Design and Evaluation of a Reaction-Force Series Elastic Actuator Configurable as Biomimetic Powered Ankle and Knee Prostheses

by

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Submitted to the Program in Media Arts and Sciences
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Abstract

All commercial leg powered prostheses have been, up to this point, a one-size fits-all design, and of those existing systems, none has yet managed to fully achieve biological walking range of motion, torque and power. Yet, no human body is the same as the next. A configurable prosthesis potentially offers improvements in battery run-time, prosthesis mass, acoustic noise, user comfort, and even enables sport and economy modes within the same fundamental hardware. In this thesis, a reaction-force, series-elastic actuator (RFSEA) is presented that is capable of achieving biomimetic ankle and knee kinetics and kinematics during level-ground walking across a range of body masses, heights and walking styles. The platform is configurable to inertial load by swapping a simple-to-manufacture flat-plate composite spring that allows tuning the actuator dynamics to match different user requirements. The RFSEA also comprises a high torque and pole-count drone motor that directly drives a ball screw with a tunable, low-gear ratio lead. The design enables high dynamic range providing a closed-loop, torque-controlled joint that can demonstrate arbitrary levels of impedance. This control fidelity is important to support smooth control in free-space and high-inertial output conditions, such as the swing and late-stance phases of walking, respectively. A simulation framework is presented that defines mechatronic design specifications for the motor, spring, and gear-reduction components. The optimization procedure clamps output joint dynamics to subject-specific biological gait data, and searches for minimum electric energy solutions across the motor, gear-reduction and spring component space. A second optimization procedure then searches for optimal linkage and spring geometry to best approach the design targets as constrained by the availability of discrete drivetrain components. In this thesis, ankle and knee designs are presented with optimized components using biological joint data from a non-amputee subject walking at 2.0m/sec with a body mass equal to 90Kg. For these designed biomimetic joints, system specifications are verified using bench test evaluations, and preliminary human gait studies. With a minimum viable actuator mass of 1.4Kg, the platform has a nominal torque control bandwidth of 6Hz at 82Nm, a repeated peak
torque capacity of 175Nm, peak demonstrated power over 400W (with theoretical limits over 1kW), a 110 degree range of motion, as well as torque and power densities of 125Nm/kg and 286W/kg, respectively. Configured as an ankle-foot prosthesis, there are 35 degrees of dorsiflexion and 75 degrees of plantar flexion, and as a knee the full 110 degrees of flexion are available to enable activities on varied terrain such as stairs and inclines. Walking dynamics are evaluated with a finite state-machine ankle controller piloted by N=3 subjects with below-knee amputation walking at 1.5m/sec on an instrumented treadmill and one subject walking on stairs. In preliminary experiments, net positive work of 0.2J/Kg, peak joint torque of 1.5Nm/Kg, and peak mechanical power of 4.3W/Kg all fall within one standard deviation of the intact-limb biological mean. Configured as an ankle-foot prosthesis, the system mass is 2.2Kg including battery and electronics, and as a knee the system mass is 1.6Kg, making the RFSEA platform the lightest, most adaptable, and most biomimetic leg system yet published.

Thesis Supervisor: Professor Hugh M. Herr
Title: Director, Center for Extreme Bionics, Director, Biomechatronics Group, MIT Media Lab
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The following person served as a reader for this thesis:

Thesis Reader .................................................................

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Acknowledgments

I acknowledge the community at-large in their efforts to expand our understanding of gait in our quest to end physical disability and to push beyond the boundaries of biology.

Thank you committee members: Dr. Sangbae Kim, Dr. Amos Winter and of course Dr. Hugh Herr. Their unique expertise in nimble, running, biomimetic robots, low-cost lower-extremity prostheses, and biomechatronics all contributed to the solutions developed throughout this thesis.

The Biomechatronics Group at the MIT Media Lab has, as a whole, been particularly supportive throughout this process. I believe we can always do more together than one can alone.

A special thanks to the test-pilots who donned these powered prostheses. A great patience is required when working with prototypes. Much of our system can only ever be fully tested on a person, and often there are surprises that take time to solve. Thank you for putting in the time necessary to find solutions, and weathering the unexpected behavior of prototypes.

Many of the manufacturers who fabricated components also proved invaluable in delivering high quality parts on time. In particular: Jean-Francois at Dephy Inc. in Massachusetts for building custom embedded systems, Precision Robotics in California for machining high quality precision components, T-Motor in China for custom motors, and MIT Central Machine for reworking components made at other shops or post-machining ballscrews. A special thanks to Mark Belanger in the MIT Edgerton Student Shop who has always been supportive, and without question let me use his old Haas CNC late one weekend to machine last minute components.

Most importantly, I thank my family and close friends for being so supportive throughout this process. It has been a long slog and you stuck with me to the end. Anna, a phd-widow for much of this process, you are a kind and patient human. Johanna and Erran you have been supportive every step of the way. Thank you Mom and Dad: you work so very hard, and that inspires me to work so intensely to achieve these goals. You walked me around the MIT campus as a child, fostering my early interest in technology. When I was in the second grade, seven years old, I wrote that when I grow up I wanted to build humanoid robots, or work with personal computers. Now, thirty-two years later, I am walking across the MIT campus with the terminal degree for my field, having built one of the most humanoid of robots. What else could be expected from the son of a scrap-metal dealer and an occupational therapist?
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1-1 Arthur Kuo explains how push-off force reduces impact force during collision of the contralateral leg, efficiently redirecting energy and reducing joint fatigue. Figure borrowed from [7].

1-2 (a) Negative energy occurs during heelstrike, through controlled dorsiflexion until the positive energy injection at powered-pushoff. Borrowed from [8]. (b) A plot simulating ankle energy during one 1.25 m/s gait cycle. Push-off occurs at about 50% of the gait cycle. This plot aligns with the stance phases shown in (a). The solid line is ankle joint energy. The dashed line is energy storage in a series spring, for example an ESAR prosthetic foot. The difference between these is the motivation for a powered ankle prosthesis. The heelstrike (collision) and pushoff align with (a).

1-3 Closeup of the TF8 actuator, also referred to as MIT RFSEA. Original photo credit: John Soares, and modified by Matt Carney.

2-1 The TF8 Actuator is designed to operate as either a knee or ankle actuator – shown here configured as an ankle prosthesis.

2-2 (a) This data is knee torque data from nine subjects walking at 2m/s with torque scaled by a user mass of 90kg. The mean and one standard deviation are shown in grey. (b) Power spectral analysis of knee torque contributions up to 70% of the joint power during a gait cycle are used to define design requirements. The sum of torque contributions greater than 2.5% of full scale are used as the magnitude requirement. Knee torque frequency contributions amount to 73Nm at 6Hz with a maximum torque of 118Nm.
(a) This data is ankle torque data from nine subjects walking at 2m/s with torque scaled by a user mass of 90kg. The mean and one standard deviation are shown in grey. (b) Power spectral analysis of ankle torque contributions up to 70% of the joint power during a gait cycle are used to define design requirements. The sum of torque contributions greater than 2.5% of full scale are used as the magnitude requirement. Ankle torque frequency contributions amount to 82Nm at 4.8Hz with a maximum torque of 160Nm.

Schematic of the series elastic actuator and parallel elastic element used for the design process (rack-pinion model adapted from [67]). Our coordinate frame assumes ankle angle and torque do positive work on the environment – plantarflexion is positive; for knee flexion angle and torque is also positive.

Effects of series spring stiffness on motor performance and ankle gait energetics for 90kg user walking at 2.0m/s. The linearized model of motor capacity is shown against the performance of the actuator with varying series elasticity. Increasing stiffness begins to saturate motor capabilities.

Effects of series spring stiffness on motor performance and ankle gait energetics for 90kg user walking at 2.0m/s. Spring deflection affect on both electrical and mechanical energetics.

Preliminary validation of the model with the TF8 ankle actuator described in [61] with a human test subject with unilateral below knee amputation. User mass is 75kg, walking 1.5m/s, and $K_s = 378 \frac{kNm}{m}$. Solid line is the cumulative mean electric energy, with purple shading showing one standard deviation from 28 strides. Simulated data is the dotted line. The experimental procedure for testing is explained in [61].

Contour plots of ankle powered prosthesis electric COT for one gait cycle are affected by series spring and parallel spring stiffness and overall transmission ratio. These analyses are based on the motor T-Motor U10 kV100 Plus [61]. The row shows (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively. These do not include a parallel spring, they are the SEA configuration. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions; some regions may be infeasible.
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3-1 The TF8 Actuator is designed to operate as either a knee or ankle powered prosthesis – shown here configured as an ankle prosthesis. Image used with permission, copyright Andy Ryan.

3-2 Ankle joint trajectories of a 90kg person walking 2m/s on level-ground. The maximum extent of these motion profiles are used to help define the actuator design specification. The ankle joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.

3-3 Knee joint trajectories of a 90kg person walking 2m/s on level-ground. The maximum extent of these motion profiles are used to help define the actuator design specification. The knee joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.
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3-28 The moment-arm changes throughout the range of motion of the output. The top plot shows a singularity that is never reached due to the joint being set to -35 to 75 degrees maximum ranges. The middle and bottom plots illustrate how the design optimization places the maximum moment-arm distance at the required power stroke of knee as well as allocating lower effective gear ratio within the range of higher velocity.

3-29 The TF8 Actuator is shown configured as an ankle powered prosthesis with the main components labeled. The actuator minimum build height is 171mm measured between standardized mounting plates, shown here with standard prosthesis components attached. All dimensions are in millimeters.

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Chapter 1

Introduction
Today’s lower-extremity powered prostheses are one-size-fits-all designs that cannot adapt to individual user needs nor do they entirely achieve biological kinetics and kinematics. These limitations reduce the ease of performing certain tasks, and may in some cases contribute to the differing results of metabolic energy benefits found in studies performed with such devices [1]–[4]. A common metric for evaluation of prostheses is the metabolic rate of a person during walking. While reductions in metabolic cost of walking have been found using powered prostheses, compared to passive devices, this energetic cost of transport (COT) still remains greater than that of non-amputees, particularly at common self-selected walking speeds and faster (≥ 1.25ms) [1], [4], [5]; the field has struggled with achieving reduced metabolic cost of walking with powered prostheses, tethered or not (this paper assumes all prostheses are untethered unless otherwise stated). Is this a limitation in technology, mechanical design, controls or have engineering decisions combined with narrow academic targets limited the scope of design efforts?

With the rapid explosion of simulation based machine learning and recent developments in neural interfaces [6] we may quickly approach a point where high-level intentional control has surpassed available mechatronic hardware capabilities. A goal of this thesis is to provide the reader: with a framework to approach designing advanced powered prostheses, some insights into the design tradeoffs inherent with machine design, and a demonstration of what is possible when we push the design constraints and apply rigorous machine design principles. The application of this process is presented as a new robotic prosthetic actuator that not only improves upon the state-of-the-art kinetics and kinematics, but is adjustable to the personalized needs of the individual user. By creating a hardware platform that can be tuned to the individual and more closely approaches the capabilities of biological lower-extremity joints, we advance the field of neurally controlled powered knee and ankle prostheses.

![Diagram](image.png)

Figure 1-1: Arthur Kuo explains how push-off force reduces impact force during collision of the contralateral leg, efficiently redirecting energy and reducing joint fatigue. Figure borrowed from [7].
The motivation for powered prostheses comes from understanding the significance of powered push-off during walking gait. The forward momentum created during powered push-off not only provides forward motion, but it also reduces the impact force felt by the contralateral leg at heel-strike, improving walking efficiency. Imagine: when your toe pushes off the ground your center of mass becomes ballistic as it rotates above the contralateral leg currently in stance. As you finish the swing phase and your heel strikes the ground there is a reaction force on the order of $10 - 15N/kg$ in a direction opposing your forward motion. Greater push-off force actually helps propel your body such that the impact force is timed correctly to redirect your motion forward, fully utilizing the change in potential and kinetic energy to bring your center of mass to the next step. Fig. 1-1 borrowed from [7] schematically shows the energy redirection at heel-strike. Compliance in the knee does also absorb some of this impact energy as do the tendons absorb and redirect this energy through stance. Fig. 1-2(a) borrowed from [8] schematically shows this energy redirection during stance. Phases one through three show how ankle energy is negative during most of the stance phase. The final stance phase then shows a positive energy injected toward the environment from the ankle during push-off. Ankle joint energy and that of a simulated series spring are shown in Fig. 1-2(b) \(^1\). The series spring in this plot is equivalent to a modern energy storage and recovery (ESAR) prosthetic foot. From this figure it is clear that an ESAR foot can store and recover energy during stance (neglecting the loss of energy due to hysteresis in the springs), but it cannot provide more energy than is put into it. This plot demonstrates there is an offset between the biological ankle positive energy contribution and that of the passive spring. At 1.25m/s there is roughly $\geq 0.1J/kg$ of additional positive net work provided by a biological ankle during walking. As walking velocity increases so does the work done. At 2.0m/s energy increases to over 0.4J/kg. The spring cannot provide this positive work that would normally be generated by muscles, meaning passive prosthesis users must compensate for the lack of push-off by using extra motions from other parts of the body. Ultimately, this results in passive prosthesis wearers having trouble maintaining higher than nominal walking speeds. The aim of the powered prosthesis is to provide this added energy at push-off to produce biomimetic joint behavior throughout the walking phases of stance and swing, and across a range of walking speeds. Powered prostheses can provide the positive net work that reduces impact loads, and improves walking efficiency that passive devices fundamentally cannot.

\(^1\)All biological data sets in this thesis originates from [9] and are described in more detail in the Chapter 2.
Figure 1-2: (a) Negative energy occurs during heelstrike, through controlled dorsiflexion until the positive energy injection at powered-pushoff. Borrowed from [8]. (b) A plot simulating ankle energy during one 1.25 m/s gait cycle. Push-off occurs at about 50% of the gait cycle. This plot aligns with the stance phases shown in (a). The solid line is ankle joint energy. The dashed line is energy storage in a series spring, for example an ESAR prosthetic foot. The difference between these is the motivation for a powered ankle prosthesis. The heelstrike (collision) and pushoff align with (a).

If one is affected by a lower extremity limb difference, the ankle work deficiency shows up as increased effort to walk, and joint pain, particularly in the contralateral knee, but it can also show in the hip or back. Without power available at the affected joints more proximal and contralateral muscle groups such as the hip flexors are over utilized to swing the affected side forward, increasing net metabolic energy expenditure for each step [10]. Joint overload requires a user to over-commit center of mass motions, thus reducing stability. Insufficient kinematics and kinetics at the prosthetic joints lead to non-normative and pathological gait styles that contribute to long-term effects: early onset of osteoarthritis in unaffected or contralateral joints (for unilateral amputees), osteopenia and osteoporosis in affected sides and chronic back pain in amputees [11]. Reduced trailing leg push-off force increases impact loading and peak knee external adduction moments during leading limb impact [4], [12]. More acutely, these pathological gaits often affect body-image anxiety, confidence, and well-being.
due to personal and societal perceptions that come with asymmetric gait [13]–[16].

Human gait requires a net energy balance to enable locomotion, yet mechanical energy is dissipated at the swing to heel strike transition due to collision losses, Fig. 1-1 [7], [8], [17]. McGeer demonstrated that potential energy can overcome these transition losses with his passive dynamic walkers where gravitational potential energy converted to kinetic energy compensates for the energy imbalance [18]. However, for level ground walking, and other real-world applications, another source of energy is required. For pedal locomotion, push-off at late stance provides this net positive work necessary to redirect and propel the body center of mass to the next step [8], [19] – at a cost of metabolic energy expenditure due to muscle contraction [20], [21]. Biology reduces the energy expenditure of muscle contraction by exploiting the energy storing and power delivery capability of tendons. These elastic tensile members also act as mechanical means of redistributing energy throughout the body. Energy stored in the strain of these serially attached high elasticity tendons can be rapidly released to deliver more power than the metabolically expensive muscle contraction can do alone. This elasticity reduces the velocity and distance muscle fibers sarcomeres must travel, keeping them within the efficient regime of their power stroke [22], [23]. This energy storage and release behavior applies not just to level ground walking, but in all the states of locomotion from walking to jogging to running as well as ascent, descent, squatting, etc..

Modern commercially available passive prosthetic ankles aim to partially restore functionality by providing a means to mimic normative kinetics by storing and releasing elastic energy in the foot and ankle complex [12], [24], and kinematics with a foot shape [25], [26] or ankle motion that enable more biological rollover motion. The Endolite Echelon VAC [27] or Fillauer Motion Foot [28] are common commercially available passive devices that aim to provide this functionality. Active-passive or microprocessor controlled devices such as the Össur Proprio Foot [29], Ottobock Meridium and Triton Smart Ankle [30] are the current state-of-the-art commercially available advanced quasi-passive prostheses. Similarly, passive knees provide a means to approach knee kinematics by providing a mechanism to allow knee flexion during swing phase and stability during heel strike. The Ottobock C-Leg microprocessor knee [31] can adjust joint angle and damping rates but does not provide closed loop impedance control necessary to dynamically replicate biological motions.

Powered prostheses make use of robotic actuators that can provide kinetics and kinematics that attempt to mimic the behavior of biological joints. These actuators provide positive work that can reduce a person’s metabolic cost of transport [32], and
can include closed-loop dynamic control that simulates normative behaviors at the joint. The one commercially available powered ankle prosthesis, the emPower from Ottobock (previously BionX) [30], has been shown to improve metabolic COT during level-ground [1], [3]. Providing active push-off force can also reduce impact loading on the contralateral leg, reducing long-term comorbidities – a strong motivator for powered ankle prostheses. Traditional position control is helpful in swing phase to move a limb out of the way, but, it has limited stability when collisions occur, such as heel-strike events. Hogan showed impedance control is a means of providing controller stability in robotic systems with contact dynamics, such as collisions, while also mimicking natural neuromuscular behaviors [33], [34]. Impedance control effectively simulates a virtual spring and damper system, allowing control of rebound and energy dissipation that enforce desired joint positions.

The primary goal of powered devices has been restoring normative gait and reducing metabolic COT during level ground walking, while there has been limited attention to dynamic motions on stairs, inclinations, kneeling, jogging, or even fall recovery. This limited scope is reasonable from an energetic point of view and compared to passive devices there is a substantial benefit to achieve normative levels of power at toe push off for ankles. Perhaps, though, focus on energy expenditure during walking is too academic of an approach when simple motions that require larger range of motion, such as sit-to-stand, are not achievable with the limited range of motion of walking-only knee and ankle hardware. Energetic reductions can be difficult to measure for people with above knee amputation (AKA). Reasons for this difficulty may be because learned pathological gait patterns to match hardware kinematic requirements may be difficult to overcome in short laboratory visits. Further, the limited energy expenditure at the knee during nominal walking speeds does not alone account for metabolic differences. Nonetheless, above knee amputees do have a greater rate of falling than below knee amputees. This high fall rate is due to instability, and the rigid trajectory requirements of available knee products on the market, that when unmet, lead to unreliable locking. Powered impedance-controlled devices may improve stability and widen the operating range of knee trajectories. Yet, little hardware has been developed with the velocity capability necessary to perform a trip recovery maneuver that is all too common of an occurrence. AKA subjects in particular have severely pathological gait imposed by these specific hip trajectories required to engage and disengage the traditionally available knee devices. Even further, rarely if at all is an AK subject given access to combined powered knee and ankles.Powered prostheses can improve gait stability, and pathology, limiting hip
and trunk asymmetries that improve the determinants of gait [17], [35], [36]. Moving beyond the narrow focus of metabolic COT, it may become possible to realize greater benefits for the users of this powered prosthesis technology.

Taking a cue from human centered design: it may be helpful to widen the focus beyond level ground walking to include more demanding tasks such as stair or slope ascent and descent, where substantial work on body center of mass, and muscle contraction occur – this could be prime proving grounds to find improvements with powered knees. Widened design requirements such as range of motion, torque and power, or task specificity like jogging, jumping, toe stands, or even high velocity maneuvers such as trip recovery may force open a design space for kinematics and kinetics that enable more biologically accurate capability that ultimately lead to improved performance from level ground walking to dynamic activities.

