gait. This way each cam profile has a ’groove’ along which the cam follower can ride. The width of a lane should not be wide enough for the width of the cam follower, but also account for the possible tilt of the cam follower in the transverse plane in order to transition between cam profiles.

Due to the circular geometry of the cam profile, the width of the cam profile also affects the vertical distance between the cam profiles and the spring. Increasing the width lowers the cam profiles. In order to prevent a clash between the spring and the cam profiles during the rotation of the ankle, a maximum width of the cam profiles should be determined for given torque angle curves. This maximum width in turn limits the width of the cam follower as well.

3. Spring

The use of a new follower requires two major changes in the internal geometry of the spring. 1) Roller bearings need to be added in the spring to ensure a rolling contact between the axis and the spring. Under load, the axis rotates with the candy piece. As the follower has a rolling contact with the spring and rolls over cam profile as well, no energy is lost due to slip. 2) Thrust bearings are needed to account for the mediolateral forces caused by the tilt of the follower. The axis is tapped at the extremities in order to fit a bolt. This bolt engages with the thrust bearing to keep the axis in place when mediolateral forces are action upon the system due to the tilted orientation of the follower. This internal geometry is illustrated in figure 8.

4. Transitioning

The transitioning between cam profiles would not result in loss of potential energy stored in the spring due to the spherical geometry, but this new concept would introduce energy losses due to friction. Another limitation in the energy storage is due to geometrical constraints. As the follower is riding in ’grooves’, some of the ankle range of motion is required in order to allow the tilting of the follower between cam profiles without clashing against the wall separating the two cam profiles. During this additional ’transitional range of motion’ the cam profiles must be identical. This decreases the total amount of energy that can be stored due to a reduced range of motion in which energy is captured.

5. Challenges

Rotating the ankle joint in order to deflect the spring causes reaction forces between the cam follower and the cam profile. The magnitude of this reaction force depends on the response torque necessary at the ankle joint. The cam profile and follower should not plastically deform during the torques required for level ground walking.

One of the challenges determining the dimensions of the cam follower. The complexity of the tilted motion proves problematic when performing a finite element model (FEM) analysis. There is also a tradeoff between the cam follower width and the maximum energy that can be captured. In order to perform under high reaction forces, it is preferred to design the follower as wide as possible in order to distribute the force. This in turn requires wider paths and widening the cam profiles decreases the vertical distance between the cam profiles and the spring. Widening the cam follower can cause a clash between the cam profiles and the spring.

As mentioned above, the cam follower disk cannot be one part, but is rather two parts combined. Bolts are used to pretension the clamps around the spherical inner ring of the follower. As the cam follower disk becomes wider, the size of bolts used for the clamps must decrease. The bolts should be able to fit within the smallest part of the outer ring and the largest part of the inner ring. This can be seen in figure 8. The width of the cam follower also affects the maximum tilt angle before the cam follower clashes with the spring. The space in between the spring in which the cam follower is placed is kept as small as possible so the axis through the follower and spring doesn’t deform under load. This axis in turn must be minimized so that the inner spherical part of the follower can allow for a larger tilt angle. This axis is also illustrated in figure 8. A certain amount of tilt is necessary in order to switch between cam profiles.

Manufacturers of custom spherical balls often only take orders of 100 parts or more. Therefore, a custom design and machining of the parts seemed the best solution. The design of bearings should account for tight tolerances to not influence the virtual work calculations too much, but also provide enough space for the lubrication/grease in order to properly function.

Despite the size of the total ankle-foot prosthesis, the freedom of design of the dual path cam follower transmission is limited. There is not one correct solution for this problem due to the interdependencies between the physical and geometrical properties of the cam follower, cam profile and the spring.

Currently the cam follower does not tilt easily under high reaction forces. It was assumed that the follower could be forced to tilt during movement, rather than in a static position. Think of turning the wheels of a car, comparing a driving/rolling car versus one that is stationary. This makes the current mechanism unusable during amputee gait. Due to the lack of structural integrity, the complexity of the design, the not always occurring transitioning and the energy losses due to friction, this concept was discontinued.