The struggle with mechanical design of humanoid robotic systems and even more so with prosthetic devices is the desire to minimize distal mass while keeping clearance requirements low, and, while remaining reasonably well within the confines of biological mass and volumes. The energy to move a point mass is $E = \frac{1}{2}I\omega^2$, where $I = \frac{1}{2}mr^2$ demonstrates the quadratic contribution location from center of rotation has on energy. Verifying these equations empirically [37] showed the effects of increased distal mass do negatively affect metabolic cost of walking. Beyond energetics, there is also the ever present suspension problem: centripetal acceleration causes a distal mass to pull a socket off a user during swing phase. The mechanical interface to the body relies on suction and friction through the liner, or a distal ratcheting pin, thus limiting holding force, and the swung mass allowance. This is a challenge in prosthetic design where the wish to minimize energy of the user must be balanced with minimizing the clearance requirements of devices to broaden the range of user base applicability. In all cases, minimizing mass beyond even the equivalent body segment is always desirable. To achieve high performance with lower mass we should allow ourselves to veer from strict adherence to biological volumes, such that design spaces may be found that reduce mass while performing necessary functionality.

It is the role of the design engineer to demedicalize prostheses, to enable functionality without the bounds of traditional frailty design. Graham Pullin states in Design Meets Disability: "There is no such thing as a culturally neutral design language: a lack of intonation speaks volumes...Whether or not graphics are consciously designed, they will inevitably express strong messages to different people, positive or negative [38]." It is the duty of the designer and engineer to assume a certain level of cultural authority as well as authenticity in design. Pullin backs up this claim by
asking: “But might flesh-colored prostheses and miniaturized hearing aids send out tacit signals that impairment is something to hide? In contrast, a shift in perspective from corrective prescription to fashionable eyewear has been influential in helping to all but remove the stigma associated with wearing glasses.” There are two models of disability: the medical model describes a person with a disability, where the social model describes a person as disabled by society – an intentional lack of access to accommodate a body with differences from the norm. Eyeglasses compensate for a visual impairment and so fall into the category of the medical model, and yet their adoption by the fashion industry and their conversion into the most fashionable facial prosthetic device of modern times remains aligned with the mainstream model: compensation for a biological deficit. Society made no additional affordances for the prosthetic device, yet society not only accepts the eye-glass as fashion, it champions it in a multi-billion dollar industry and adorns nearly anyone who can afford anything from a few, to thousands of dollars for their cosmetic designs. The eye-glass de-medicalized itself by conforming society to acceptance, it transcended the cultural appropriation of disability.

Eyewear shows that this breach of protocol — adherence to medical or social models of disability — is allowed and may even be championed, with proper awareness of the social context of the technology. The prosthetic leg, the primary focus of this work, need not hide itself, nor fit itself into a pink plastic pouc. In fact, doing so does the object a disservice, by assuming a submissive, oppressed role for disability; it could be argued that hiding disability enables oppression by societal norms. Building prostheses with limited functionality further medicalizes the device by enforcing the social model of disability. A robotic prosthetic leg is not normal, it is extraordinary, just as every body is, in whatever form it takes. The role of the design engineer who meets disability is to identify how the product to be designed might define its identity in relation to the wearer, their relationship to society, and how this device might affect their psychological momentum. What does the product say to the wearer and in wearing the item what does the wearer say to society? Intentional design language leads the conversation, providing the wearer the opportunity to frame the conversation as they see fit.

How then do we balance the design requirements of functionality with aesthetic? Through rigorous design practice.

Mechanical design is often a brute-force search across a design space. To reduce the search parameters common practice is to look at previous designs and iterate. The trouble is this can leave a designer in a local optima, rather than the global
optimum. Taking a first principles approach to the architecture offers a potential path to a more global optimum. One way to achieve this is to explore effects and trade-offs of the primary components of the series elastic actuator: the spring, motor and drivetrain configuration. The following chapters present a method to evaluate the design trade-offs and performance of a series elastic actuator kinematically clamped to biological gait data. This method is then used to design and build a powered prosthesis actuator, that is then tested with human subjects.

A search objective of minimizing electrical energy consumption, subject to constraints such as motor and battery capabilities is used to identify optimal component specifications. Multiple search steps are performed to identify the motor, gear ratio, joint linkage geometry, series spring stiffness and energy capacity; the search starts broad, and is then narrowed down by available hardware components. A second search then optimizes the spring geometry with an objective of matching a desired energy storage capacity subject to material and geometric constraints.

The design goal is to find an architecture that achieves aggressive activity targets while fitting within biological equivalent mass limits, and reasonable size criteria, while maximizing duration and range of activity. Evaluation of the system capability is evaluated with traditional linear feedback control techniques, and replay of biolog-
ical trajectories with walking controllers to verify real-world capability. A follow-up question asks: if it is possible to build a compact, high fidelity torque controlled actuator that achieves biologically accurate kinematics and kinetics, is there a control system that can then exploit these capabilities to improve functionality to further improve metabolic cost of locomotion for unilateral transtibial, transfemoral or bilateral transtibial amputees? That is in fact the inspiration for this work: to build prostheses that can exploit the development of new neural interfacing techniques that enable proprioception and volitional control, such as the Ewing Procedure (Agonist-antagonist Myoneural interface)[6].

Chapter two details a process for the design of high power density series elastic actuators based on cyclical piece-wise continuous biological trajectories. After presenting the means of simulation, an analysis of electric energetic consequences of design trade-offs across varied terrains is explored. Chapter three then applies these tools and insights toward the design, build, and evaluation of a novel actuator (referred to as TF8) that outperforms many metrics of currently published research-grade powered ankle and knee prostheses. In Chapter four the same simulation methods are then used to layout two future actuator designs that enable further customization: a smaller version of TF8, the TFtiny, and a motor integrated series elastic cycloidal actuator (MISECA).

The actuator presented here is the first single actuator capable of working without modification as both a human knee and/or an ankle powered prosthesis. The TF8 is designed to achieve biologically equivalent torque and velocity for a 90kg user walking at 2.0m/s (near jogging pace). At a minimum actuator configuration the system weighs 1.4kg, has a build height of 171mm and a ROM of 110 degrees. The actuator can generate repeated peak torque of 175 Nm and over 400 W of power giving it a torque density of 125 Nm/kg, and a power density of 286 W/kg, and full closed-loop dynamic control with 6Hz bandwidth at 82 Nm. It is able to generate positive net work during walking that match within one standard deviation of biological energy values and tracks biological torque and power with an $R^2 \geq 0.85$ and $R^2 \geq 0.90$, respectively. The design is, however, not without errors. In an ankle configuration the build height with a foot and pyramid adapter reaches 223mm and a total mass of 2.18kg including battery, electronics, foot and adapter. This is on the tall side for some subjects with longer residuum. The actuator, nonetheless, remains the highest torque and power density research-grade system yet published, while also enabling high accuracy closed-loop torque control. It is the first powered prostheses to be operable as both a knee and ankle, and the first prosthesis that can be tuned to
individual user mass requirements by replacing a simple flat-plate composite spring. The following chapters present the first iteration of personalized bionic legs.
Chapter 2

Simulation
2.1 Introduction

The motivation to build powered ankle prostheses is to generate the roughly 0.1-0.2 J/kg net positive work done by the ankle during walking that cannot be provided by a traditional ankle-foot prosthesis [39], [40]. Providing this energy has been shown to reduce the energetic cost of walking in average fitness-level amputee populations [1], [4], [41], improve qualitative feel and ground control, reducing falls, and normalizing gait to reduce social stigma of pathological gaits as well as early onset of co-morbidities [11]. Though the knee operates primarily as an energy absorber during walking (net negative work through a stride), there are positive work dynamics that people with above knee amputation must compensate for with their unaffected body portions. These extended efforts by proximal and contra-lateral muscles can lead to co-morbidities and pathologies, and similar to the ankle, can potentially be avoided with the dynamic control of powered knee devices throughout swing and stance phases of walking.

The first powered ankle, published by Au, Weber, Herr [41], [42], used a series elastic actuator (SEA) configured with a unidirectional parallel spring (PS) to improve torque bandwidth in the transition from controlled dorsiflexion to powered-push-off. Au et al. showed that the actuator ability to contribute energy during powered push-off improved the metabolic cost of walking [42]. Since then, numerous actuators have been designed following a similar topology of series and parallel springs [43]–[47]. Researchers have studied the energetic consequences of direct drive, series elastic and varied parallel spring configurations, but their focus has primarily been on level-ground walking and running [41], [48]–[51]. This study expands the previous energetic analyses to include stair ascent and descent in addition to level ground walking.

Reductions in metabolic Cost of Transport (mCOT) for people with lower-extremity limb differences wearing powered prostheses compared to those with conventional passive devices has been accomplished [41], [52]. However, differing results of metabolic energy benefits from such devices have also been reported [2]–[4]. In [4] a tethered ankle prosthesis emulator showed no metabolic benefit from positive net work during stance; however, the authors acknowledge the controller was not tuned to individual users even with ±10kg variations in test subject mass. There are substantial differences in user tuning preferences, so these results should not be considered valid. Further, the authors do comment that comfort and improved balance afforded by autonomous (dynamic quasi-stiffness) untethered powered prostheses may improve energetic outcomes. Studies with military servicemembers who received substantial
rehabilitation and strength training have shown no statistical metabolic differences between their unaffected counterpart populations [53], [54], leading to some conclusions that muscle strength and advanced rehabilitation can provide equivalent results for unilateral below knee subjects with amputation. Neuromusculoskeletal simulations have also shown equivalent mCOT for bodies that retain 100% muscle strength [55]. However, many, if not most patient populations do show muscle loss and associated increased metabolic cost of walking. In results that have shown metabolic improvements, the energy cost of walking in average fitness-level patients still remains greater than that of un-affected persons, particularly at common self-selected walking speeds and faster [1], [4], [5]; the field has struggled to reduce the metabolic cost of walking with powered prostheses, tethered or not.

These limitations in clinical studies may be due to differences between machine and biological kinetic and kinematic capabilities, along with the distal mass of such devices. Many clinical energetic studies have used the commercial product BiOM from BionX which has limited range-of-motion (ROM) from zero dorsiflexion to twenty-five degrees plantar flexion and a mass of 2.3kg [56]. These qualities may limit the effectiveness of such devices for smaller patients where the distal mass is substantial compared to their equivalent leg segment. Though well designed for energy efficient level-ground walking, these systems quickly hit kinematic limits at moderate walking velocities and varied terrain, where joint angle extension increase to double that of slow walking [57].

Mimicking nature with synthetic hardware is a capability that the biomechatronics field is closely approaching. Though the performance metrics of academic and commercial applications differ, affecting unique mechatronic design trade-offs. The commercial-off-the-shelf powered ankle and knee systems must provide robust, safe mobility to patients. Perhaps because of these requirements they are not yet able to provide biologically accurate kinetics with kinematics, nor do they enable access to the underlying control systems [29], [56]. In contrast, research hardware has the benefit of less strenuous robustness requirements, allowing more flexibility in design. To improve upon the commercial options, a growing handful of academic research groups around the world have been building individualized platforms for study.

The series elastic actuator (SEA) [58] has been the core actuator technology of powered prostheses due to the energetically favorable features of a contractile element in series with an elastic element. This configuration of spring is a biomimetic representation of the biological muscle and tendon unit [23]. The SEA has a number of benefits that include: reduced impact loading on the drivetrain, force control from
position measurement and control, decoupling of drivetrain impedance from output impedance, and the ability to deliver more power than a motor alone when operated as slingshot [59]. Some ankle prosthesis designs include a unidirectional parallel elastic element to improve control bandwidth, peak power, and energetics in level-ground walking [41], [44], [48], [50], [60]. However, though the unidirectional parallel spring can reduce peak power, it actually performs slightly worse in net energy consumption than an SEA alone on varied terrain, while also increasing system complexity and mass.

This chapter aims to expand the guidance around the mechatronic energetic consequences of the primary design features of lower-extremity powered prostheses when walking on level ground, and during stair ascent and descent. It is hypothesized that the unidirectional parallel spring optimal for level-ground walking has negative energetic consequences during stair ascent and descent where increased range of motion is present. Secondly, limited range of motion may improve energetics during level-ground walking, but also negatively impacts stair ascent and descent. The chapter is outlined with: a process for evaluating the design space and searching for minimum energy combinations of motors, reduction-ratios, series and unidirectional parallel stiffness, and joint range of motion limits. These tools are then applied to simulate ankle prostheses ascending and descending stairs in addition to level-ground walking, as well as knee actuators during level-ground walking. The energy analysis is validated with preliminary tests of level-ground walking with built hardware. Application of these results in built hardware, shown in Fig. 2-1, are presented in the following chapter and an accompanying mechatronic design paper [61] that is separated for clarity. Finally, there is a discussion of differences between the simulation and preliminary results from experiments.
2.2 Methods

Understanding the energetic consequences of design variables leads to higher performance hardware and improved design for application specificity. For this application, the design is initiated assuming series elasticity would be included as a mechanical energy storage mechanism, as has been done by many researchers of ambulating robotics. Following a process similar to [62]–[64] the performance of a SEA is evaluated kinematically clamped to mean biological gait data. The processes then searches for drivetrain component specifications for a series spring, motor, reduction ratio and in some cases unidirectional parallel springs that satisfy the search objective and system constraints. Wider design space evaluation of the energetic consequences of these hardware components is also performed to help understand the consequences of design decisions on stairs in addition to level-ground walking.
2.2.1 Design Specification

To determine the design specification for range of motion, torque, power, and system bandwidth, mean gait data from a total of one thousand unique gait cycles of walking data from nine able-bodied subjects collated from [9] is normalized and scaled by body mass. In addition to walking, mean stair ascent and descent ankle trajectories from [65] are used to evaluate energetics of stairs. A performance target and evaluated component contributions are based on a 90 kg user walking at a near jogging pace of 2.0 m/s. The choice of these heavier and faster than average targets is to design an actuator capable of achieving dynamics that would not be power limited on a wider population of test subjects.

![Figure 2-2](image)

Figure 2-2: (a) This data is knee torque data from nine subjects walking at 2m/s with torque scaled by a user mass of 90kg. The mean and one standard deviation are shown in grey. (b) Power spectral analysis of knee torque contributions up to 70% of the joint power during a gait cycle are used to define design requirements. The sum of torque contributions greater than 2.5% of full scale are used as the magnitude requirement. Knee torque frequency contributions amount to 73Nm at 6Hz with a maximum torque of 118Nm.

The widely accepted measure of an actuator’s ability to achieve a desired performance is the torque bandwidth: a measure of the ability of an actuator to assert a specified torque at a specified frequency. Much of the literature states the ankle bandwidth design requirement is 2-4Hz at 50-140Nm torque for human level-ground walking. This specification originates with [42], where they defined bandwidth as a
Figure 2-3:  (a) This data is ankle torque data from nine subjects walking at 2m/s with torque scaled by a user mass of 90kg. The mean and one standard deviation are shown in grey. (b) Power spectral analysis of ankle torque contributions up to 70% of the joint power during a gait cycle are used to define design requirements. The sum of torque contributions greater than 2.5% of full scale are used as the magnitude requirement. Ankle torque frequency contributions amount to 82Nm at 4.8Hz with a maximum torque of 160Nm.

The frequency range within which 70% (-3 dB range) of the system energy is contained in the torque signal when evaluated with Parseval’s Theorem.

To evaluate torque bandwidth for more complex trajectories other than the ankle such as the knee, or other arbitrary joint trajectories the [42] method is expanded to determine a generalized bandwidth magnitude requirement. The sum of peaks over 2.5% of the maximum absolute torque spectral density are summed, and the maximum frequency of these torque contributions are taken as the frequency bandwidth. This method was verified against the [42] method applied to a 75 kg 1.25 m/s walking able-bodied subject. Shown in Fig. 2-2 are knee torque trajectories collected from [9], and scaled to a 90 kg user walking at 2 m/s. The result is a desired frequency bandwidth of 73 Nm at 6 Hz. To be noted: the specificity of this method is somewhat misleading due to its frequency domain approach to analysis of an entire gait cycle, rather than true time-domain features. Nonetheless, this method provides an effective estimation of torque control requirements to achieve desired trajectory components.
2.2.2 Modeling and Optimization

To model the mechanical and electrical system dynamics a kinematically clamped analysis method clamps an actuator output to explicitly track the prescribed joint load torque, angle and velocity trajectories [62], [64], [66]. Linearized kinematic and electric dynamic models locked to these vectors output required behavior from the: spring (displacement), motor (velocity and torque), and associated power electronics (voltage and current). These models do not include non-linear viscous friction terms but the search method does include saturation constraints on motor torque, velocity, current, and voltage. The result is a model that generally trends correctly but is operating in an open-loop dynamic condition with minimal damping, thereby potentially exhibiting large velocity response at impact conditions e.g. parallel spring engagement or ground contact. Mean torque, velocity, and angle trajectories aggregated from [9] were mass normalized, scaled, and fed into the dynamic equations for a series elastic actuator with or without unidirectional parallel elasticity. The actuator schematic is illustrated in Fig. 2-4.

![Schematic of the series elastic actuator and parallel elastic element used for the design process (rack-pinion model adapted from [67]).](image)

Figure 2-4: Schematic of the series elastic actuator and parallel elastic element used for the design process (rack-pinion model adapted from [67]). Our coordinate frame assumes ankle angle and torque do positive work on the environment – plantarflexion is positive; for knee flexion angle and torque is also positive.

Kinematic analysis from free body and mass acceleration diagrams of the joint
components give the following equations that reference Fig. 2-4:

\[ \tau_l = \tau_s + \tau_p - J_l \ddot{\theta}_l \]  

(2.1)

where,

\[ \tau_s = r_s F_s = r_s K_s \left( \frac{\theta_m}{N} - r_s \theta_l \right) \]  

(2.2)

\[ \tau_p = \begin{cases} -K_p(\theta_l - \theta_{p0}), & \theta_l < \theta_{p0} \\ 0, & \theta_l \geq \theta_{p0} \end{cases} \]  

(2.3)

and where, joint load is \( \tau_l \), the series elastic actuator torque contribution is \( \tau_s \), the unidirectional parallel spring contribution is \( \tau_p \), the load inertia is \( J_l \) and load acceleration is \( \ddot{\theta}_l \), the moment arm of the output is \( r_s \), the series elastic element stiffness is \( K_s \), the motor angle is \( \theta_m \), the load angle is \( \theta_l \), the reduction ratio is \( N \), the parallel spring rate is \( K_p \), the parallel spring engagement point is \( \theta_{p0} \). In the case of the knee \( J_l \) assumes a point mass of an equivalent leg segment [68] located one third of a leg distance away from the knee joint. Positive joint angle and torque is in the plantarflexion direction, based on a reference that positive work is done to the environment. The load inertia is not necessarily known as it changes through stance and swing phases; for this analysis inertia was assumed operating in free-space and used equivalent leg segment mass, as defined in Section 3.2.1, for the ankle or knee analyses depending on the mass of the next distal link. Here \( r_s \) is the moment arm acted on by the linear actuator force \( F_s \) at the joint axis. In the case of linearly actuated joints such as the commonly utilized ball-screw drive represented in Fig. 2-4 the moment arm can be a function of joint angle, \( r_s(\theta_l) \). The motor position is \( \theta_m \), and the drivetrain reduction ratio is \( N = N_s r_s \), where for the case of the ballscrew \( N_s = \frac{2\pi}{\text{lead}} \). Plugging \( \tau_s \) and \( \tau_p \) into equation (2.1), and rearranging, the force in the series spring actuator is \( F_s \):

\[ F_s = \frac{1}{r_s} [\tau_l - K_p(\theta_l - \theta_{p0}) + J_l \ddot{\theta}_l] \]  

(2.4)

Passing this force through the transmission results in an effective torque that the motor sees:

\[ \tau_m = (J_m + J_{tr}) \ddot{\theta}_m + b_m \dot{\theta}_m + \frac{F_s}{N_s} \]  

(2.5)

\[ b_m = \frac{K_s I_{nl}}{\omega_{nl}} + b_r \]  

(2.6)
where, $J_m$ is motor rotor inertia and $J_t$ is transmission inertia. Along with the motor and transmission inertias, the damping term $b_m$ given by [69] represents viscous friction in the motor, where, $K_t$ is motor constant, $I_{nl}$ is no-load current, $\omega_{nl}$ is no-load speed, and $b_r$ is viscous damping in the drivetrain and rolling elements estimated based on [41]. Rearranging (2.2) to solve for motor angle $\theta_m$ gives:

$$\theta_m = N\left(\frac{F_s}{K_s} + r_s \theta_t\right)$$ \hspace{1cm} (2.7)

In accounting for drivetrain inefficiencies motor torque is adjusted by an estimated overall conversion efficiency\(^1\). Most modern power electronics enable four quadrant motor control allowing some amount of regeneration of power back into the power bus, and is thus assumed in this applicable in this power conversion. The estimate biases motor torque by a bidirectional power flow with the piecewise-continuous power estimate used in [51], [69]:

$$P_{mech} = \tau_m \dot{\theta}_m$$ \hspace{1cm} (2.8)

$$\tau_m = \begin{cases} 
\tau_m \cdot \eta, & P_{mech} < 0 \\
\tau_m \cdot \frac{1}{\eta}, & P_{mech} > 0
\end{cases}$$ \hspace{1cm} (2.9)

$$\eta = \eta_t \cdot \eta_e$$ \hspace{1cm} (2.10)

where, $\eta$ is overall conversion efficiency, $\eta_t$ is mechanical transmission efficiency, and $\eta_e$ is electric efficiency.

Torque and angle at the motor are then fed into the linearized electromagnetic model of the motor to determine required current and voltage in the motor windings to achieve the specified torque and velocity outputs:

$$i_m = \frac{\tau_m}{K_t}$$ \hspace{1cm} (2.11)

$$v_m = i_m R_m + K_t \dot{\theta}_m + \frac{di}{dt} L$$ \hspace{1cm} (2.12)

$$P_m = i_m v_m$$ \hspace{1cm} (2.13)

where, $i_m$, $v_m$, $P_m$ are motor current, voltage, and power, respectively. $R_m$ is motor winding resistance, and $L$ is winding inductance. With (2.9) adjusting $\tau_m$, the motor

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\(^1\)Many transmissions do not maintain constant efficiency throughout their power band so model fidelity could be increased if detailed mappings were identified for candidate architectures
current $i_m$ is already scaled to account for system efficiency losses.

To identify the necessary operating parameters for powered prosthesis joint actu-
ators, an optimization process similar to [62], [63] is used. The search objective is to
minimize the electrical cost of transport (eCOT) per gait cycle, and while subject to
physical constraints such as motor and power electronics capabilities. The Cost of
Transport (COT) is defined:

$$COT = \frac{\int P_m(t)dt}{\frac{1}{2}mg \cdot \Delta x}$$

where, $P_m$ is power or work-rate, $m$ is user mass, $g$ is gravity, $\Delta x$ is distance traveled.
Units for the COT are $[\frac{J}{N \cdot m}]$. For metabolic studies $P_m$ would refer to the human
metabolic rate and Equation (2.14) would define mCOT, and in electric energetic
studies this $P_m$ refers to the electric power at the motor and Equation (2.14) refers
to eCOT.

The electrical power flow through the motor is integrated over a gait cycle to
determine the electric energy consumed by the motor to produce the specified torque
and velocity trajectory. The piecewise adjustments in (2.9) account for inefficiencies
in the drivetrain and power electronics, thus the motor current (2.11) includes a first
order approximation of regeneration efficiency drops.

To identify ideal component parameters a gradient descent optimization procedure
is applied with non-linear constraints imposed by the motor and power electronics.
The search objective includes minimizing the eCOT:

$$\min \quad eCOT \tag{2.15}$$

s.t. $|\tau_m| - \tau_{allow} \leq 0 \tag{2.16}$

$|\omega_n| - \omega_{n_{des}} \leq 0 \tag{2.17}$

where, $\tau_{allow}$ is the maximum achievable motor torque for specified allowable cur-
rent and specified velocity, $\omega_n$ is lumped-mass natural frequency of the motor and
drivetrain [58], [67], and $\omega_{n_{des}}$ is the desired natural frequency.

For motors with limited manufacturer data availability the following relations are
used to estimate motor capabilities:

$$\tau_{allow} = -\frac{\tau_o}{\omega_o} |\dot{\theta}_m| + \tau_o \tag{2.18}$$
where, $\tau_0$ and $\omega_0$ are stall torque and no-load speed, or the maximum values given by the manufacturer, for a specified nominal bus voltage. The constraints of (2.16) and (2.18) are used to limit torque to the maximum achievable at each desired operating velocity based on the system power bus voltage. This value of $\tau_{\text{allow}}$ can be scaled by the ratio of desired operating power bus voltage and the manufacturer specified voltage, $v_b/v_o$, to identify operating characteristics for a desired system configuration. Increasing the nominal voltage above manufacturer recommendations has been done and does improve power capacity of the motor [70], [71]. The caveat here is that there are dielectric, power electronic, and safety limits that ultimately must be considered when realizing these simulations in actual hardware.

### 2.2.3 Static Evaluation

A first estimate of a parallel spring stiffness comes from a static inspection of 2.1 and 2.3. In a static condition the inertial contribution is neglected. With the goal of reducing load on the motor, the torque contribution $\tau_s$ is negated, and $\tau_p$ is set to equal $\tau_l$, such that:

$$K_p = \frac{\max(\tau_l)}{\min(\theta_l - \theta_{po})}$$

(2.19)

This estimate provides a parallel spring that can support the full bodyweight of the user during controlled dorsiflexion, completely unloading the motor. This is estimate for nominal walking speeds aligns with the parallel spring specifications in [41].

### 2.2.4 Energetic Studies

The series spring in the SEA improves the power and energetic response of the motor [58], [72]. To understand how the series spring affects the actuator response and power capacity of the motor, the nominal spring stiffness output from the optimization is compared against a spring 50\% stiffer, and an effectively rigid spring with three orders of magnitude greater stiffness. The eCOT is used to evaluate the energetic performance of different actuator configurations. Validation of the simulations with built hardware is described in detail in Section 3.4.

Electric cost of transport surface and contour maps are generated for both knee and ankle target trajectories by sweeping across transmission ratio $N$, series spring stiffness $K_s$, parallel spring stiffness $K_p$, and parallel spring engagement point $X_{po}$, and evaluating motor voltage and current response. These studies include the as-built
linear actuator configuration. These maps are similar to the cost of transport maps in [41], but also include 3D surface geometry to better visualize the design space. The energy contour maps demonstrate five different actuator architectures: SEA, SEA with a unidirectional parallel spring (SEA+PS), SEA with variable transmission (SEA+VT), SEA with a parallel spring and variable transmission (SEA+PS+VT), and a SEA with limited dorsiflexion angle (SEA-θ). Also considered was a bidirectional parallel spring, but it is found to be detrimental for the ankle and inconsequential for the knee, and therefore was not considered in detail. The graphical mapping of parameters builds understanding of the relationships between actuator components. Utilizing the RSR from Eqtn. (3.2) candidate motors for these studies are identified. The U10 Plus Kv100 from T-motor [73] was found to be an ideal candidate and is used to compare the following studies.

2.3 Results

Motor behavior informs feasibility of component configurations. Increasing series stiffness improves control bandwidth but can push a motor beyond its capabilities. Section 2.3.1 presents the results of three series spring stiffness studies. Figs. 2-10 – 2-13 visualize the trends and steepness of variance from optimal (presented in the last three Results sections). A summary of the key results of the energy map experiments of Section 2.3.2 are in Tables 2.1 and 2.2.
Figure 2-5: Effects of series spring stiffness on motor performance and ankle gait energetics for 90kg user walking at 2.0m/s. The linearized model of motor capacity is shown against the performance of the actuator with varying series elasticity. Increasing stiffness begins to saturate motor capabilities.
Figure 2-6: Effects of series spring stiffness on motor performance and ankle gait energetics for 90kg user walking at 2.0m/s. Spring deflection affect on both electrical and mechanical energetics.
2.3.1 Series Stiffness

Series elasticity can improve power delivery at the output by storing and releasing energy in cyclical maneuvers. In Fig. 2-5 the motor torque/speed (top) and power (bottom) constraints are plotted. The solid red line is the physical limitation of the motor, below this line is technically achievable though could be limited by thermal behavior that is not considered in this analysis – thermal analysis would require established hardware to verify physical characteristics. The dotted lines of increasing density represent motor trajectories when three different spring configurations of increasing stiffness are each coupled in series with the actuator: $K_s = 271 \frac{kNm}{m}$, $344 \frac{kNm}{m}$, and $271 \frac{MNm}{m}$.

In Fig. 2-6 joint kinetics are shown in solid red, and again increasing stiffness series springs are represented as the increasing dot density plots. The right axis and red lines are the mechanical kinetics of the joint. The top figure showing travel the screw, followed below by the mechanical output power of the joint, and the cumulative mechanical energy of the joint and series springs. The left blue axes and associated dotted lines are from top to bottom: the spring displacement of the actuator, electric power at the motor, and cumulative electric energy at the motor (including dissipative losses). Three nominal spring stiffnesses are shown, a light, heavy and effectively rigid structure. The optimal spring according to Fig. 2-5 reduces motor power and distance traveled requirements by 50% and overall energy consumption by about 30% compared to an actuator without series stiffness.

Preliminary data from level-ground walking experiments with the TF8 actuator, described in Section 3.4, show alignment with estimates. Fig. 2-7 is cumulative electric energy measured at the motor leads compared to simulated data for an equivalent mass subject. Simulated data expected 35 J per gait cycle, while measured mean data resulted in 28 J with a standard deviation of 4.9 J. While nominal results align well there are deviations in the waveform at 5-20% and 55-70% gait cycle.
Figure 2-7: Preliminary validation of the model with the TF8 ankle actuator described in [61] with a human test subject with unilateral below knee amputation. User mass is 75kg, walking 1.5m/s, and $K_s = 378 \frac{kN\text{m}}{m}$. Solid line is the cumulative mean electric energy, with purple shading showing one standard deviation from 28 strides. Simulated data is the dotted line. The experimental procedure for testing is explained in [61].

2.3.2 Parallel Stiffness

Contour and surface maps of reduction ratio and series stiffness affects on simulated system energetics kinematically clamped to an unaffected biological ankle joint undergoing stair descent, level-ground and stair ascent are shown in Figs. 2-8 - 2-10. For clarity, these plots do not consider electro-mechanical drivetrain limitations such as load, thermal or power supply conditions. In reality, there are large infeasible zones in these plots that need to be considered in realizing designs. Fig.2-8 is the SEA configuration without a parallel spring, while Fig. 2-9 includes an optimized parallel spring of $K_{popt} = 3.6 \frac{N\text{m}}{\text{rad}\cdot\text{kg}}$, and Fig. 2-10 includes a parallel spring as specified by the static approximation (2.19) applied to level-ground walking and referred to as $K_{pste} = 6.8 \frac{N\text{m}}{\text{rad}\cdot\text{kg}}$ and as specified in [41]. Data points are shown with constant reduction ratio, $N$, across each parallel spring condition assuming each actuator has a static nominal gear ratio.
Figure 2-8: Contour plots of ankle powered prosthesis electric COT for one gait cycle are affected by series spring and parallel spring stiffness and overall transmission ratio. These analyses are based on the motor T-Motor U10 kV100 Plus [61]. The row shows (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively. These do not include a parallel spring, they are the SEA configuration. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions; some regions may be infeasible.
Figure 2-9: Contour plots of ankle powered prosthesis electric COT for one gait cycle are affected by series spring and parallel spring stiffness and overall transmission ratio. These analyses are based on the motor T-Motor U10 kV100 Plus [61]. The row shows (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively. These analyses are of the SEA+PS include a unidirectional parallel spring with an optimized $K_p = 3.59 \frac{Nm}{rad\cdot kg}$ to support a portion of body weight torque. The parallel spring angular stiffness is normalized by subject mass. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions; some regions may be infeasible.
Figure 2-10: Contour plots of ankle powered prosthesis electric COT for one gait cycle are affected by series spring and parallel spring stiffness and overall transmission ratio. These analyses are based on the motor TMotor U10 kV100 Plus [61]. The row shows (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively. These include a unidirectional parallel spring designed to match the quasi-stiffness during controlled dorsi-flexion as selected by [41] where $K_p = 6.79 \text{ Nm/rad}$, fully supports body weight torque. The parallel spring angular stiffness is normalized by subject mass. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions; some regions may be infeasible.
2.3.3 Limited Range of Motion

Limiting the range of motion during dorsiflexion has an equivalent energy savings as including an optimized parallel spring. The commercial device the BiOM (EmPower) exploits by limiting dorsiflexion to 1° with a hardstop, removing the complexity of a parallel spring engagement mechanism. Comparing the limited ROM energy plots of Fig. 2-11(a-c) with Figs. 2-8 - 2-10 level-ground walking has a 20% improvement, stair ascent has a 32% improvement, but stair descent shows 16% less regeneration capability.
Figure 2-11: Contour plots of ankle powered prosthesis electric COT for one gait cycle are affected by the range of motion of the joint. Here dorsiflexion is limited to one degree to match the capabilities of the BiOM. The experimental conditions are otherwise the same as those in Fig. 2-10(a-c).
2.3.4 Knee

Knee energetics with and without a parallel spring are shown in Figs. 2-12 and 2-13, respectively. In Fig. 2-12 descent, level-ground, and ascent gaits are shown without a parallel spring, and Fig. 2-13 again include descent, level-ground, and ascent including a parallel spring chosen from a search enforcing the use of a parallel spring and included automatic selection of engagement angle. The results of this are shown in Table 3.2.
Figure 2-12: Contour plots of knee powered prosthesis electric COT for one gait cycle are affected by series spring stiffness and overall transmission ratio. These analyses are based on the motor TMotor U10 kV100 Plus [61]. Shown is (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively without parallel springs. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions – some regions may be infeasible for a given motor and power electronics.
Figure 2-13: Contour plots of knee powered prosthesis electric COT for one gait cycle are affected by series spring and parallel spring stiffness and overall transmission ratio. These analyses are based on the motor TMotor U10 kV100 Plus [61]. This row shows: (a) stair descent, (b) level-ground walking, and (c) stair ascent, respectively. These include a unidirectional parallel spring with an optimized $K_p = 0.34 \frac{N m}{rad \cdot kg}$, where the parallel spring angular stiffness is normalized by subject mass. Data points mark COT for a specified nominal transmission ratio that is defined as the optimal for level-ground walking (middle column). These plots do not consider maximum operating conditions—some regions may be infeasible for a given motor and power electronics.
2.3.5 Summary of Results

The Tables 2.1 and 2.2 summarize the results of the energetic maps. In these results, the SEA configuration is used as a normalization factor for each walking condition; the percent deviation in eCOT from the SEA architecture is shown in the table. Lower values are favorable.

For the ankle with a parallel spring the optimized $K_{po}$ value has the potential to improve level-ground walking energetics by as much as 21%. This reduction occurs when the drive-train is also optimized to match the torque contribution of the parallel spring. Stair descent shows a 3% improvement with the parallel spring. However, stair ascent requires an additional 18% eCOT with the $K_{popt}$. The variable transmission ration VT does not show any improvement with an SEA alone, but when coupled with a parallel spring there are improvements across all terrains.

Comparing the limited ROM energy plots of Fig. 2-11(a-c) with Figs. 2-8 - 2-10 level-ground walking achieves a 21% improvement, stair ascent has a 33% improvement, but stair descent shows 76% worse performance.

<table>
<thead>
<tr>
<th>Ankle*</th>
<th>Descent</th>
<th>Level-Ground</th>
<th>Ascent</th>
</tr>
</thead>
<tbody>
<tr>
<td>SEA</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>SEA+PS</td>
<td>-3</td>
<td>-21</td>
<td>18</td>
</tr>
<tr>
<td>SEA+VT</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>SEA+PS+VT</td>
<td>-3</td>
<td>-21</td>
<td>-6</td>
</tr>
<tr>
<td>SEA-θ</td>
<td>76</td>
<td>-21</td>
<td>-33</td>
</tr>
</tbody>
</table>

* The SEA configuration is used as a normalization factor in these results; the percent deviation in COT from the SEA architecture is shown in the table. Lower values are favorable.

The normalized results of knee motion are shown in Table 2.2. For the knee, without a parallel spring there is a near zero energy consumption (eCOT=0.005) for level-ground walking. With a constant reduction ratio, descent produces a net small amount of negative work while during ascent there is substantial (25x) positive work done by the knee actuator than in level-ground walking. With the "optimized" parallel spring there is a 4.8x increase in energy required for level-ground walking, with a COT=0.025 $[\frac{J}{Nm}]$. However, there is a 4.2x increase of energy regeneration during descent and a 23% reduction in energy cost of ascending stairs.
2.4 Discussion

The largest system design affects to system energetics are: on-the-fly adjustment of reduction ratio for varied tasks with a parallel spring, reducing range of motion, or the inclusion of a parallel spring with optimized drive-train. Series spring stiffness changes are also beneficial but are not as task dependent as initially expected. The analysis showed an improvement in level-ground and stair descent with the inclusion of a parallel spring, when \( N \) is also optimized to benefit from the parallel spring torque contribution during controlled dorsiflexion. This result is in conflict with some other studies that have shown that a SEA alone operates with the lowest electric energy consumption amongst other configurations of parallel springs (bidirectional or unidirectional) [48], [49]. It is likely those studies kept the gear ratio constant across all analyses. These results show when the actuator reduction ratio remains the same between SEA and SEA + PS then the SEA configuration does retain the lowest energy operation. Limiting dorsiflexion range of motion, as is done on the BiOM has an equivalent energetic benefit as an SEA+PS with optimized reduction ratio of 21% energy savings compared to a full ROM SEA joint. Reducing ROM is a more cost-effective way of extending battery life than adding the complexity of a uni-directional parallel spring to hardware, but it affects the natural kinematics of a user. The trade-off in exchanging walking duration for user feel is an engineering consideration that has strong affect on desired user experience.

In level-ground walking an optimal reduction for the SEA ankle is about 65:1 with the specified motor combination used in this study, and there is little energetic improvement with raising the reduction-ratio for stair ascent or descent. When a parallel spring is employed there is a pronounced improvement in level-ground walking, however, to handle stair ascent there is a need to double the gear ratio by shifting from \( N=37 \) to about \( N=80 \), else there is a large energetic cost.

A varying reduction ratio in the ankle could benefit task changes from level-ground...
walking to stair ascent when a parallel spring is included. The eCOT minimum-well remains fairly flat and constant for an SEA. The parallel spring energetic minimum-well sharpens and shifts toward lower reduction-ratios with increasing parallel stiffness, potentially warranting adjustable transmissions. For example, in Fig. 2-10(e,f) the level-ground SEA+PS ankle system prefers $N=37$, but stair ascent energy costs twice as much as level-ground and 18% more than an SEA. If the reduction-ratio could increase to $N=83$ the SEA+PS would use 6% less energy than an SEA.

Similar and more pronounced behavior is found in the case of the knee actuator where the walls of the energy-well change shape drastically. In stair descent adjusting $N=60$ rather than the level-ground optimal reduction ratio of $N=27$ would enable a 5x improvement in regeneration and a halving of the energy required for stair-ascent. Some recent designs have looked to take advantage of a variable gear ratio knee by adjusting the moment arm length for different terrains, though these are not yet powered [74].

Screw driven actuators naturally vary their reduction ratio as the moment arm passes along the swept arc of the rotary output joint. This means designs should be careful to recognize that decreasing gear ratios quickly ramp into steep energetic consequences. In these systems the linkage geometry itself should also be optimized to match the power-stroke of the mechanics with desired gait trajectories.

The series spring contributes to flattening of the motor energy for the first half of the gait cycle, but electric energy is consumed during the acceleration and deceleration of the knee in swing. The parallel spring contributes a load reduction during a negative energy portion of a gait cycle, quickly followed by a large positive energy gait phase, such as in the ankle.