Besides the failure of the first two concept due to friction, both concepts required a complex and time-consuming process to achieve the 3D CAD model by using the cam profiles output by MATLAB. A steep learning curve for learning to model complex 3D geometries is prone to errors and can affect the reproducibility of the design and experiments.

iii. Final design

1. Multiple parts

The final concept subdivides the cam profile in multiple parts, mainly a ‘sliding’ part where the cam profiles differ in shape and a ‘transition’ part where the cam profiles are identical in shape. A schematic overview of the new mechanism can be found in figure 9. In the final design parts ‘C’ and ‘D’ were machined out of one piece in order to prevent tolerance stacking. Part ‘B’ was split into two parts in the vertical direction to decrease the post machining efforts. Another reason is that the top piece of part ‘B’ can be machined from a lighter and cheaper material that is also easier to post machine. This is possible as the bottom piece of part ‘B’ experiences a Hertzian contact stress, where as the top piece distributes the load over a larger area. By separating the part in two pieces, it is also possible to use shims/spacers to account for any negative tolerance errors. The bottom piece of the sliding part and the transitioning parts are constructed with hardened D2 tool steel. A free running tolerance fit was desired for the sliding part, however, the machining capabilities available did not allow for such small tolerances.

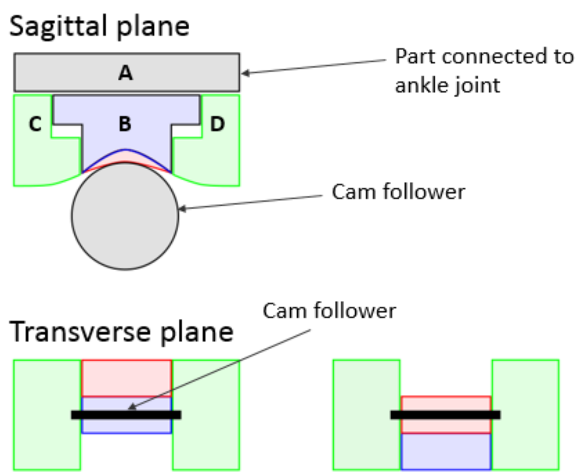


Figure 9: Conceptual design of the final concept. On top is a cross section of the cam profile in the sagittal plane. Here ‘A’ is the part that connects to the ankle joint, ‘B’ is the sliding part, ‘C’ and ‘D’ are the transitioning parts. On bottom is the cross section in the transverse plane, where on the left the follower is engaged with one cam profile and on the right, it is engaged with the second cam profile. The engagement of the cam profile depends on whether dorsi- or plantarflexion is being performed. The green parts represent the transition zone, where both cam profiles have an identical shape.

4. Assembly and manufacturing

a. Bill of materials

This chapter discusses the mechanical parts of the device. A description of the control board can be found in the supplementary document 'VSPA_Control_Board.pdf'. The SolidWorks part files, drawings and MATLAB code can be found in the supplementary documents as well.

	Component	Company	Part number/name	Quantity	Total Cost ($)
Mechanical	Base plate	SuNPe	-	1	85
	Base		-	1	223
	Left stand		-	1	123
	Right stand		-	1	123
	Motor block		-	1	62
	Spring clamp		-	1	28
	Motor holder		-	1	62
	Slider		-	1	54
	ACB clamp		-	1	38
	ACB holder		-	1	77
	Left magnet holder		-	1	54
	Right magnet holder		-	1	54
	Cam profile transition		-	1	177
	Cam slider top Delrin		-	1	38
	Cam slider top Aluminum		-	1	62
	Cam slider bottom		-	1	154
	Cam base		-	1	154
	Cross brace		-	1	28
	Leaf spring		-	1	346
	Cam follower	MiSUMi USA	NATV6	1	22.37
	Cam follower shaft		SFHR6-37	1	9.36
	Support shaft/pivot		PSFHR8-37	1	10.66
	Rubber plates (white)	Amazon	SoleTech Diamond 50-55 Durometer Soling Sheet, 18" x 36" (12 Iron, 6mm, 1/4", White)	1	17.50
	Rubber plates (black)		SoleTech Diamond 50-55 Durometer Soling Sheet, 18" x 36" (21 Iron, 11mm, 7/16", Black)	1	35.00

<table>
<tr><td rowspan="10">Mechanical</td><td>Delrin</td><td rowspan="4">McmasterCarr</td><td>2638T82</td><td>1</td><td>7.25</td></tr>
<tr><td>Stand bearings</td><td>6153K710</td><td>2</td><td>23.90 (11.95 each)</td></tr>
<tr><td>Coupler</td><td>6208K323</td><td>1</td><td>78.09</td></tr>
<tr><td>Dry-Film Lubricant</td><td>1163K18</td><td>1</td><td>17.01</td></tr>
<tr><td colspan="5"></td></tr>
<tr><td>Angular contact bearing</td><td>IBSCO</td><td>SSMER2SD503
Or a different bearing with the following dimensions:
Bore diameter: 1/8 inch (0.125")
Outside diameter: 3/8 inch (0.375")
Width: 5/32 inch (0.1562")
Alternative is mentioned as these were hard to acquire.
If the bearings are switched out, the design of the lead screw, ACB holder and ACB clamp should be adapted as well</td><td>2</td><td>56.00 (28.00 each)</td></tr>
<tr><td colspan="5"></td></tr>
<tr><td>Lead screw</td><td>Helix Linear Technologies</td><td>POWERAC 1/4"-20 RS W/PLASTIC NUT

Custom lengths were required. See the drawings section at the end for the exact dimensions.</td><td>1</td><td>274.00</td></tr>
<tr><td colspan="5"></td></tr>
<tr><td>Pyramid adapter</td><td>Bulldog Tools</td><td>36R-TP</td><td>1</td><td>64.64</td></tr>
<tr><td></td><td colspan="5"></td></tr>
<tr><td rowspan="4">Fasteners and screws</td><td>Shoulder bolts bearing</td><td>McMasterCarr</td><td>90323A245</td><td>2</td><td>11,74 (5.87 each)</td></tr>
<tr><td>ACB/leadscrew nut washer</td><td></td><td>90917A431</td><td>1</td><td>12.07</td></tr>
<tr><td>Motor holder and motor screw</td><td></td><td>91290A012</td><td>1</td><td>13.44</td></tr>
<tr><td>Spring clamp and spring screw</td><td></td><td>93070A105</td><td>1</td><td>14.26</td></tr>
</table>

Fasteners and screws	Leadscrew nut		90730A006	1	5.66
	Base plate and base screw		91294A186	1	4.82
	Base and stand screw		91290A176	1	7.54
	Roller holder and roller screw		91294A002	1	6.89
	ACB clamp and holder screw		90274A018	1	2.79
	Stand and stand bearing screw		92467A115	1	7.78
	Base and motor block screw (side)		91290A144	1	8.70
	Cross brace and stand screw		91294A125	1	5.36
	Base and motor block screw (bottom)		91294A190	1	5.02
	Base and motor holder screw		91290A111	1	8.42
	Cam base and cam profile transition screw		91294A192	1	5.02
	Cam slider top and bottom screw		91263A442	1	8.86
	Magnet distance adjustment screws		91290A018	1	13.30
	Magnet screw (magnet to holder and magnet to cam profile)		96640A021	1	5.86
	Magnet holder screw		91294A002	1	6.89
Total					1953.74

b. Mechanical components

This section will discuss all the machined parts, their functionality and the sequence of assembly. The internal geometry and the machined parts of the DESR foot are illustrated in figure 10. On the left of figure 10 are the outer frame components and on the right is the internal geometry with the remainder of the machined components. First each part will be briefly discussed in terms of functionality. Next the sequence to assembling the parts together will be discussed.

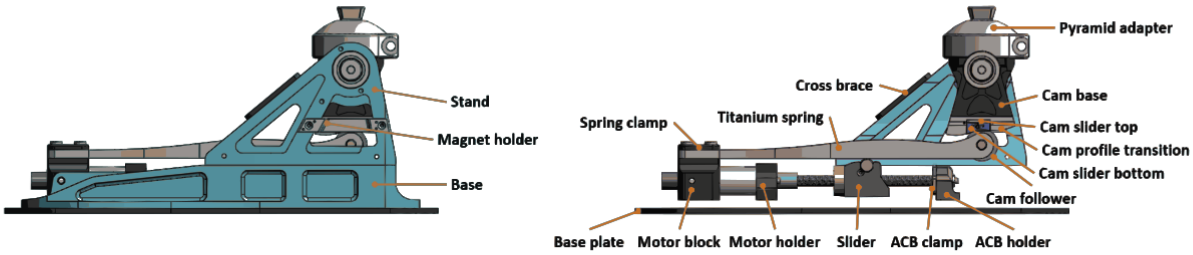


Figure 10: On the left is a sagittal plane view of the DESR foot with the frame components highlighted. On the right is the sagittal view as well, but here the internal geometry is more visible. All the machined parts, the cam follower and the pyramid adapter are pointed out.

1. Base plate

The main reason of having a base plate independent of the base is modularity. In the final design of the DESR foot, rubber sheets were glued to the base plate in order to increase the friction during gait and introduce some form of damping to prevent a completely rigid roll-over shape. Since the rubber plates were glued to the base plate, separating the base plate from the base plate introduces modularity.

2. Motor block

The motor block has two functions. First, it raises the spring to the proper elevation for the cam follower and the cam profile to make contact. Second, it acts as a housing for the motor.

3. The motor holder

The primary goal of the motor holder is to align the motor with the lead screw. It also acts as a hard stop in the fore-aft direction whilst placing the motor.