The knee, though, does not express this same loading behavior during level-ground walking, leaving the parallel spring less effective for that case. In level-ground walking a parallel spring can mostly operate as a bumper against which the knee may swing into during early swing and bounce from as it nears late swing. In early swing the knee is lifting the inertial load of the lower tibial shank against gravity. While a parallel spring could potentially provide energy redirection at mid-swing, it must also be fought by the motor that is meanwhile lifting the leg and controlling its velocity – something difficult to tune with a passive spring – showing in simulation an unnoticeable difference in energy consumption. A clutched parallel spring could overcome the deficiencies in stair ascent or descent where there is energetic benefit to a comparably soft parallel spring located near the max knee flexion point. In stair ascent and descent the large joint angle excursions occur while the joint is
loaded similar to the ankle during controlled dorsiflexion exploiting the benefits of the parallel spring. Similar concepts that look to leverage clutched energy storage in the knee have been employed by [64], [75]. A potential alternative may be simply to include parallel unidirectional bumper springs at the ends of travel of the knee to help support the body weight and propel the shank in reversing direction conditions that happen twice throughout the gait phase.

A note on the limitations of these studies. These studies include the as-built linear actuator configuration and should be expanded to consider purely rotational, constant gear ratio actuators as well as potentially more sophisticated continuously variable transmission configurations. Not all design configurations in Figs. 2-10, 2-13 are realizable in hardware due to invalidating the constraints defined in (2.15-2.18) and Figs. 2-5 and 2-5. These constraints limit the achievable operating regimes and clearly define real-world component specifications, but they limit clarity in visualizing the energetic maps and are left for the reader to apply to their own design process.

2.4.1 Series Stiffness

Adjusting series spring stiffness between larger and smaller mass walking subjects can change energetic outcomes: e.g. a 90kg person walking on a spring specified for a 75kg user results in a 7% increase in eCOT, and could potentially cause increased spring excursion beyond the capabilities of the hardware. This emphasizes the importance of changing spring stiffness to suit the mass of the user, or even tuning stiffness to desired max walking speeds. Though not examined in this study, [49] suggests that due to differences in gait profiles series stiffness must not adjust for varied running speeds, but $K_s$ does have impact on walking speeds.

The energy contribution of the spring is evident in Fig. 2-5 where motor capacity is mapped, and Fig. 2-6 where spring deflection is in the top row, spring power across the gait in the second row, and cumulative energy of the motor and springs demonstrate the effects of different stiffness springs in the bottom row. The torque and power requirements of a fully stiff or direct-drive system approximated as the $K_s = 271 \frac{MNm}{m}$ finely dotted line extends beyond the solid line linearized motor constraints. In this case the motor cannot achieve the specified gait trajectories. Adding a spring in series (effectively reducing the drive stiffness) brings the motor requirements within the voltage and current limited constraints of the motor and power electronics. Similarly, increasing spring deflection enables the spring to store and release energy, reducing the power demand on the motor. The cumulative energy in the bottom row of Fig.
2-6 also shows that reduced stiffness dampens the power requirements of the motor. Likely some of the difference between simulation and experimental results in Fig. 2-7 are due to an overestimation of total actuator series stiffness, in addition to potential stiction and damping in the drivetrain, but may be most dramatically affected by over estimates of motor, drivetrain and output inertial characteristics. When this actuator was designed the structural stiffness specification was modeled after human bone, which deflects up to several millimeters during heavy loading [76]. The additional series elasticity of the structure was not included these simulations. The reduced overall series elasticity likely more closely resembles the softer spring in these simulations. The energetic analyses are sensitive to inertial characteristics due to the quadratic velocity multiplier and the frequent motor reversal required in cyclical gaits. The simulation parameters include a large inertial output link that likely drives the energetic estimate to be more conservative than necessary.

This variation between experiment and simulation energetics may also be effected by using a finite-state machine controller to estimate a biological waveform and comparing it to a gait profile from unaffected persons. Though biomimetic, the finite-state machine walking controller does not directly track biological trajectories. Additionally, efficiency of the drivetrain was overestimated in the simulations. Nonlinearities in system behavior may also be culprits: the regions of deviation are during relatively low-torque conditions where stiction in the drivetrain may be damping motor response. Additionally, this being preliminary data, further tuning of the low-level control algorithms is expected to improve torque tracking performance.

### 2.4.2 Hypothesis Revisited

The initial hypotheses expected a parallel spring to be helpful in level-ground walking but to be costly for the larger range of motion maneuvers of stair ascent and descent. These hypotheses have mixed results: it was incorrect regarding the parallel spring where it turns out that during stair descent the parallel spring slightly (-3%) improves energetics of the system rather than hurts it, and similarly, the limited range of motion SEA-θ has a substantial 33% energetic improvement for stair ascent. In support of this hypothesis, during stair ascent with a SEA+PS the energy cost increases 18% compared to the SEA alone. The parallel spring, though effective for level-ground walking may, if over-sized, actually require the motor to operate outside of its achievable working regimes for stair climbs and other similar large range of motion activities. Finally, the SEA-θ has a 76% increased COT during stair descent. It
should be noted that this result is less negative than the SEA, but is still technically a negative COT. A surprising result is a variable transmission ratio combined with a parallel spring can provide energy improvements across all terrains for knees and ankles. Future studies should verify what conditions can actually be achieved by the hardware to further validate the simulations.

### 2.5 Conclusions

Realization of hardware can force variance from target component specifications. Limitations in commercially available components such as ideal gear ratios and springs can force a design away from strictly optimal configurations. Previous researchers have explored energetics of level-ground walking and running, but to the authors knowledge the field had not yet explored the energetic design trade-offs that enable stair ascent or descent and limited range of motion. This analysis aims to provide a map to understand the design trade-offs for lower-extremity powered prostheses in varied terrain, in addition to a means to search for ideal components when sample joint trajectories are available. Strict adherence to optimization results from a continuous search domain can leave one blind to feasible or infeasible design regimes, and is generally not possible to achieve when building with discrete components. Using both tools together, the energetic map and optimization procedures, can provide insight into design trade-offs for future actuator architectures.

For the ankle either a parallel spring or limited dorsiflexion can each improve level-ground walking. Both methods produce $\leq 21\%$ improvements, but each one has negative consequences for either stair ascent or descent, respectively. A variable transmission alone provides minimal improvement due to the flat energy well of the SEA on its own. A surprising result is that a combination of SEA+PS+VT ankle can make improvements across all terrains studied.

The knee benefits substantially from a variable transmission, with a $\geq 500\%$ increase in negative energy during stair descent, and a more conservative 38% improvement during stair ascent over an SEA. This analysis finds a parallel spring at the knee is energetically costly in level-ground walking but equivalently helpful in the larger range of motion maneuvers of stair descent and ascent. Limiting knee flexion would simply negatively affect gait so it was not considered.

An SEA ankle is relatively simple and likely lighter weight than an ankle that combines two additional mechanisms. A knee has much larger benefit than the ankle from a variable transmission during stair descent. For the initial application of these
methods a simple lightweight SEA hardware configuration is chosen that attempts to compromise between the needs of both a knee and an ankle actuator, such that it can be implemented for both applications, described in Chapter 3 and [61]. However, it does lead to the question: would it be possible to build a powered prostheses that exploits the advantages of each mechanism? Is the added hardware worth the added mass? This is for the designer to determine.

A secondary question is can the negative energy truly be utilized? In mechatronic control there is a chance this energy can be redistributed through a four quadrant motor driver and back into the power bus. However, due to power transfer efficiency full regeneration is not always achievable. In hardware the system efficiency is found to be about 60%, meaning the low power regeneration modes are not necessarily fully utilized.

Finally, improvement to the analysis method could be warranted for higher fidelity analyses. The kinematic clamped analysis is a good starting point, however, its constraints are aggressive and the dynamics simplified. An actuator model that includes controller effort and simulation with shank inertias and walking dynamics would likely more closely resemble reality. Modeling shank and even proximal segment dynamics would provide more accurate inertial and coriolis forces applied at the actuator. These dynamics effect the controller effort. The method of disqualifying designs based on behavior that fails search constraints may be limiting. Evaluating against controller effort with a deviation allowance could allow generally better agreed behavior across the wider trajectory.
Chapter 3

Design

Hardware is hard. Where one solution fails another may succeed.
3.1 Introduction

More than a decade since the first powered ankle prosthesis, research in lower-extremity rehabilitation robotics remains limited by a lack of commercial and academic hardware platforms capable of producing full biological dynamics. The commercial off the shelf powered ankle and knee systems are not able to provide biologically accurate kinematics, closed-loop torque control, nor do they enable access to the underlying control systems for researchers to explore fundamental control paradigms [29], [56]. To improve upon the commercial options, a growing handful of academic research groups have been building individualized platforms for study. Most of these academic platforms have remained within their own labs, requiring each lab to build its own hardware. This study aims to describe a method of design validated with an architecture that may enable a broader audience to achieve well performing hardware. The following describes the mechanical design of an autonomous, untethered, wearable, series elastic actuator topology that aims to achieve biologically relevant kinetics and kinematics of both a human knee and ankle. By reducing hardware complexity the design remains within the mass bounds of the human equivalent leg segments. The architecture presented as-built can achieve biomimetic knee and ankle trajectories, and it can also be configurable for high efficiency as either a knee or an ankle, by replacing a single flat-plate composite series elastic element.

The motivation to build powered prostheses is to: a) generate the roughly 0.1-0.2J/kg net positive work done by the ankle during walking that cannot be provided by an un-powered passive ankle-foot prosthesis [39], b) reduce the energetic cost of walking [1], [42], c) to improve quality of feel and ground control helping to reduce falls, and d) normalize gait to reduce social stigma of pathological gaits as well as early onset of co-morbidities. Nearly sixty percent of the energy contribution of forward motion in walking comes from the ankle [40], [77]. Passive (spring or damper type) prostheses are unable to provide the net positive work of the ankle, as shown in Fig. 1-2. This lack of work at the joint contributes to increased effort from more proximal and contralateral joints, leading to decreased mobility for some users [1], [4], as well as increased impact loading on the contralateral leg. The effects of this lack of power can lead to comorbidities such as the early onset of osteoarthritis, osteopenia, and osteoporosis [11].

To replicate the work done at the biological ankle Au, Weber, and Herr built the first powered ankle prosthesis. It used a series elastic actuator (SEA) [58] configured with a parallel spring to improve torque capacity, bandwidth, and energy consump-
tion through controlled dorsiflexion and powered-push-off stance phases of walking. The actuator was shown to contribute energy during powered push-off and thus improved the metabolic cost of walking [42]. Various actuators have since been designed following a similar actuator topology [43]–[47]. Though powered devices have been shown to reduce the metabolic cost of transport (mCOT), they are generally heavier than their passive counterparts. Always, we wish to reduce weight due to poor suspension (means of attachment between the body and prosthesis). Micro-processor devices exist that do not provide positive work, but do have intelligent mode switching to control joint orientation or damping modes. Finding the balance between power, weight and functionality is an active area of research in the field.

There are two device types that fit between powered and micro-processor controlled devices, the quasi-passive and the semi-powered polycentric ankle prostheses. The quasi-passive relies on a spring to store and return energy, but has a means to adjust the neutral position of the spring to enable either additional energy storage and release, or adjust the toe position for toe clearance during swing-phase. The quasi-passive can in some cases even adjust toe position for different terrain orientations such as slopes. These devices though, do not provide power for toe push-off. Instead, they trade-off weight for functionality. One such device is the Variable Stiffness Quasi-Passive Ankle (VSPA) that has a motor to vary the sprung length of a cantilever beam that is point loaded by a cam built into the ankle [78]. This device is lightweight and has low clearance height, but is not able to provide the additional power at push-off that is characteristic of unaffected biological gait.

The poly-centric ankle is a semi-powered ankle that is not capable of providing net positive work, but it is able to adjust ankle angle throughout the stance phase of walking in order to improve comfort for the user. A polycentric actuator uses a four-bar linkage to shift the ankle position through stance and swing. The device does not attempt to match biological joint motion as the instantaneous center (IC) of rotation tracks an arc much greater than biological joint rotation and translation, but, instead the concept is to exploit this behavior to better align the ground reaction force with the tibia to reduce moment induced loads on the sockets of transtibial amputees – a loading condition that can cause pain in the residual limb. The UMass Amherst Ankle uses a polycentric actuator design with a roller screw and the field’s favorite motor the Maxon EC-4 Pole. This device provides a lower clearance height than most at 184mm, though it supports only a limited range of motion at 10 degrees, and does not provide powered push-off. Its focus is adjusting reaction force location to reduce moment on the stump of the user to improve comfort, rather than explicitly
to produce walking power [79]. The p2 Ankle from [80] similarly boasts impressive specifications of a 120mm build height, 1.0kg mass and 55 degree range of motion. It also uses the EC-4 Pole, this time attached to a helical spur gear and roller-screw drive-train. The combination of helical gear, followed by a roller screw likely nearly eliminates backlash. However, for this system torque control is dependent on measuring motor current, which is difficult to measure accurately through such a large drive-train. Effective closed-loop control performance cannot be expected when the torque command is translated through such a drivetrain and the constantly varying transmission ratio due to the linkage mechanics. They present static tests at four joint angles showing 15% inaccuracy in torque measurements – this is substantially low for a torque control device. Neither of the polycentric systems are closed loop torque control device, nor are they capable of providing net positive power.

Powered prostheses make use of robotic actuators that can provide kinetics and kinematics that attempt to mimic the behavior of biological joints. These actuators provide positive work and closed-loop dynamic control that helps return the body to normative behaviors and can reduce metabolic cost of transport [32]. Position control has been traditionally used in swing phase, because many systems are operating in an open-loop torque mode. In open-loop control torque is commanded at the joint but without a load (e.g. the ground) the joint may slam into a hardstop. Position control in swing is a problem due to its limited stability when collisions occur, such as an unexpected heel or toe-strike event such as may happen when someone trips. Position controllers can go unstable with impact loads, and so a sudden fall response from a user might lead to system instability. To manage controllability during impact Hogan showed that impedance control is a means of providing controller stability in robotic systems with contact dynamics, while also mimicking natural neuromuscular behaviors [33], [34]. The impedance control effectively simulates a virtual spring and damper system, allowing control of rebound and energy dissipation, and being robust to impact.

There exists only one commercially available powered prosthesis ankle device capable of providing impedance controlled power delivery: the emPower from Ottobock (previously BionX) [30]. The emPower system provides up to 165Nm of torque at push off, has a clearance height of 220mm and weighs 2.3kg, but has a limited total range of motion of 25 degrees of plantar flexion and zero dorsiflexion [56]. This lack of dorsiflexion improves electric energy consumption and provides safety when unpowered, but handicaps its ability to provide natural kinematics during the controlled dorsiflexion phase of early stance which can be as high as 14 degrees even
during level-ground walking [81]. Further, it cannot operate on substantial inclines, nor adjust for varying shoe heel heights. Half a dozen research platforms exist with similar specifications. The *Walk-Run* is of very similar construction to the original Au and Weber powered ankle [42] and emPower [56] system. The *Walk-Run* includes: a EC-4 Pole 200W Maxon motor, belt drive and ball screw drive with a 1.9kg mass, 40 degree total ROM and 10 degrees of dorsiflexion, but has an unwieldy build height of 300mm [47]. The *AMP-Foot 2.0* and *AMP-Foot 3.0* both come in at a hefty 3kg, a total ROM of 50 degrees with 20 degrees of dorsiflexion and utilizes a linkage and clutching mechanism mounted distally in the foot [43], [44]. The *Sparky* actuator provides limited information with claims of an actuator mass at 0.95kg but it is not clear if this includes the mounting hardware required by its vertical arrangement, nor is any torque, velocity or total range of motion data published beyond tracking an ankle angle walking trajectory [45], [46], [82]. The Vanderbilt generation 2.0 knee and ankle systems utilize a three stage belt and chain drive system and with outrunner EC-Flat 60 Maxon motors to drive the joints. Kinematics and kinetics of normal walking are achievable with this system, with ankle joint range limited by the addition of an energy conserving parallel spring. However, the system mass is on the higher end of the acceptable range for equivalent body segment and reported torque and power are on the lower end 110Nm [83], [84].

Though well designed for flat ground, and self-selected walking speed, these ankle systems quickly hit kinematic limits at higher walking velocities where stride length and joint angle extension increase [57] and these range of motion limitations are compounded when encountering sloped surfaces where nominal joint angle orientation must adjust to surface angle. Mean data from nine subjects walking at 0.75m/s to 2.0m/s shows mean maximum dorsiflexion angle increases from 8.3 to 19.3 degrees while plantarflexion angle changes from 19.8 – 14.9 degrees. Since the American Disabilities Act (ADA) compliant power wheelchair ramp slope specification declares a maximum 7.1 degree inclination, fast walking up such a ramp eliminates or limits proper kinematic function in all the ankles described except for the Vanderbilt legs even without considering user specific orientation preferences. Certainly the field for robotic prosthetic leg development should do better than the wheel chairs it drives to improve upon!

Powered knee actuators tend to be large and disregard clearance height that would allow for separate powered ankles. Many also forgo full torque control throughout the entire gait cycle, and instead provide closed-loop control only during specific phases. The commercially available Össur Power Knee weighs 3.1kg, has a clearance height of
270mm, and from the datasheet appears to only support limited position control in specific states, rather than provide full torque controlled impedance modes [29]. The Vanderbilt generation 2.0 knee is a 2.7kg mass with a 85Nm max load, but power capability is not provided. The University of Utah Romeo knee sports an actively variable transmission (AVT) that can be adjusted when the system is unloaded and is combined with a hydraulic variable damping module from Ottobock to provide damped control during stance. The system has no force measurement and so operates in open loop torque mode and closed loop position control, so impedance control is not feasible. The tradeoff to save mass is to design the power module to operate only in specific phases such as for stair climbs, and a limited 90 degree range of motion, but the added complexity of variable moment arm and damping system causes the system to still come in at 1.7kg and a 290mm build height. The MIT Clutched Series Elastic Actuator (CSEA) has two different springs for compression and extension and makes use of a brake on the motor shaft to limit the energy expenditure for holding torque against knee flexion [66]. This system was designed primarily to conserve energy rather than exploit control strategies and has a 2.7kg mass and 280mm clearance. Finally, a biomimetic approach to impedance control was proposed with the MIT Agonist-antagonist knee actuator that uses two series elastic actuators acting against each other to set the joint stiffness[85] . This double actuator system has substantial torque capability at 130Nm, but mass of 3kg and build height of 330mm. All of the above mentioned systems (except for Romeo) have 120 degree range of motion, less than biological, but enough to support sit-to-stand and cross-legged sitting. What is most apparent is that in each configuration a strict design objective, such as energy efficiency, drives the design while other constraints such as build height, or complex control strategy capacity are more liberally applied.

Combined knee and ankle prosthetic systems can enable coordinated powered motion at the knee and ankle allowing for more complex and biomimetic control techniques such as neuromuscular reflexive control. The trade-off that nearly all these systems exhibit (with the exception of the Vanderbilt Gen. 2 leg) is they are designed to a single specific height, restricting usability to a very limited patient group. The third generation Vanderbilt leg is a combined knee and ankle system and appears to use smaller motors and larger gear ratio to achieve system dynamics capable of running and stair ascents. Though larger gear ratios tend towards more noise, the system appears well specified in its ability to achieve torque and range of motion for stair climbing and even running for median mass patients. The major trade-off with this system is that it appears to have a fixed height of 452mm limiting smaller patients.
who are well matched to the torque capability of the system from making use of the device. The AMPRO 3 is a fixed height combined knee and ankle prosthesis that makes use of harmonic drives, and spiral series elastic springs for torque control [86]. The system mass is a hefty 5.95kg, a value larger than the equivalent body segment should accommodate considering that its design specification of peak 123Nm torque at the ankle supports only a roughly 70kg, medium sized, individual walking at nominal walking speeds. A novel feature of the AMPRO is that there is a passive subtalar degree of freedom, though its range is limited to prevent center of mass from passing over the zero moment point.

Comparison of different actuators in this space can be difficult as published data does not always fully represent the system capability, e.g. the common metric of peak torque is quite different from delivering torque at desired velocity (the power capability), and more so the magnitude of controlled torque bandwidth. In lower extremity prosthetics the time domain is often normalized to percent of gait, which is effective for comparison, except that biological trajectories generally scale with walking speed, so a gait state is less explicit than overall power delivery on the time domain. Similarly, the mass normalized torque and power output that is preferred in biomechanics is helpful for normalizing results, but can be deceiving for actual maximum actuator capabilities. A measured 200W prosthesis output for a lightweight person may result in a high power/kg rating for the actuator, but that same output ratio may not be achievable on a heavier mass person. For this reason, overall power and torque capabilities may be better metrics to evaluate actuator specifications. For robotic and prosthetic systems perhaps an additional helpful verification of system capability is to evaluate the ability to track torque and velocity trajectories on the time domain such that real-world capability is explicitly demonstrated. The reflected-speed-rate of actuators – the relationship to power and drivetrain inertia – limit their ability to accelerate and decelerate at load, and is an important metric for series elastic actuators in particular, where the motor and drivetrain must get out of the way of the spring when reversing direction under load. System bandwidth is often published, but the magnitude at which the bandwidth is evaluated is not always specified. To add ambiguity, many of these actuators utilize linkage drives whose overall gear ratio are joint angle dependent, such that bandwidth may vary throughout the range of motion.

Many of these designs have focused around flat, level-ground walking, where primary kinetic behavior occurs as a large power maneuver during powered plantar flexion. Commercial devices such as the BiOM [87] even limit joint range to zero
dorsiflexion in order to reduce electric energy expenditure. To improve control bandwidth and electrical energetics parallel springs are frequently used, at a cost of range of motion and terrain adaptability. Though well designed for flat ground, and self-selected walking speed, these systems quickly hit kinematic limits at higher walking velocities and varied terrain. At higher walking speed, stride length and joint angle extension increase[57]. Mean data from nine subjects walking at 0.75m/s to 2.0m/s shows dorsiflexion angle increases from 8.3 to 19.3 degrees, and plantar flexion angle changes from 19.8 to 14.9 degrees. These range of motion limitations are compounded when encountering sloped surfaces or stairs. During stair descent [65] found a 21 degree mean dorsiflexion angle during stance and 40 degree mean plantar flexion angle in swing phase as the toe approaches the next step. Since the American Disabilities Act (ADA) wheelchair ramp slope specification declares a maximum 7.1 degree inclination, fast walking up such a ramp eliminates or limits proper kinematic function in all the ankles described, except for the Vanderbilt legs, even before considering user specific orientation preferences.

This chapter presents the mechanical design of the MIT Reaction Force Series Elastic Actuator TF8\(^1\) (MIT RFSEA), an actuator that aims to improve upon the field, to achieve biomimetic kinetics and kinematics equivalent to that of a human knee and ankle while remaining within the mass constraints of equivalent leg segments and being customizable to differing user needs. The architecture presented is built to be a highly functional research platform capable of achieving the design specifications for both knees and ankles with a 90kg person walking at 2m/s (on the edge of jogging). The actuator is adjustable such that it can be tuned to individual user needs by replacing a simple flat composite spring. For example, a person weighing 50kg could manage many more steps in a day than a 100kg person by adjusting the series elastic element stiffness to match their mass requirement. Similarly, a 100kg person who prefers a slower pace could make use of a softer spring to also reach more steps. A heavier person walking fast will, nonetheless, require more energy. That said, the TF8 is designed to be energetically optimal for all users by making use of the simulation framework described in the previous chapter and in an accompanying paper [88]. As described above, the design process simulates motor and series elastic actuator dynamics clamped or locked onto a reference joint trajectory. By plotting the effects of user mass, series stiffness, reduction ratio and motor properties a combination of component specifications that enable a compromise design is found. There exists

\footnote{The TF8 name comes from being the eighth major design iteration of what was originally to be a powered knee prosthesis for transfemoral, or above knee amputee (AKA) subjects.}
a design that can operate as both an ankle and knee – possibly slightly saturating velocity at the knee, and torque at the ankle for one or two data points, and yet being adjustable such that it could be tuned to perform exceptionally for an individual task given the right components. Presented are the optimization procedures used to design the mechanical linkage geometry, and the series springs as implemented. Further, the non-straightforward operating behavior of this moment-coupled cantilever-beam reaction force series elastic actuator are described. Benchtop experiments validate the performance of the actuator, as do human walking experiments. Preliminary results of below-knee, level-ground walking demonstrate positive net-work is generated at the joint with torque and power tracking biological trajectories with a $R^2 \geq 0.85$. Stair ascent and descent also demonstrates biological torque trajectory morphology with one test subject. Finally, the TF8 is shown making introductory steps as a combined knee and ankle powered prosthesis. Discrepancies between expected and measured behavior are also discussed with future recommendations for improvement.
Figure 3-1: The TF8 Actuator is designed to operate as either a knee or ankle powered prosthesis – shown here configured as an ankle prosthesis. Image used with permission, copyright Andy Ryan.
3.2 Mechanical Design

Mechanical design is often a manual, brute-force search across a wide design space composed of conflicting criteria. The process is necessarily an iterative process, where the overall architecture only crystallizes as understanding of each of the subsystems achieve increasing resolution. To reduce the search parameters common practice is, when possible, to iterate upon previous designs. This approach can accelerate the design process, though it also has the potential to leave designers in a local optima, rather than the global optimum when the underlying parameters are not re-evaluated for changing applications. For this effort the design objective is to find the minimum electric energy consumption per stride for a given configuration of components.

The studies in Chapter 2 lead to the understanding that series elasticity is an important part of the architecture as a mechanical energy storage and release mechanism. The independent variables to search for are then: the spring, motor and reduction ratio. In previous designs, a unidirectional parallel element has been used to improve control bandwidth and electric energetics in level-ground walking by supporting the controlled dorsiflexion phase of stance [42], [44], [60]. However, the parallel element also increases system complexity and mass. The analyses in Chapter 2 lead to a conclusion that while energetic improvements exist for an ankle, they are not substantial enough, further complicate a design, and as such should not be included in this design that aims to achieve ankle and knee capabilities. To design and build an autonomous lower-extremity powered prosthesis capable of providing biologically equivalent kinetic and kinematic trajectories while minimizing electric energy consumption the resulting design should be: equal to or lighter weight than an equivalent normative biological limb segment, enable physical tasks beyond level-ground walking such as sit-to-stand, inclinations, stairs, jogging, or fall recovery, and if possible be capable of functioning as both a knee and an ankle prosthesis.

3.2.1 Performance Specification

The design specification for the range of motion, torque, power, and system bandwidth, is determined by normalizing and scaling by body mass a total of 1005 unique gait cycles of walking data from nine able-bodied subjects collated from [9]. For this design study performance targets are based on a 90kg user walking at 2.0m/s – a pace on the boundary of walking and jogging. The resulting joint trajectories, in Figs. 3-2 and 3-3, are for ankles and knees respectively. The maximum extents of these profiles are used to help determine the actuator designs specification. The design goal to
Table 3.1: Actuator Design Specification

<table>
<thead>
<tr>
<th>2m/s</th>
<th>Ankle</th>
<th>Knee</th>
<th>Target+</th>
<th>Result</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Range of Motion(^1)</td>
<td>19-0-19</td>
<td>70</td>
<td>45-0-65</td>
<td>35-0-75</td>
<td>Deg</td>
</tr>
<tr>
<td>Velocity</td>
<td>6.0</td>
<td>8.6</td>
<td>7</td>
<td>6.8(^*)</td>
<td>rad/s</td>
</tr>
<tr>
<td>Max Torque</td>
<td>160</td>
<td>118</td>
<td>175</td>
<td>175</td>
<td>Nm</td>
</tr>
<tr>
<td>Max Power</td>
<td>552</td>
<td>313</td>
<td>550</td>
<td>350</td>
<td>W</td>
</tr>
<tr>
<td>Bandwidth Magnitude</td>
<td>82</td>
<td>73</td>
<td>82</td>
<td>82</td>
<td>Nm</td>
</tr>
<tr>
<td>Bandwidth Frequency</td>
<td>4.8</td>
<td>6.0</td>
<td>6.0</td>
<td>6.2</td>
<td>Hz</td>
</tr>
<tr>
<td>Segment Mass(^2)</td>
<td>2.6</td>
<td>2.6</td>
<td>2.0</td>
<td>1.6</td>
<td>kg</td>
</tr>
</tbody>
</table>

\(^1\) Actuator specification is determined based on a 90kg user walking at 2m/s.
\(^2\) Range of motion as shown: dorsiflexion - neutral - plantar flexion. For a knee walking requires up to 72°, sit-to-stand and stair ascent require \(\leq 95°\), and 21-0-40 for an ankle [65], [89].

\(^*\) The equivalent body segment mass is 2.9% of a 90kg person [68], [90].

\(^\text{Calculated due to power electronics limitation, based on observed behavior. }\)
\[\omega = 48V \cdot \eta \cdot 100Kv \cdot \frac{2\pi}{60} \cdot \frac{1}{N}.\]

enable performance of daily tasks and walking on uneven terrains requires a larger range of motion than that defined only by walking [89] and are included as additional joint range of motion directives. The range of motion is further extended to include the maximum ROM of ankle and knees during stair ascent and descent with data from [65], plotted in Figs. 3-4 and 3-5. The methods described in Chapter 2 and [88] are used to establish the control bandwidth specifications. The design parameter specifications are outlined in Table 3.1.

The mass allowance of the device is derived by evaluating the mass of equivalent leg segments. The mass of a human calf and foot segment is 5.82% of body mass, while the mean calf mass is 4.35% and the mean foot segment mass is 1.47% [68], [90]. A reasonable assumption of the mass accommodation of a prosthetic ankle-foot is the mass of a foot and one third of the calf mass, or 2.9% of body mass. For a 90kg subject this allots 5.2kg for a combined above-knee and below-knee prosthesis, and for an individual below-knee prosthesis 2.6kg is an equivalent mass allowance. Despite these allowances, it should be noted: prostheses should generally be as light as possible due to limitations with prosthesis suspension – the mechanical attachment to the body is an unsolved problem.
Figure 3-2: Ankle joint trajectories of a 90kg person walking 2m/s on level-ground. The maximum extent of these motion profiles are used to help define the actuator design specification. The ankle joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.
Figure 3-3: Knee joint trajectories of a 90kg person walking 2m/s on level-ground. The maximum extent of these motion profiles are used to help define the actuator design specification. The knee joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.
Figure 3-4: Ankle joint trajectories of a 90kg person walking down stairs, from [65]. The solid line is the mean, the grey fill is one standard deviation from mean. The maximum extent of these motion profiles are used to help define the actuator design specification. The ankle joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.
Figure 3-5: Ankle joint trajectories of a 90kg person walking up stairs, from [65]. The solid line is the mean, the grey fill is one standard deviation from mean. The maximum extent of these motion profiles are used to help define the actuator design specification. The ankle joint data shown are: (a) torque, (b) angle, (c) power, (d) velocity.
3.2.2 Simulation and Component Specification

A gradient descent optimization procedure with non-linear constraints is used with numerical models of the mechanical and electrical system dynamics to find the electric energy optimal configurations of components: motor, transmission ratio, series stiffness \( K_s \), parallel stiffness \( K_p \), and parallel spring engagement point \( X_{p0} \). The initial search pass is a continuous search across the entire design space. Due to physical hardware and manufactured component limitations further discretized searches are performed based on the most closely acceptable available components. Mean torque, velocity and angle trajectories aggregated from [9] are mass normalized, scaled and kinematically clamped to the dynamic equations for a series elastic actuator. To identify the necessary component parameters for powered prosthesis joint actuators, an optimization process similar to [62], [63] and described in Chapter 2 and [88] is implemented. The search objective is to minimize the electric Cost of Transport (COT) for a gait cycle and is evaluated against the constraints of: lumped-mass natural frequency, motor voltage and current. The results are sorted by minimum electric energy consumption. What emerges from the results is an actuator configuration that can achieve knee and ankle trajectories. The design can be tuned to achieve optimal minimum energy for each joint by either an adjustment to the spring stiffness or gear ratio. In defining the design architecture a linear ballscrew and its necessary linkage are chosen as the actuator drive train. After component specification a secondary optimization searches to identify the joint linkage geometry and minimal mass spring that most closely match the desired output specified by the search.

Top results of simulating a 90kg person walking at nominal 1.25m/s and very fast 2.0m/s speeds are shown in Table 3.2. Due to manufacturer limitations ballscrew availability is highly discretized. At these "miniature" sizes leads are available in 2, 3, 4, 5, and 10mm increments. Thus the as-built 5mm lead system is shown in the table with the next available optimal screw leads shown of 4 and 10mm. The spring could be optimized to an individual user, but for comparison is kept constant across these results at a stiffness of 378kN/m. The results of this table can also be clearly visualized in Fig. 3-6, where a screw lead of 5mm is marked with dotted line on the overall transmission ratio axis. A screw lead of 10mm would equate to a transmission ratio of 27:1. System overall energy conversion efficiency is assumed to be 90% mechanical efficiency and 90% electric conversion efficiency. The effective overall cycle efficiency of 60% is estimated based on the bidirectional power trajectory conversion as described in Equations (2.3).

Overlaying the ankle energetic map from Section 2.3.1 with the knee energetic
Table 3.2: Actuator Simulation Results

<table>
<thead>
<tr>
<th>Joint</th>
<th>Speed(m/s)</th>
<th>Lead(mm)</th>
<th>$K_s$(kN/m)</th>
<th>$E$(J)</th>
<th>$V_{max}$</th>
<th>$V_{rms}$</th>
<th>$I_{max}$</th>
<th>$I_{rms}$</th>
<th>$P_{max}$</th>
<th>$F_s$</th>
<th>$\omega_n$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ankle</td>
<td>1.25</td>
<td>4</td>
<td>378</td>
<td>32.8</td>
<td>22.5</td>
<td>7.6</td>
<td>29</td>
<td>14</td>
<td>305</td>
<td>3086</td>
<td>3.3</td>
</tr>
<tr>
<td>Ankle</td>
<td>2.0</td>
<td>4</td>
<td>378</td>
<td>69.0</td>
<td>22.6</td>
<td>10.3</td>
<td>34.5</td>
<td>16.4</td>
<td>538</td>
<td>3523</td>
<td>3.3</td>
</tr>
<tr>
<td>Ankle</td>
<td>1.25</td>
<td>5</td>
<td>378</td>
<td>33.9</td>
<td>18</td>
<td>6.2</td>
<td>35</td>
<td>15.4</td>
<td>305</td>
<td>3086</td>
<td>4.1</td>
</tr>
<tr>
<td>Ankle</td>
<td>2.0</td>
<td>5</td>
<td>378</td>
<td>68.3</td>
<td>18.1</td>
<td>8.4</td>
<td>40</td>
<td>17.5</td>
<td>538</td>
<td>3523</td>
<td>4.1</td>
</tr>
<tr>
<td>Knee</td>
<td>1.25</td>
<td>5</td>
<td>378</td>
<td>23.7</td>
<td>36</td>
<td>16.2</td>
<td>25</td>
<td>10.8</td>
<td>86</td>
<td>1211</td>
<td>4.1</td>
</tr>
<tr>
<td>Knee</td>
<td>2.0</td>
<td>5</td>
<td>378</td>
<td>34.6</td>
<td>47.5</td>
<td>19.8</td>
<td>32.3</td>
<td>17</td>
<td>309</td>
<td>2608</td>
<td>4.1</td>
</tr>
<tr>
<td>Knee</td>
<td>1.25</td>
<td>10</td>
<td>378</td>
<td>8.1</td>
<td>17.9</td>
<td>8.1</td>
<td>27.2</td>
<td>10.9</td>
<td>86</td>
<td>1211</td>
<td>8.2</td>
</tr>
<tr>
<td>Knee</td>
<td>2.0</td>
<td>10</td>
<td>378</td>
<td>13.8</td>
<td>23.6</td>
<td>9.9</td>
<td>46.9</td>
<td>18.1</td>
<td>309</td>
<td>2608</td>
<td>8.2</td>
</tr>
</tbody>
</table>

map of Section 2.3.4 shows a region of compromise, Fig. 3-6. There is a region where both knee and ankle can operate, and experience reasonable energy cost. For a commercial product a more optimized configuration would be preferred, but for a laboratory prototype this operational zone could be appropriate to reduce hardware complexity and maintenance requirements. Performance can be further improved by adjusting spring stiffness to the joint: a stiffer spring for the ankle and a softer spring for knee. The ability to use different lead screws as examined in Table 3.2 are clearly shown in this figure, showing a means to further improve performance.

Varying the spring stiffness based on user mass (or applied torque) can have a positive overall affect on eCOT. A near linear relationship between mass, and stiffness is apparent in Fig. 3-9, where the series stiffness and user mass design space is explored for 1.25m/s walking. The ability to tune the spring stiffness to match a user enables a system to be maximally efficient for a wide variety of users or applications. For example, the optimal spring stiffness in Fig. 3-9 is not the same as for that of the knee in Fig. 3-10. Being able to further tune the actuator by changing gear ratio would even further improve energetics as Fig. 3-10(b) shows a marked improvement in energetics for walking when the gear ratio and spring are tuned appropriately. The spring stiffness and reduction ratio dependency of these plots illustrate a driving factor in the design of the TF8: the desire to have the ability to exchange both spring and screw.
Figure 3-6: Overlaying the knee level-ground energetic map with that from the ankle there is a compromise region where both systems can operate with the same hardware configuration. The dotted lines show the as built overall gear ratio and expected COT performance. Initial spring estimates were for a 278kN/m spring, shown here. This is evaluated with a 90kg person walking at 2m/s.
Figure 3-7: (a) Ankle electric performance estimates for the 90kg subject walking at 1.25m/s. (b) Ankle electric performance estimates for the 90kg subject walking at 2.0m/s.

Figure 3-8: (a) Knee electric performance estimates for the 90kg subject walking at 1.25m/s. (b) Knee electric performance estimates for the 90kg subject walking at 2.0m/s.
Figure 3-9: COT as related to user mass and series spring stiffness walking at 1.25m/s. (a) The ankle actuator, as built with lead=5mm screw. (b) The ankle actuator, with the optimal lead=4mm screw.

Figure 3-10: Knee COT as related to user mass and series spring stiffness walking at 1.25m/s. (a) Results with the as built system with a screw lead=5mm. (b) The energetics of the actuator assembled with an optimal lead of l=10mm.
3.2.3 Motor

An initial guess at potential motor candidates can be estimated by inspection of static parameters. To compare motor capabilities the motor torque generating efficiency must be evaluated with relation to the effect of rotor inertia and the expected reduction ratio. The motor constant:

\[ K_m = \frac{K_t}{\sqrt{R}} \]  

(3.1)

with units [Nm/√W], can be used to compare motor torque generating efficiency, where, \( R \) is motor winding resistance, and \( K_t \) is torque constant with units [Nm/A]. This measure misses the effects of rotor inertia. To account for this Sensinger defined the \textit{Speed Rate} as a benchmark to use in evaluating motor dynamic performance by normalizing the motor constant with rotor inertia [91]. However, this term does not consider the effect of drivetrain on reflected inertia, a necessary consideration in overall system performance.

To improve comparison across motors the speed-rate should be normalized by the application-specific reflected inertia. That is, for a given motor and load combination there is a required transmission ratio that affects the reflected inertia of the motor rotor. A small rotor motor may have low inertia, but requires a large gear ratio to achieve the specified output torque. This gearing reflects the inertia as viewed by output by a power of two. To account for this gearing effect and provide a more meaningful output comparison a new term is presented as the \textit{Reflected Speed Rate} (RSR):

\[ RSR = \frac{K_m}{J_m N_{max}^2} \]  

(3.2)

where, units are [Nm/√W·Kg·m²], and

\[ N_{max} = \frac{\max(|\tau_l|)}{\tau_o} \]  

(3.3)

is the maximum gear reduction required to achieve the maximum target load torque \( \tau_l \), and \( \tau_o \) is motor stall torque. This relation provides a more effective means to compare motors with their required reduction ratios.

Motor manufacturers of hobby-grade outrunner motors often provide limited, if any, datasheets for their products. The basic parameters frequently provided \( K_v, R, \)
$V_i$ can be used to estimate the parameters necessary to simulate motor behavior:

\[ K_t = \frac{60}{K_v2\pi} \]  
(3.4)

\[ \tau_o = \frac{K_I V_i}{R} \]  
(3.5)

\[ \omega_o = \frac{V_i}{K_t} \]  
(3.6)

\[ J_m = \frac{m_m}{2} r^2 \]  
(3.7)

\[ B_m = \frac{K_I I_n}{\omega_o} \]  
(3.8)

where, $\tau_o$ is stall torque, $\omega_o$ is no-load angular velocity. The motor mass $m_m$, and rotor radius $r$, are used to estimate rotor inertia $J_m$ when not given by the manufacturer, $I_n$ is no-load current, and $B_m$ is internal damping. This inertia estimate while not strictly accurate, comes within roughly 15% of inertial parameters found from 3D models built from disassembled motors. Similarly, the damping term is not accurate but gives a rough estimate that aligns well with other published values.