4. Slider

The slider is connected to the lead screw by means of a plastic nut. Since the motor and lead screw are connected by a coupler, applying a torque with the motor causes the lead screw to turn, and thus the slider moves in the fore-aft direction. Since sliding is involved, Delrin is placed both at the bottom of the slider and on top of the cam base in order to reduce friction. This is illustrated in figure 11. Placed on top of the slider is a commercial pin from MISUMI that acts as the pivot point when the spring is deflected.

5. ACB clamp

The acronym ‘ACB’ is short for angular-contact bearing. The lead screw needs to be able to rotate near friction less. However, the slider connected to the leadscrew introduces some form of friction. Therefore, the bearings press fitted onto the leadscrew undergo some form of axial loading whilst the slider is moved. The ACB clamp acts as a hard stop for these ACB bearings.

6. ACB holder

The ACB holder acts as a housing for the ACB bearings and aligns the center of the bearings with the motor head, so that the lead screw is aligned with the motor as well.

7. Cam follower

The cam follower is a roller bearing connected to the means of a MISUMI pin with the primary goal of deflecting the spring according to the shape of the cam profile.

8. Cam slider bottom

The dual cam path transmission consists of three parts when viewed from the sagittal plane. The first is the cam slider bottom, where the mediolaterally placed cam profiles have different shapes because they have different torque angle curves. This is one of the two parts that makes up the whole cam slider. As the bottom piece, it places the load on the cam follower upon rotation of the ankle joint.

9. Cam profile transition

The second part of the cam profile is the cam profile transition. This part of the cam profile is identical in shape for the mediolaterally placed cam profiles, as the torque-angle curves in the 'transition zone' are equal. When the cam follower is in contact with the cam profile transition, the cam slider is under a no-load condition and can freely slide in the mediolateral direction in order to switch between cam profiles.

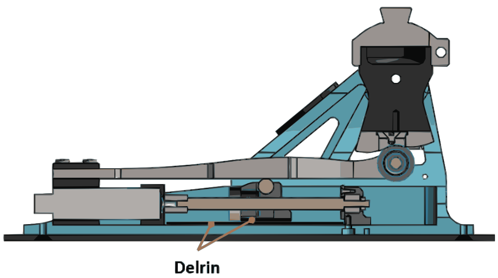


Figure 11: Delrin is added between the bottom of the slider and the top of the base in order to reduce the friction between the components.

10. Cam slider top

The cam slider also has a top part. There are several reasons for subdividing the cam slider into two pieces. It is easy to post-machine if the part is too big. Multiple parts also allow the use of spaces if the part is not large enough. Furthermore, the top part can be made from a softer, cheaper and lighter material, as the load between the cam base and the cam slider is distributed and not a Hertzian contact.

11. Cam base

The cam base is the part that allows the attachment op the cam profiles and the pyramid connector, and thereby coupling the ankle joint to both the cam follower transmission and the shank respectively.

12. Pyramid adapter

The pyramid adapted provides the ability to attach the DESR to the socket of the amputee.

c. Assembly sequence

Even though the assembly can be performed in multiple ways, a few things are recommended.

1. Delrin glue

Use a glue that is designed to handle shear stresses

2. Shimming and sanding

The Delrin on the slider or on the cam base must be sanded down so the motor does not reach high current values. An alternative is to add spacers between the cam base and the frame stands to decrease the preload on the spring due to the cam profile

3. ACB bearing orientation

Pressing the ACB bearings in the ACB holder. Angular contact bearings are designed to accommodate both radial and axial loads. Two angular contact bearings should be press fitted into the ACB holder in a face-to-face arrangement. This orientation can accommodate axial loads in both axial directions, where each bearing accommodates the load in one direction. It is possible to use the back-to-back arrangement as well, however, both the face-to-face and back-to-back arrangements have their pros and cons. The back-to-back arrangement can account for larger support moments generated due to axial loads off set from the bearing. This moment is illustrated in figure 12. The face-to-face arrangement allows for smaller support moments but can compensate more for misalignment. In the DESR foot, the face-to-face arrangement was chosen to compensate for possible misalignment due to the thread machining capabilities and play of the slider. After the press fitting of the bearings is done, the ACB clamp can be attached to the ACB holder.

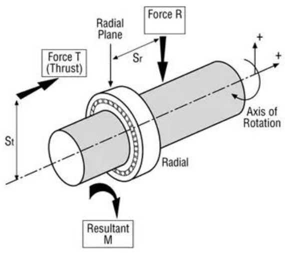


Figure 12: Support moment on bearing [8]

This document talks about the conceptual design, the mechanical design of the device and provides additional information for better understanding the design of the DESR foot. Additional documents can be found in the supplementary materials.

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