The optimization searched through 40 different motors that included traditional internal rotor motors and drone-type outer rotor motors. The chosen motor is a frameless T-Motor U10 Plus KV100 outrunner motor [73] capable of producing instantaneous peak torque of 4Nm and 2400rpm at 24V DC bus voltage, and up to 4Nm and 4400rpm at 44V. The RSR of this chosen motor is 221 compared to the RSR=5.4 for a 200W EC-4 Pole Maxon motor. This high dynamic torque efficiency is due in part to the low winding resistance and high torque constant of the motor. Larger motors do tend to perform better as they are more efficient at generating torque than smaller motors, the U12 from T-Motor has a 107mm rotor diameter and an RSR of of 950. However, wearable robots such as these powered prostheses are mass and size limited and the U10 is about the largest acceptable size motor.

A secondary reason to select a large diameter, high torque motor is beneficial acoustics. Lower required reduction ratios means slower angular velocities at the motor and so lower frequency audible noise from the drive-train. The perceived loudness of sound is nonlinear and higher frequency noises are experienced as louder by humans, particularly near the 2000-5000Hz range [92]. The emPower likely has a gear ratio somewhere around 200-300:1 resulting in motor velocities up to 16000rpm\(^2\). Estimating the drive pulley on the motor shaft to have 20 teeth, this gives a potential

\(^2\)Estimated from 6rad/s desired ankle velocity, and maximum motor velocity of 16,700rpm from the EC-4Pole 200W Maxon motor
audible noise generated at about 5000Hz, well within the most sensitive portion of human hearing. With the low reduction ratio of the TF8 design there are motor velocity reaching up to 4400rpm with a theoretical audible noise of 50hz. This is not quite accurate due the rolling elements in the ballnut and bearing assemblies rotating generating noise, but it is certainly lower frequency and is perceived to be lower noise level than higher ratio, higher frequency drivetrains.

Figure 3-11: Acoustic weighting of perceived loudness for humans, the blue line is the dBA weighted scale commonly used in the US. Borrowed from [93].

Ability of the motor to achieve desired load trajectories is evaluated against its operational capacity. Figs. 3-12 through 3-15 show how load requirements are evaluated against motor capacity. The dotted red lines are the nominal operating limits of the motor, the solid lines represent the maximum peak operating capability of the motor, and the solid blue line is the required trajectory of the motor as it tracks output load through the drivetrain. In these plots: the top plot is the torque speed capability, the bottom plot is the available power delivery from the motor, (a) is the performance of the motor as built with the nominal configuration of a 5mm lead ballscrew, (b) is the performance of the motor when combined with its optimal ballscrew as defined in Table 3.2. Figs. 3-12 and 3-13 are ankle loads at 1.25m/s and 2.0m/s, respectively. While Figs. 3-14 and 3-15 are knee loads at 1.25m/s and 2.0m/s, respectively.
At nominal walking conditions of 1.25 m/s the as-built TF8 with 5mm lead ballscrew is able to achieve all required torque, velocity and power requirements for both ankle and knee trajectories. At fast, nearly jogging 2m/s walking speeds the ankle torque may saturate at one data point, and the knee velocity may saturate at three target points. These trajectories are interpolated to 100 data points, meaning 97% of the target velocities are achievable with the nominal actuator configuration. These results show that while adjusting the gear ratio to more closely match the ankle and knee would better utilize the motor, and improve efficiency, it is not strictly required to achieve a wide range of operating conditions with a single actuator.
Figure 3-13: These plots are the ankle motor trajectories and limits of a 90 kg person walking at 2.0 m/s with different reduction ratios. (a) Using a screw lead of 0.005 mm for a nominal gear reduction of N=52.8 (b) Using a screw lead of 0.004 mm for a nominal gear reduction of N=66.
Figure 3-14: These plots are the knee motor trajectories and limits of a 90 kg person walking at 1.25 m/s with as designed and optimal gear ratio. (a) Using a screw lead of 0.005 mm for a nominal gear reduction of N=52.8 (b) Using a screw lead of 0.010 mm for a nominal gear reduction of N=23.

Figure 3-15: These plots are the knee motor trajectories and limits of a 90 kg person walking at 2.0 m/s. (a) Using a screw lead of 0.005 mm for a nominal gear reduction of N=52.8 (b) Using a screw lead of 0.010 mm for a nominal gear reduction of N=23.
3.2.4 Series Elastic Elements

The series elastic actuator (SEA) [58] has been the core actuator technology for the design of powered prostheses due to its energetically favorable features of a contractile element in series with an elastic element. This configuration of spring is a biomimetic representation of the biological muscle and tendon unit [23]. The series elastic element serves three main purposes in this design: to store and release mechanical energy, to decouple the inertia of the drivetrain from the output, and to reduce impact loads on the drivetrain. For example, when walking transitions from middle to late-stance (Figs. 1-1 and 3-39 shows walking gait phase designations) the body center of mass overhangs the foot center of pressure, stretching the Achilles tendon as it combines with the gastrocnemius and soleus muscle tension to provide supporting torque about the ankle. As late-stance transitions to push-off, the energy stored in the stretch of tendons is rapidly released along with contraction of the serially connected muscles. This rapid release of elastic energy results in a power output greater than the muscles can provide on their own [22], [23]. By leveraging similar principles the elastic element in series with an actuator can both store energy and then release supplemental mechanical power along with the electric power from the motor [59], allowing the motor to operate closer to its prime operating parameters, given the motor can move fast enough to get out of its own way (limited by torque and inertia).

The cyclical motion of gait is well suited to leverage the energy storage and power delivery capacity of springs [59]. The goal of the series elastic element design in this application is to maximize energy storage in the spring while also maximizing system control bandwidth – two opposing objectives. Increasing spring stiffness increases the sprung mass natural frequency, but also reduces the amount of energy that can be stored for a given loading condition. High stiffness also puts greater precision requirements on encoder based force sensing, something that can increase system reliability/safety. Reduced spring stiffness improves energy storage but also requires the motor to travel greater distance to reverse direction at high load and reduces the natural frequency of the sprung drivetrain components. Generally, it is preferred to keep the system natural frequency outside of the operating regime of the actuator because small commands can drive the system unstable.

Compressing a spring requires work, a force applied over a distance. This work deforms a volume of material by twisting, bending or pushing to change its shape. This deformation of a volume of material is strain, and the energy embodied in strain...
increases until material structural bonds begin to fail. The amount of stress a material can withstand over a given amount of deformation is directly related to the micro-structure of the material. In the case of composites it is also related to the fiber layup orientation.

A definitive authority on spring design, the 1944 text Mechanical Springs by J.M. Wahl [94] is an exhaustive study of all things spring related. Wahl compares various spring geometries with an axially loaded beam as the normalization factor. Wahl finds that when reasonable design consideration is taken and particularly when considering the imprecision of fatigue analysis, springs designed appropriately will all perform within a reasonable margin of each other. He thus suggests packaging concerns frequently account for the spring selection choice more so than any other numerical geometric efficiency comparison. In the case of wearable and mobile robots mass and size are major packaging concerns.

### Strain Energy

Spring design seeks to find materials that maximize strain and stress while also fitting geometric requirements. To fully utilize material volume the goal is to make an equal distribution of strain throughout the entire volume of material while it is stressed just to the yield limit. What this means is the ideal spring material has low elastic modulus and high yield strength. The average strain energy density for uniaxial deformation, in units $[\frac{J}{kg}]$, can be used to compare candidate materials [94]:

\[
\Delta U = \frac{1}{2} \sigma \epsilon \Delta V = \frac{1}{2} \frac{\sigma^2}{E} \Delta V \tag{3.9}
\]

\[
\frac{\Delta U}{\Delta m} = \frac{1}{2} \frac{\sigma^2}{E \rho} \tag{3.10}
\]

where, $\epsilon$ is strain, $\sigma$ is stress, $E$ is Young’s Modulus, $\Delta V$ is change in volume.

A selection of potential spring materials are shown in Table 3.3. Polymers such as nylon or polyurethanes perform well, though their internal viscoelastic damping results in noticeable hysteresis and their strain-rate dependent modulus of elasticity is not favorable for the this analysis[96], though could potentially be exploited with more sophisticated non-linear analysis. Non-isotropic materials such as composites can be tuned to maximize mass utilization when their high tensile-strength fibers

\[^3\text{It is sometimes beneficial to take the spring slightly past the elastic yield, causing some plastic yielding! This process called presetting work hardens the material and increases the yield stress slightly.}\]
Table 3.3: Strain energy density of materials

<table>
<thead>
<tr>
<th>Material</th>
<th>E</th>
<th>G</th>
<th>$\sigma_f$</th>
<th>$\rho$</th>
<th>$\Delta U$</th>
<th>$\Delta m$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[GPa]</td>
<td>[GPa]</td>
<td>[MPa]</td>
<td>[g/cm$^3$]</td>
<td>[J/kg]</td>
<td></td>
</tr>
<tr>
<td>GC-70-UCL$^2$</td>
<td>137</td>
<td>52.7</td>
<td>836</td>
<td>1.55</td>
<td>1646</td>
<td></td>
</tr>
<tr>
<td>GC-67-UCB$^2$</td>
<td>40</td>
<td>15.4</td>
<td>381</td>
<td>1.88</td>
<td>961</td>
<td></td>
</tr>
<tr>
<td>Maraging Steel</td>
<td>200</td>
<td>76.9</td>
<td>1208</td>
<td>8.08</td>
<td>451</td>
<td></td>
</tr>
<tr>
<td>Ti-6Al-4V</td>
<td>114</td>
<td>42.2</td>
<td>585</td>
<td>4.43</td>
<td>339</td>
<td></td>
</tr>
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<td>1.4</td>
<td>32</td>
<td>1.1</td>
<td>332</td>
<td></td>
</tr>
<tr>
<td>17-7 Ph CH900</td>
<td>204</td>
<td>78.5</td>
<td>660</td>
<td>7.8</td>
<td>137</td>
<td></td>
</tr>
<tr>
<td>7075-T6</td>
<td>72</td>
<td>27</td>
<td>172</td>
<td>2.8</td>
<td>73</td>
<td></td>
</tr>
</tbody>
</table>

$^1$ Allowable fatigue endurance limit is assumed 0.5$\sigma_u$ for steel, 0.4$\sigma_u$ for composites and plastics, and 0.3$\sigma_u$ for aluminum alloys.

$^2$ Unidirectional composite fibers aligned in 0° [95].

are aligned with the loading condition. Though, composites also express hysteresis due to internal damping in the matrix support material, the superior strain energy density of composites clearly make them the winner in designs that can exploit fiber orientation.

Three types of loading conditions can induce stress in a material: axial, bending, shear.

$$\sigma_{axial} = \frac{P}{A}$$  \hspace{1cm} (3.11)

$$\sigma_{bending} = \frac{My}{I}$$  \hspace{1cm} (3.12)

$$\tau_{shear} = \frac{Tc}{J}$$  \hspace{1cm} (3.13)

where, $P$ is axial force, $A$ is area, $M$ is moment, $T$ is torsion, $y$ is distance from the neutral axis, and $c$ is distance from neutral torsional axis, $I$ is second area moment of inertia, $J$ is polar area moment of inertia. Examining the above equations and applying Castigliano’s Theorem the displacement and stiffness factors can be determined for each of the main spring types: simple axial tension, cantilever beam, torsion spring, spiral spring, round bar, and helical. Looking at the uniaxial case, deformation can be found from the following:

$$U_{axial} = \int \frac{1}{2} \sigma \varepsilon \Delta V = \frac{P^2 L}{2AE}$$  \hspace{1cm} (3.14)

$$\delta_{axial} = \frac{\partial U}{\partial P_i} = \frac{PL}{AE}$$  \hspace{1cm} (3.15)
The energy and subsequent deformation can also be found for pure bending and shear (torsion):

\[
U_{\text{bending}} = \frac{M^2 L}{2EI}, \quad \theta_{\text{bending}} = \frac{ML}{EI} \quad (3.16)
\]
\[
U_{\text{shear}} = \frac{T^2 L}{2GJ}, \quad \theta_{\text{shear}} = \frac{TL}{GJ} \quad (3.17)
\]

where, \( G \) is shear modulus. This approximation is however only valid for small displacement. For moderately large deflection a nonlinear strain term is required to include beam element rotations \([97]\). This additional beam theory was neglected for this analysis though it would provide greater fidelity of analysis as would including a damping term.

Figure 3-16: Finite element analysis (FEA) section view of a machined rectangular helical compression spring. The section view shows the stress (strain) distribution is highest at the surface as well as inner diameter of the coils. A hollow spring coil spring would be more ideal in this case, requiring an additive manufacturing process and non-optimal spring materials.

Maximum material utilization for strain deformation occurs when any of the Eqtns. 3.14-3.17 are fully utilized. Bending is an ideal way to store strain energy. Traditional coil springs induce bending generated by moment loading, however they are subject to stress concentrations on their internal radii. This stress condition is visible in the finite element analysis (FEA) of a machined rectangular cross-section spring in Fig. 3-16. As can be seen in the figure, a coil spring would be best fabricated with hollow coils to reduce mass. Constructing a spring of this nature likely
requires an additive manufacturing process which limits use of optimal strain-energy
density materials. A number of other spring types are possible. The belleville washer
or flat-disc stacks well for arrangements similar to the coil spring, and they can be
fabricated from composite materials. However, belleville washers are also susceptible
to high stress concentrations at the inner radius, as seen in Fig. 3-17. A flat plate in
bending is the most efficient way to store bending energy due to the constant moment
applied along its full length. This loading condition has limited stress concentration
compared to the other designs.

![FEA of a flat-disc spring, similar to a bellevile washer but without the
pre-loaded dished shape bias. Stress is concentrated at the inner radius and is not
evenly distributed throughout the material.](image)

To understand the strain conditions it is helpful to examine the shear and moment
diagrams of a point load and moment-coupled cantilever beam. A point loaded can-
tilever beam has internal shear strain at the point load that reverses direction at the
supporting reaction force at the fixed end, shown in Fig. 3-18. The beam experiences
a moment couple at its fixed base but there is no moment support at the point of
applied load. This generates a strain energy of:

\[
U_{\text{bend}} = \frac{M_f^2 L}{6EI}
\]  

(3.18)

In contrast, a cantilever beam in pure bending has a constant moment across its
entire length that matches Equation (3.16). This is the loading configuration of
the spring implemented in TF8 as shown schematically in Fig. 3-20 (a), and its
accompanying shear and moment diagrams in Fig. 3-20 (b). This configuration
enables the maximum strain loading condition which is three times the energy storage
of a simple point loaded cantilever beam.

There is one other configuration that could enable a more controlled motion of the
spring end, and that is to support the spring with a pivot and rolling pivot in a tra-
Figure 3-18: (a) The equivalent loading of a point load, $F_s$ applied to the end of a cantilever beam. (b) A shear force is felt across the beam due to the reaction force, $F_f$ at the fixed end. Similarly, at the fixed end a reaction moment $M_f$ supports the moment generated by the point load. The moment decreases linearly along the length of the beam to the point of the applied load, where no moment is supported by the beam.

Additional simply-supported cantilever beam configuration. Figure 3-19 demonstrates this configuration in (a) schematic form, and (b) shear moment examination. This rather complicated configuration again enables a moment application to the beam, but only on one end. Thus the loading conditions match the point-load but with an inverted arrangement.

These shear-moment diagrams are verified with finite element analyses (FEA) of point and bending loads. In Fig. 3-21 (a) an even distribution of stress and strain can be seen along the full length of the spring. At the neutral axis, as is expected, there is low strain. A cantilever beam with a point load cannot support a moment and so has stress concentration near the fixed base, Fig. 3-21 (b).

To maximize material utilization the series elastic element in TF8 is packaged as a moment-coupled cantilever beam so that it primarily undergoes moment bending. The reaction force of the motor and screw is applied by way of a moment arm clamped to the beam, generating a force-couple. The fixed end of the spring is similarly clamped to the frame of the actuator. The energy stored in the spring is dominated by bending and axial strains, though a full accounting also includes shear strains due

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4Building a foam-core composite spring would leverage this feature and would be an opportunity for further mass savings and hence improvement to sprung mass.
Figure 3-19: (a) The equivalent loading of a moment coupled to a simply supported cantilever beam. (b) The shear and moment diagrams are inverted from 3-18, but are equivalent. A vertical shear force is felt across the beam due to the reaction force, $F_f$ at the rolling end. The horizontal shear force is directly coupled to the pinned support. The applied moment decreases linearly along the length of beam until it is supported by a reaction force at the rolling end at the left.

Figure 3-20: This is the spring configuration used in TF8. (a) The equivalent loading on the spring is effectively a moment applied to the end of a cantilever beam. The screw force $F_s$ is nominally aligned with the cantilever beam and offset by distance $t_f$ shown in Fig. 3-26. This moment arm applies a force couple to the beam through its bolted interface, effectively creating a pure moment at the end of the beam. (b) The shear and moment diagrams show the beam has a constant moment across the entire length of the beam – this is ideal for maximum utilization of material under strain.
Figure 3-21: (a) Finite element analysis of a 3kN applied at 10 degrees and 2.5Nm rotational torque (motor reaction force) both about the shown axis. This is a downward facing load with a total displacement of 14.6mm. (b) FEA of a cantilever beam with a point load applied. This is an examination of a multiple leaf springs arrangement to increase stiffness. Here a 6mm displacement is enforced on two sandwiched 4mm sheets with beam length=70mm, and width = 60mm, of unidirectional carbon fiber. Interleaf friction is set at $\mu = 0.05$. The resultant force is 4kN.

to screw askew orientation under heavy loading:

$$U_{bend} = \frac{L(P_x t + LP_y)^2}{2EI}$$  \hspace{0.5cm} (3.19)

$$U_{axial} = \frac{P_x^2 L}{2AE}$$  \hspace{0.5cm} (3.20)

$$U_{shear} = \frac{P_y^2 h}{2AG}$$  \hspace{0.5cm} (3.21)

$$U_{total} = U_{bend} + U_{axial} + U_{shear}$$  \hspace{0.5cm} (3.22)

where, $P_x, P_y$ are the screw force $F_s$, contributions defined as: $P_x = F_s \cos \theta_p$, $P_y = F_s \sin \theta_p$, and $\theta_p = \frac{F_s}{K_x t}$.

$$P_x = F_s \cos \theta_p$$  \hspace{0.5cm} (3.23)

$$P_y = F_s \sin \theta_p$$  \hspace{0.5cm} (3.24)

where,

$$\theta_p = \frac{F_s}{K_x t}$$  \hspace{0.5cm} (3.25)

$K_x$ is linear spring stiffness, and $t$ is the moment arm length as described in Fig. 3-26 and the component specifications.

Spring geometries $L, h, \text{ and } b$ are numerically solved by specifying the desired
stiffness $K_s$, and stored energy (3.22) defined by the component specifications in Section 3.2.2. Searching for minimum mass configurations that satisfy the allowable design constraints for beam height, width, and length leads to an optimal spring design. The beam height is constructed from:

$$h = \sqrt[3]{6 \left( P_x t + P_y L \right) / b \cdot \sigma_f} \quad (3.26)$$

where, $b$ and $L$ are beam width and length, respectively, and $\sigma_f$ is the allowable fatigue limit of the material.

**Spring Specification**

The overall component specification defined a nominal series stiffness of $K_s = 271 \, kN/m$, capable of storing 9J of mechanical energy. To achieve this the spring optimization resulted in a 8.6mm thick spring. However, because the overall structural stiffness was designed to match that of bone (Section 3.2.5), it reduced the overall effective series stiffness of the structure – something that had not originally been accounted for in the analysis. To account for this reduced series stiffness the configuration as-built instead includes a 42mm x 12,4mm thickness beam with a sprung length of 86mm. The spring is a unidirectional E-glass fiberglass composite (GC-67-UB) manufactured by Gordon Composites [95].

The overall actuator series stiffness is measured by locking the actuator output and measuring the motor displacement along the screw when high loads are commanded. The results of this experiment are shown in Fig. 3-24, where the actuator is taken to 175Nm of torque or 4500N force on the screw. The measured linear stiffness from these experiments define the actuator overall series stiffness to be 378kN/m. The maximum deflection is limited to 13mm linear translation by mechanical features built into the structure; at 13mm of deflection the spring experiences nearly 265Nm of torque and stores 19 Joules of mechanical energy.

Estimates of spring deflection, and linear travel of the screw, while tracking the torque and motion trajectories of the 90kg user walking at 2m/s are shown in Figs. 3-22 and 3-23. This data is used to verify the energy storage in the spring in addition to the estimated motion of the motor itself. An added benefit of this analysis is the linear displacement of the screw is also used for estimating lifetime of the ballscrew assembly with the root mean square (RMS) and maximum screw forces as well as cycle life of the bearings. This analysis allows a relatively accurate assessment of the
screw and bearing life.

An additional benefit of the cantilever beam configuration is its capacity to constrain the rotation of the motor, negating the need for additional guide bearings for the SEA arrangement. Traditional SEA designs require linear guideways to support the spring bodies. In the case of this design the spring is its own flexural spring. To verify torsional loads from the motor reaction force are not a problem, the efficiency of the ballscrew is considered. The motor may generate as high as 4Nm of torque as it transmits torque into translation of the screw. This deflection is estimated with FEA analyses of Fig. 3-21 and is found to be inconsequential compared to the up to 400Nm that can be expressed on the spring under maximum load (this is due the moment-arm coupling screw force to the spring being nearly twice the length as the rotary arm at the output of the actuator).

![Series Spring Displacement and Screw Force vs Gait Cycle](image)

Figure 3-22: Ankle estimated spring deflection and force in the screw during a 90kg person walking at 2m/s gait cycle.

**Spring Characterization**

Total structural loop stiffness must be characterized to measure joint torque by way of spring deflection. The angular stiffness of the cantilever beam is combined with
Figure 3-23: Knee estimated spring deflection and force in the screw force a 90kg person walking at 2m/s gait cycle.

The elasticity of the overall actuator stiffness to determine the linear stiffness of the effective series spring. In order to evaluate the spring stiffness the finished actuator is bench tested in a locked output state and commanded to apply the maximum load rating of 175Nm. The axial force in the screw is plotted in Fig. 3-36 and compared to calculated displacement of the moment arm. To characterize the behavior the spring the Euler-Bernoulli (E-B) bending beam approximation of a beam with a pure moment can be used to define the angular stiffness as:

$$K_\theta = \frac{EI}{L} \eta_k, \quad (3.27)$$

where, $\eta_k$ is a fitting factor to accommodate additional elasticity in the structure not captured by the beam equation. The small angle approximation applies to the expected deflections in this system, allowing simplification of the force measurement based on the linear displacement of the screw, and effective linear stiffness of the actuator, $K_x$:

$$F_s = \Delta K_x, \quad (3.28)$$
Figure 3-24: A load cell in-line with the ball-screw measured real force in the screw. The maximum screw force of 4500N is equivalent to maximum rated load of the actuator of 175N. Results shown here are with the 12, 4mm thick, 378kN/m fiberglass spring. A transfer function estimate of the axial force to spring linear displacement and a linear model of an Euler-Bernoulli bending beam track to experimental data.

\[
\Delta = X_m - X_l(\theta) = (\theta_m - \theta_{m0}) - (c(\theta_l) - c(\theta_{l0}))
\]

\[
K_x = \frac{K_{\theta}}{t^2},
\]

Here, \( \Delta \) is the displacement of the screw measured in the difference between motor position \( X_m \) and expected motor position due to output joint orientation \( X_l(\theta) \), \( \theta_{m0} \) is initial motor position at startup, \( c(\theta_{l0}) \) is initial screw length at startup.

The damped hysteresis behavior of the composite spring can be estimated by a single pole spring-damper transfer function. In Fig 3-24 comparison of differing pole/pair combinations and an Euler-Bernoulli beam are compared to the measured data. However, the results show variation of up to 5% so the Euler-Bernoulli (3.30) was fit and used for encoder-based force sensing. The estimated transfer function is below:

\[
H_{spr}(s) = \frac{3.433e07}{s + 98.85}
\]
In practice encoder based force sensing is highly dependent on the precision of system geometry and backlash. To calibrate the force sensing joint angle and motor angle measurements are collected with the actuator at various locations in its range of motion. This is then used to numerically solve for corrected linkage geometry by minimized the error between measured and calculated forces. Real screw force is measured with an in-line 4.5kN load cell (Futek LCM300 [98]) in order to verify the encoder based force measurements using Eq. (3.28).

3.2.5 Structure

To determine a target maximum deflection for the structural components that ground the actuator frame to a socket a biological equivalent is used: the deflection of tibial and femoral bones under equivalent walking conditions. For geometrical parameters, the level-1 cross-sectional thickness and second area moment of inertia of tibial bone as specified by Milgrom et al. [99] is used. In-vivo experiments show deflection angles of 0.5-1 degrees anterior-posterior, 1.5° torsion and 0.5° medial-lateral during normal walking [100]. With a modulus of elasticity of about 6 GPa [101], a value close to nylon, the tibia has a measured stiffness in the anterior-posterior direction of 6 Nm/mm, 3.5 Nm/mm in the medial-lateral direction [102]. While the Young’s Modulus of Elasticity of femoral cortical bone measures as 17 GPa longitudinal, and 11.5 GPa transverse by [103]. Estimated moment loads on the tibia are given in a bodyweight metric with a maximum load of 75 BWmm, that is 75 times body weight times millimeter [104].

\[
\delta = \frac{0.075BW \cdot m \cdot g \cdot l}{2 \cdot E \cdot I} \quad (3.32)
\]

\[
\delta = 3.37 mm \quad (3.33)
\]

The design architecture of the TF8 as a RFSEA means the motor moves with the spring. The volume through which the motion of the motor tracks must be kept clear of structural material. The estimates of spring deflection from Figs. 3-22 and 3-23 are used to estimate the deflection of the motor mounted on the moment-arm (described in more detail in Section 3.2.7). This estimate of beam bending is used to trace this motor motion path and is the reason for the curved features in the frame.

\[\text{Actual deformation is affected not only by gravitational loading but also muscle tendon tension and the geometry of the insertion points [100]. However these additional loading conditions were left out of this analysis.}\]
Figure 3-25: Borrowed from [99]. Illustration of bone segment second area moment of inertia estimates.

of the actuator as seen in Fig. 3-1.
3.2.6 Drivetrain and Linkage Geometry

Joint torque is generated by a linear actuator that acts on a moment arm about the joint axis, the geometry of which is optimized to match the power stroke of the gait cycle. The linear actuator is composed of a ballscrew integrated into the motor rotor. The rotation of the screw is limited by a push-rod end affixed to one end that pivots around an orthogonal axis located a projected distance $r$ from the joint axis. The optimization uses a gradient descent search to find the linkage geometry to match the power-stroke of an ankle gait cycle by setting the search objective to minimize eCOT.

The linkage parameters searched for are defined in Fig. 3-26 (a) and are defined respectively as: $f$ offset from femur, $f_k$ offset from knee along femur, $t$ offset from tibia, and $t_k$ is the offset from the knee along tibia. The labeling of these parameters relates to a knee joint because this design was initially focused toward the design of a knee actuator. The parameters used to determine the instantaneous moment arm length are defined in Fig. 3-26(b) and by applying the law of cosines the screw length and moment arm can be defined as:

$$a = \sqrt{f^2 + f_k^2}$$

$$b = \sqrt{t^2 + t_k^2}$$

$$T_\theta = \arctan \left( \frac{t}{t_k} \right)$$

$$F_\theta = \arctan \left( \frac{f}{f_k} \right)$$

$$C_\theta = \pi - \theta_t - (T + F)$$

$$c = \sqrt{a^2 + b^2 - 2 \cdot a \cdot b \cdot \cos(C_\theta)}$$

$$A_\theta = \arccos \left( \frac{a^2 - (c^2 + b^2)}{-2bc} \right)$$

$$r = b \cdot \sqrt{1 - \left( \frac{a^2 - (c^2 + b^2)}{-2bc} \right)}$$

where, $r$ is the moment arm or projected distance from the actuator pivot that the screw force applies along the length $c$ which measures the distance between the motor pivot and rotary output arm pivot, and $C_\theta$ is used for construction of $c$. The geometry $a$ and $b$ are used for convenience in the calculations above.

The results of optimizing the linkage geometry match the high gear ratio to the
Figure 3-26: (a) The optimization procedure searched for the $t, t_k, f, f_k, L$ geometry parameters. The objective was finding moment arm geometry with the most efficient configuration of the power stroke as specified by the load trajectory and actuator dynamics. (b) Parameters used for calculating instantaneous moment-arm length for control.

high torque regions of the load trajectory as well as the high velocity motions with the low gear ratio configuration. Fig. 3-27 illustrates the effective gear ratio throughout the range of ankle motion during walking as does Fig. 3-28 show the knee orientation. The overall gear ratio is nominally $52 : 1 \pm 3.5$ during level ground walking but this varies throughout the gait cycle. At high flexion the gear ratio can sweep as low as $2 : 1$ as the actuator approaches limits to its controlled range of motion. In the actual hardware a different rotary output arm is used on the knee to provide a hard-stop at zero degrees while shifting the zero point 30 degrees to provide full 110 degree motion of the joint. This motion is achieved with a Thomson Linear 5mm lead ball-screw (BSPRM012L05M) and nut (KGM-N-1205-RH) mounted directly inside the motor rotor, and located a projected perpendicular distance nominally 41 mm from the joint axis.
Figure 3-27: The moment-arm changes throughout the range of motion of the output. The top plot shows a singularity that is never reached due to the joint being set to -35 to 75 degrees maximum ranges. The middle and bottom plots illustrate how the design optimization places the maximum moment-arm distance at the required power stroke of stance phase of the ankle as well as allocating lower effective gear ratio within the range of higher velocity.

The motor is a frameless U10 Plus KV100 motor (fabricated by T-Motor) built with a custom rotor and stator to integrate the ball-nut directly into the rotor and to replace the motor bearings with thin-section angular contact bearings capable of supporting the high axial forces in the screw. Axial pre-load comes from a nut that also retains a magnetic motor encoder rotor disk. The motor bearing stack and associated load path can be seen in Fig. 3-30(c).

The ballscrew and motor are mounted in needle-bearing\textsuperscript{6} pivots at the motor

\textsuperscript{6}In two built actuators the needle-bearings are replaced with plain bearings. Oscillating motion can be hard on needle-bearings because of high hertz-contact stress is not shared evenly across bearings, but is instead taken by only a small portion of the rolling elements. The plain bearings handle the oscillating motion better and have not experienced pitting seen in the two actuators assembled with needle-bearings.
Figure 3-28: The moment-arm changes throughout the range of motion of the output. The top plot shows a singularity that is never reached due to the joint being set to -35 to 75 degrees maximum ranges. The middle and bottom plots illustrate how the design optimization places the maximum moment-arm distance at the required power stroke of knee as well as allocating lower effective gear ratio within the range of higher velocity.

3.2.7 Mechanical Architecture

The actuator architecture of TF8 could be called a moment-coupled cantilever-beam reaction-force series elastic actuator (RFSEA). The configuration shown in Figs. 3-1 and 3-29 show the actuator assembled in an ankle embodiment. TF8 applies a
torque to the joint by coupling a linear actuator to a moment arm a distance from the joint axis. The linear actuator is composed of an outrunner motor with integrated ballscrew to generate linear force. Reaction-force from this linear actuator induces a moment on the cantilever-beam spring by way of a moment arm clamped to the spring, creating a force-couple. The spring finally serially grounds the load to the frame of the actuator. Four bolts attach the spring enabling it to be swapped to match the actuator to its application, such as users of different mass and to match the dynamics of either an ankle or knee.

Figure 3-29: The TF8 Actuator is shown configured as an ankle powered prosthesis with the main components labeled. The actuator minimum build height is 171mm measured between standardized mounting plates, shown here with standard prosthesis components attached. All dimensions are in millimeters.

Configured as an ankle the output joint placement is designed to match the relative orientation of the BiOM powered ankle [56] when mounted on an Össur Vari-flex foot [105] – the unloaded height of the ankle is 67mm and lateral placement with respect to mounting holes in the Vari-flex is matched. Matching alignment allows a direct
Figure 3-30: (a) Spring deflection occurs when a torsional load (dotted arrow) is applied at the rotary output, reaction force from the motor/screw causes deflection of the spring. (b) Shows the load path from the applied load at the joint, through the linear actuator, the cantilever spring and to the structural frame. (c) Close-up of the ballscrew integrated into the motor and the associated load-path. The compression load-path is shown as the dotted line, while the tension load-path is shown as the solid line through the motor bearing stack. (c) The full range of motion of the ankle configured TF8 prosthesis. Motion is labeled as $\theta_d$ is dorsiflexion, and $\theta_p$ is plantarflexion.
kinematic comparison to the BiOM and emPower. The minimum working actuator configuration as shown in Fig. 3-30(a) has a mass of 1.36kg. The overall hardware mass as a knee, not including embedded systems or battery measures 1.6kg, and the breakdown of that mass distribution is shown in Table 3.5.

Table 3.4: Minimum Configuration Mass Distribution of the TF8 Actuator

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (g)</th>
<th>(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td>549</td>
<td>40</td>
</tr>
<tr>
<td>Structural Components</td>
<td>374</td>
<td>27</td>
</tr>
<tr>
<td>Spring</td>
<td>116</td>
<td>9</td>
</tr>
<tr>
<td>Ballscrew</td>
<td>100</td>
<td>7</td>
</tr>
<tr>
<td>Spring Clamping Hardware</td>
<td>95</td>
<td>7</td>
</tr>
<tr>
<td>Load Cell</td>
<td>54</td>
<td>4</td>
</tr>
<tr>
<td>Fasteners</td>
<td>48</td>
<td>4</td>
</tr>
<tr>
<td>Encoder Hardware</td>
<td>25</td>
<td>2</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>1362</strong></td>
<td><strong>100</strong></td>
</tr>
</tbody>
</table>

Table 3.5: Mass Distribution of the TF8 Knee

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (g)</th>
<th>(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Structural Components</td>
<td>556</td>
<td>35</td>
</tr>
<tr>
<td>Motor</td>
<td>549</td>
<td>34</td>
</tr>
<tr>
<td>Spring</td>
<td>116</td>
<td>7</td>
</tr>
<tr>
<td>Ballscrew</td>
<td>100</td>
<td>6</td>
</tr>
<tr>
<td>Spring Clamping Hardware</td>
<td>95</td>
<td>6</td>
</tr>
<tr>
<td>Load Cell</td>
<td>54</td>
<td>3</td>
</tr>
<tr>
<td>Fasteners</td>
<td>48</td>
<td>3</td>
</tr>
<tr>
<td>Pyramid Adapter</td>
<td>48</td>
<td>3</td>
</tr>
<tr>
<td>Encoder Hardware</td>
<td>25</td>
<td>2</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>1592</strong></td>
<td><strong>100</strong></td>
</tr>
</tbody>
</table>

The actuator configured as an ankle, as shown in Fig. 3-29 is 2.0kg, the mass breakdown of which is in Table 3.6. The foot mass is for a size 28 Vari-flex foot with a custom rubber insole attached. The electronics weigh 53g while the wiring and additional fixtures for managing the wiring include an additional mass of 52 g, these are accounted for in the mass tables. Not included in the mass tables is the battery as it is a component that can be adjusted for different applications. The battery used for walking experiments is composed of two 3S 11.1V 1.0Ah lithium polymer hobby-grade batteries connected in series for a nominal operating voltage of 22.2V; the battery combined mass is 180g and is usually located off-board, mounted to the
socket. The fully equipped actuator configured as shown in Figs. 3-1 and 3-29 with on-board electronics, flex-foot, adapters and battery together weigh 2.0kg.

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (g)</th>
<th>(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Structural Components</td>
<td>556</td>
<td>28</td>
</tr>
<tr>
<td>Motor</td>
<td>549</td>
<td>27</td>
</tr>
<tr>
<td>Foot (Vari-flex)</td>
<td>249</td>
<td>12</td>
</tr>
<tr>
<td>Spring</td>
<td>116</td>
<td>6</td>
</tr>
<tr>
<td>Electronics/Wiring</td>
<td>105</td>
<td>5</td>
</tr>
<tr>
<td>Ballscrew</td>
<td>100</td>
<td>5</td>
</tr>
<tr>
<td>Spring Clamping Hardware</td>
<td>95</td>
<td>5</td>
</tr>
<tr>
<td>Foot Adapter</td>
<td>62</td>
<td>3</td>
</tr>
<tr>
<td>Load Cell</td>
<td>54</td>
<td>3</td>
</tr>
<tr>
<td>Fasteners</td>
<td>48</td>
<td>2</td>
</tr>
<tr>
<td>Pyramid Adapter</td>
<td>48</td>
<td>2</td>
</tr>
<tr>
<td>Encoder Hardware</td>
<td>25</td>
<td>2</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>2008</strong></td>
<td><strong>100</strong></td>
</tr>
</tbody>
</table>

The system specification achieved for the TF8 is a peak torque of 175Nm, a RMS nominal torque rating of 85Nm with a total range of motion of 110 degrees and a velocity of 6.8rad/s at the joint (when operated at 48V). The hardware mass of the knee, 1.6kg, is 65% of the weight of an equivalent leg segment for the target 90kg user. System heights are shown in Figs. 3-29 and 3-31: the height of the actuator from pyramid adapter mount to rotary output mount is 171mm, from mounting point to rotary joint is 156mm, and the overall unloaded clearance height with a Vari-flex foot is 223mm. For the knee configuration the actuator can be as short as 171 mm. The mass for the knee is described in Table 3.5 and applies to the components within the dimensions measuring 0-194mm as specified in Fig. 3-31.

The torque ratings of the device are based on fatigue life calculations on every loaded component in the assembly. That is the ballscrew, motor bearings, support needle bearings in pivots, rotary output bearings at the joint, spring, spring moment-arm, spring clamping screws, structure, and even the M2 screws holding the bearing presser flanges on the motor bearing and ballnut, all have fatigue life calculations behind them. The loading conditions used to estimate this lifetime is a 90kg person walking 2,0m/s 5000 steps, everyday. The L10 fatigue life as defined in respective manufacturer design guides was used to estimate a lifetime of $10^6$ cycles. These estimates are considered fairly accurate considering full joint trajectories are used to make the estimates. That is information such as screw force and displacement...
Figure 3-31: The TF8 actuator can be configured as both a knee and an ankle actuator, without modification. The minimum build height of a powered knee and ankle system are shown here. This minimum height is due to the potential for screw collision if both joints were to be in their maximum flexed joint orientation. Biological knee and ankles are offset by roughly 15-20 mm. If these actuators were configured with such an offset it would be enough to accommodate passage of the screw and would enable a further height reduction of 50 mm. The mass shown in Table 3.5 refers to the actuator components measured from 0 - 194 mm in this figure.

from Fig. 3-22 provide full state information for estimates: millimeters of travel converts into rotations at load. Tradeoffs are made to reach reasonable size and mass components.

The ballnut mounted inside the motor will be the first thing to fail after lasting
only one year when this maximum load every step of every day specification is applied. The reality is most people do not run every step of everyday, and in fact are quite a bit less active than this. This reality means even at its current state the TF8 is likely overbuilt for its primary application as a laboratory research platform. Nonetheless, every component is analyzed.

The expected life of the motor bearings is four years. These are a thin-section angular contact bearing (KAA10AGOK) combination that provides ample thrust and handles higher velocity than a four-point contact bearing (better suited for lower speed, due to high contact stress). The bearings are arranged in a back-to-back duplex pair to provide sufficient bi-directional thrust load and moment supporting capacity. The increased load capacity is not doubled for axial loads, instead it is \( C_a = C_{a1} n^{0.7} \), where \( C_{a1} \) is the thrust load capacity of a single bearing, and \( n \) is the number of rows used. Lifetime of \( 10^6 \) cycles is given by:

\[
L_{10} = \frac{C}{P^{3.42}}
\]

where, \( C \) is the thrust rating of the bearing assembly, and \( P \) is the equivalent thrust load. This is slightly different from the catalog design guide that specifies to use the radial equivalent load. A 45 minute conversation with a bearing expert (RBC Bearings 9/18/2017) verifies this is the correct way to account for loading in a primarily axially loaded configuration. To calculate actual lifetime \( L_{10} \) estimate for millions of cycles is divided by the estimated number of revolutions per cycle, times the number of cycles:

\[
Days_{walking} = \frac{L_{10}}{rev_{step} \cdot steps_{day}}
\]

The results of expected peak loading of 3200N thrust and 140N radial are a 242 day lifetime, and at an average RMS load (from the same gait cycle) a 8,673 days of bearing life is expected. A higher fidelity analysis would actually include the full force cycle, likely binning force and duration to develop an effective equivalent load per gait cycle.

The structural components, frame and fasteners are all designed to "infinite" fatigue-limit lifetime. The screws are steel and this may apply. The frame is Al7075-T6 which though strong does have a finite life, as does the composite spring. For these components the literature seems to lead to a reasonable estimate of the fatigue limit to be about \( 0.30 \sigma_u \). Thus, the repeated torque rating of 175Nm is confidently stated as a repeatable peak load based on design guide calculations across every loaded
component.

The load path transmits joint torque as a force down the axis of the ballscrew, through the motor bearing stack, through the pivots to the moment arm. This moment arm then transfers the force primarily as a moment that is then grounded into the frame that supports the output joint axle. From the bolted interface at the frame, the load could be transmitted to any desired structure. For this prosthesis application, the load is transmitted through Aluminum 7075-T6 structural routing features to a standard lower extremity prosthesis pyramid interface such that it may be mounted to a socket. These elements were designed to allow a maximum transverse deflection of 3.4mm as specified by estimates of biological bone loading [99], [100] in addition to sustaining fatigue life estimates of a $0.3\sigma_u$ ultimate stress for aluminum.

Fig. 3-30(a,b) shows the load path through the structure and the load path through the motor bearing stack. Operation of the actuator under maximum spring deflection is also shown in Fig. 3-30(a). The dotted-line arrow is an applied joint torque, $\tau_l$ and the solid arrow is force from motor displacement along the length of the screw, $F_s$. The green dotted line in both Fig. 3-30(a) and (b) shows the load path when the structure experiences compression loads. The motor bearings were replaced with angular contact bearings in order to support the axial load of the screw. The load paths through the angular contact bearings differ when experiencing compression and tension loads due to their high contact angle. The orange solid line in Fig. 3-30(b) shows the circuitous tension load path from pivots, through the bearing stack inside the motor and through the ball-nut presser-flange before passing into the ball-nut and ball-screw. These loads are physically mirrored to both pivots and distributed circumferentially throughout the projected areas.

The reaction force from the torque applied to the ballscrew must also be accounted for in the holding force of the screw to secure it torsionally. The efficiency of a ballscrew under heavy load is related to the lead angle of the screw. For this 5mm lead screw the ankle is $5.7^\circ$ equating to a minimum efficiency of $\eta_{\text{screw}} = 0.9$ [106], though this is a load dependent characteristic, being lower and lower loads [107]. The reaction torque can then be calculated as:

$$\tau_{\text{react}} = \tau_m(1 - \eta_{\text{screw}}).$$  \hspace{1cm} (3.44)

The resulting torsional load is on the order of 0.4Nm which is sustained by a locknut configuration in the push-rod end that affixes the screw to the rotary output.
3.3 Mechatronics and Controls

The mechatronic system is composed of a control unit that houses a mid-level controller, motor driver, battery management system, sensor inputs and communications to external peripherals. This architecture enables on-board autonomous control of a powered prosthesis with expansion to more computationally intensive control from external devices. The control unit is a modified version of the Dephy Inc. FlexSEA system [108]. The mid-level controller is built around the ARM Cortex-M4 STM32F427 180MHz microprocessor. This control board is referred to as Manage and is able to perform single precision float calculations in one processor cycle. The main control loop on Manage operates at 1kHz. A lower-level controller referred to as Execute is based around two PSOC-5 microprocessors. One processor monitors and manages the battery power systems. The second processor performs the motor commutation, communications and reading sensors. The Execute system operates at 10kHz. The external sensors include an absolute on-axis joint encoder (AS5048B), incremental off-axis motor encoder (RLM2), and load-cell (LCM300). Internal sensors include an inertial measurement unit, motor phase current sensor, wheat-stone bridge strain-gauge amplification circuit, and a temperature sensor. The peripheral communications allow communication between a host computer through USB, Bluetooth and RS485 for additional leader/follower FlexSEA units. For example, the RS485 is used for communication between the ankle and knee when used in combination. Additionally, an I²C bus is used to communicate with an external electromyography (EMG) amplifier board designed by Seong Ho Yeon also from the Biomechatronics Group. The block diagram of this control unit is shown in Fig. 3-32.

Selecting an encoder requires verifying the embedded system can track the signal frequency. A maximum resolution encoder mounted on a motor may lead to high frequency pulses in the MHz range that can cause lost counts or system hanging by overwhelming the interrupt service routines (ISR) of the microprocessor. If the microcontroller does not have timing hardware dedicated to a quadrature encoder (QEI) the ISR must stop the microcontroller in order to count every pulse that comes down the line. If these pulses come too quick it can lead to uncertain behavior in the system or even hanging of the system. To calculate the allowable encoder resolution:

\[
\frac{cts}{rev} = f_{update} \cdot \frac{1}{\omega_{motor}} \cdot 2\pi
\]  (3.45)

where, \( f_{update} \) is the maximum allowable update frequency on the microcontroller, and \( \omega_{motor} \) is the maximum expected motor velocity. This maximum velocity should
account for both expected steady-state operation and the potential for impulse responses such as occur at impact events with high acceleration.

### 3.3.1 Torque Compensator

The torque compensation loop in Fig. 3-33 operates at 1 kHz on the Manage control unit board and requests motor current from the motor-driver. The current controller is operating at 20 kHz. For preliminary evaluation the torque compensator is a proportional integral differential (PD) controller with a feed-forward command. Controller gain tuning is done manually with the joint in a statically locked position. The test setup is visible in Fig. 3-36. Desired torque step inputs of 30 Nm are commanded and gains adjusted to find a system response that balances overshoot with settling time, weighted more towards settling time.

There have been difficulties in achieving the results expected from simulation. These issues are discussed more in Section 4.2, and are likely related to a mismatch in motor driver switching frequency and motor inductance. As shown in Section 3.2.2, simulations show reaching desired torques and power with currents below 40A, and
voltages within the system specification. However the system has not managed to reach expected torques in closed loop mode and seems to have difficulty exerting high power torque control the motor. Critically damped step response has not been achievable with simple control, due to motor current saturation. Underdamped response does work well as shown in measured data in Fig. 3-38. The controller shown in Fig. 3-34 with simulated results Fig. 3-35 was implemented on the hardware. However, the results are not effective even at very low control commands, and the hand-tuned PID with simple feedforward as shown in Fig. 3-33 continues to outperform these models.

Figure 3-33: The basic controller architecture consists of a high or mid-level controller defining desired output impedance terms. A torque control loop then asserts a desired current to the motor driver which commands current into the motor phases.

Figure 3-34: Feedforward model based control block diagram. The low-pass filter is implemented within the feedforward term to make it realizable.

3.3.2 Impedance Controller

Joint torque is specified by the mid-level impedance controller which then commands a desired current to the motor driver to enforce at the motor. The impedance torque
Figure 3-35: Simulation results of the feed forward model based controller. (a) Bode plot of expected response. (b) Torque command with saturation at system limits.

The torque command follows the form:

$$\tau_d = K_d(\theta_{des} - \theta_t) - B(\dot{\theta}_t)$$  \hspace{1cm} (3.46)

where, $\tau_d$ is desired joint torque specified to the torque controller, $\theta_{des}$ is desired joint angle set point, $\theta_t$ is measured joint angle, and $\dot{\theta}_t$ is measured joint velocity. The initial impedance parameters used are from [109]. Results from user testing reveal these parameters must be modified substantially to accommodate the different hardware response and varying user preference.

### 3.3.3 System Characterization

This actuator is a reaction force series elastic actuator; the actuator free-space and high-impedance behavior differ from a traditional SEA. The primary difference is the spring is serially located in the ground-path of the motor, rather than the output path of the actuator. The lumped parameter assumption commonly used for a SEA is not valid for free-space motions. In the high-impedance case motor-mass is also sprung with the rotor inertia. The performance of the high-impedance condition differs from the low-impedance moving output condition by enabling an impulse load to deflect the spring and motor without rotating the motor inertia, thus improving high-frequency inertial conditions [67], [110]. To verify actuator load capabilities on the bench-top the locked output rotor experiment is used. This is the worst-case
condition of a high-impedance/inertial load. Experiments pushing the system to its designed maximum fatigue limit load rating of 175Nm and beyond to the maximum allowable deflection\(^7\) torque of 215Nm were performed on the test setup as shown in Fig. 3-36.

Figure 3-36: Benchtop test with a fixed output. This image shows the actuator applying a 218Nm output torque – at this instant the maximum deflection of the spring is noticeable. This torque is above the rated repeated peak loading condition of 175Nm; it demonstrates what high torque is achievable with this actuator.

To identify system characteristics the locked rotor, high-inertia frequency sweep is used [111], [112]. This includes sending a linear, sinusoidal chirp command of ±50Nm torque amplitude, 2.5Nm Gaussian white noise and a frequency range of 0.1-12Hz to the motor driver. Equivalent mass, stiffness and damping characteristics are then extracted from these models and used in control simulations to further tune the torque compensator. The same technique is also used to identify closed-loop response of the system when run with a compensator on the torque command.

Initial system characterization evaluated the open-loop frequency response of the actuator when a linear sinusoidal chirp was applied to the desired actuator torque. The amplitude was set at ±50Nm and frequency sweep from 0.1 Hz to 15 Hz over 5 seconds. The system aligns well with a second order mass spring damper system as

\(^7\)The maximum deflection of the spring is limited by hardstops built into the frame. The plantarflexion direction has an adjustable hardstop while the dorsiflexion is limited by the spring moment-arm contacting the mounting frame of the actuator.
characterized by:

\[ G(s) = \frac{90}{s^2 + 20s + 120}. \]  

(3.47)

The natural frequency is measured at about 1.5 Hz and calculated from the estimate above at about 1.6 Hz. These values are lower than expected from the simulation based analysis. The estimated system natural frequency was 2.1 Hz to match the desired actuator performance targets in Table 3.1. Likely the overall system series stiffness is lower than expected due to the contribution of the structural stiffness. During simulation an infinitely rigid structure is assumed. However, during design the structural stiffness was matched to that of bone, allowing up to 3mm of deflection at maximum load. This elasticity was not accounted for again in the simulation and is a likely contributor to reduced natural frequency. Additionally, the lumped mass model of the actuator was assumed to include the rotor inertia and motor mass as sprung mass. However, the spring mass and moment arm mass were not included in this sprung and lumped mass evaluation at the time of simulation as they were not known at the time. The contribution of increased mass and decreased overall series stiffness both would contribute to a lower overall actuator natural frequency.

To evaluate the closed-loop performance of the actuator the closed loop frequency response is tested with a linear sinusoidal chirp signal. The amplitude is set at \( \pm 50\text{Nm} \) and frequency sweeps from 0.1 Hz to 15 Hz over 5 seconds. The system aligns well with a second order mass spring damper system as characterized by:

\[ G(s) = \frac{920}{s^2 + 30s + 980}. \]  

(3.48)

The natural frequency is 3.9Hz. The closed-loop -3dB bandwidth is 6.2Hz with a phase margin of 61.5 degrees.

The torque PID controller is manually tuned to achieve an under-damped response with relatively quick rise-time. The response to a 50Nm step-input applied to the locked-rotor configuration is shown in Fig. 3-38. The rise time \( T_r \approx 0.072 \text{s} \), peak time \( T_p \approx 0.096 \text{s} \), with a percent overshoot of \( \text{OS} = 20\% \), and settling time of \( T_s \approx 0.232 \text{s} \). This results in a system estimate of closed-loop natural frequency \( \omega_n \approx 5.9 \text{Hz} \), and damping ratio \( \zeta \approx 0.47 \).

\[ G(s) = \frac{1368}{s^2 + 34.5s + 1368}. \]  

(3.49)
Figure 3-37: Open-loop frequency response shows a damped second order system with natural frequency at 1.5 Hz and phase offset of $-80^\circ$. Closed-loop (PID) frequency response shows a damped second order system with natural frequency at 3.9 Hz, control bandwidth at 6.2 Hz, and phase margin of 61.5 degrees. Analysis was performed with a $\pm 50 Nm$ amplitude linear sinusoidal chirp. The solid line is the estimated system response.
3.4 Preliminary Clinical Evaluation

Subject testing was performed at the Massachusetts Institute of Technology (MIT) Media Lab Biomechatronics Gait Laboratory. Participants were informed of and consented to test protocols approved by the MIT Institutional Review Board: the Committee on the Use of Humans as Experimental Subjects. All subjects self-reported to be healthy with activity levels at or above K4: having ambulatory activity with variable cadence. The Biomechatronic Gait Lab facility includes a 15m walkway, a VICON motion capture system, and an instrumented split treadmill.

These studies include three healthy male and one female subjects aged 46 ± 8 years with unilateral below knee amputation 10 ± 13 years post amputation and weighing 81 ± 10.2kg. Additionally, an initial experiment with an above knee subject was performed but no data is presented at this time. Experimental procedures included donning and adjustment of the TF8 to match the alignment and comfort of the par-
participant’s standard issued prosthesis. Control parameters were then tuned to match the comfort level of the subject as they acclimated to walking at self-selected speed with the powered ankle-foot system. The parameters available to tune include virtual parallel stiffness, push-off torque threshold, push-off torque ramp rate, push-off stiffness, toe-off angle and damping. During the experiments, on-board sensors recorded and transmitted the full actuator state across a wireless bluetooth communication protocol to a secondary computer at a rate of 100Hz. Data collected included joint angle, velocity, torque, walking state estimate, as well as motor current and voltage.

3.4.1 Finite-State Machine

To demonstrate the ability of TF8 to achieve biologically relevant kinetic and kinematic behaviors an ankle walking finite-state machine is implemented in the mid-level controller, following similar methods as in [41], [113], [114]. The state transitions are shown in Fig. 3-39. Stiffness, damping, torque thresholds, push-off toe angle, and torque ramp rate are all manually tuned to user preference during an acclimation period. After a user feels comfortable with walking on the device at self-selected walking speeds they are prepared to begin data collection studies.

Stance is triggered by a heel-strike transition identified by an absolute torque signal greater than a set threshold. Controlled-dorsiflexion superposes a unidirectional virtual parallel spring on top of the stance impedance settings. As the subject leans into the virtual parallel spring joint torque rises until reaching a specified threshold at which point the powered plantarflexion state triggers. The joint angle set point then ramps to the next position along with a transition in virtual stiffness and damping. As the user is propelled forward and lifts their foot the torque drops below a threshold and the swing phase is triggered. Swing phase then rapidly moves the toe position into a dorsiflexed state to provide clearance throughout swing and the controller waits for the next heel strike.
Figure 3-39: Schematic and human subject demonstration of state transitions: (a) State transitions are triggered by torque and angle thresholds tuned to meet test subject preferences. (b) A subject walking with the finite-state machine controller. Note during powered plantarflexion the additional series elasticity contributed by the Variflex foot that is not accounted for in the numerical models.

3.4.2 Walking Experiments

Torque and power from a 70kg person with unilateral below knee amputation walking with a finite-state machine controller on a treadmill at 1.5 m/s is shown in Fig. 3-40(a). Twenty-eight strides were acquired and aligned on a percent gait cycle plot. The ankle joint mean torque shows a slight phase-lag and undershoot of mean biological data, though does achieve about 108Nm at powered push-off and an $R^2 = 0.85$. Mechanical power aligns well with mean data, $R^2 = 0.9$, but again the 250 W measured at the joint undershoots the biological value of 380W. Net positive work with a mean of 11J is shown in Fig. 3-41(a). This cumulative joint energy lays within one standard deviation of the mean of biological data for a mass and speed matched person.

A second person, weighing 84kg, and walking at a self-selected speed of 1.0m/s is shown in Figs. 3-40(b) and 3-41(b). Six strides were identified from this walking experiment. In this case joint torque aligns well with the biological mean. Ankle power shows a small phase difference but achieves maximum spikes of 350W. The joint range of motion shows similar morphology to biological but does not reach the same maximum dorsiflexion angle of biological, maxing out at 7 degrees rather than 20 degrees.

The electric energy performance of the prosthesis measured against the simulation expectations are shown in Fig.3-42. The mean cumulative electric energy expenditure measured from a 70kg person walking at 1.5m/s is 28J, shown as the solid line. The
Figure 3-40: Preliminary results from two subjects walking with the finite state-machine controller. Subject one data (a-b) is treadmill walking at 1.5m/s. Subject two data (c-d) was collected during self-selected walking and heelstrike transitions were aligned to collate data. The purple regions are one standard deviation of data collected on the TF8 prosthesis, the black line is the mean. Grey is one standard deviation of able-bodied walking data from [9].

spread is one standard deviation of the measured data. The simulation results in a 38J mean expected electric energy expenditure. The coefficient of determination, $R^2$, showed a 0.97 fit to the expected data.
Figure 3-41: Preliminary results from two subjects walking with the finite state-machine controller. Subject one data (a) is treadmill walking at 1.5m/s. Mechanical energy output at the joint shows alignment within one standard deviation of the biological dataset for 1.5 m/s walking. The Subject two data (b) was collected during self-selected walking and heelstrike transitions were aligned to collate data. The purple regions are one standard deviation of data collected on the TF8 prosthesis, the black line is the mean. Grey is one standard deviation of able-bodied walking data from [9]. In (b) the joint angle matches data seen with other participants: subjects prefer a stiffer than biological ankle, limiting their dorsiflexion range of motion.
Figure 3-42: Cumulative electric energy consumption by the robot during the 1.5m/s walking trials. Dotted line is simulated energy consumption, shaded is mean and one standard deviation from 28 strides. The mean overall electric COT is 0.053 with a cumulative energy of $28 \pm 4J$. 

$R^2=0.97257$
3.4.3 Non-linear Stiffness

Initial experiments to verify flexibility in control are performed by applying a non-linear spring to the finite-state machine virtual spring. Rather than a linear spring a biologically matched torque angle curve is interpolated in a lookup table. As the participant leans into controlled dorsiflexion the control unit looks up a mass scaled desired output torque based on the instantaneous measured joint angle. The biological torque angle curve has a different rise and release curve during mid-stance and late-stance. Initial experiments required reaching the maximum joint angle in order to transition to the release or push-off spring-rate. The participant was not comfortable with this much dorsiflexion during walking so the second non-linear spring trajectory was not achieved in these initial experiments.

Figure 3-43: Tracking performance of the nonlinear stiffness term based on biological torque-angle data. Biological torque-angle scaled to an equivalent user mass of 50kg is $\tau_b$, torque measured on the device is $\tau_m$, and $\tau_d$ is the desired torque commanded at the actuator.

The non-linear stiffness from a biological gait replayed on the actuator is shown in Fig. 3-43. The grey line is the biological torque/angle curve. The blue line is the desired torque calculated, and the purple line is the measured torque. The measured torque tracks well during controlled dorsiflexion but the participants quick release of the ankle alters the release rate of the spring, giving a sharper release spring-rate. The Maximum torque is lower than would be expected for an equivalent unaffected individual. The participant prefers a shorter stride and joint range of motion than
the unaffected biological dataset suggests. In order to achieve transition into the push-off state the output torque is scaled by a reduced user mass of 50kg rather than the 84kg of the user. Even as such, the user had to greatly adjust their stride to an uncomfortable level in order to reach the nearly 15° transition threshold.

3.4.4 Stairs

Preliminary stair walking studies were performed with two unilateral BKA participants: one male and one female. One study participant donned a BiOM, and a TF8 actuator while the other wore only the TF8 for these studies. Each device was tuned to comfortable walking for the participant. The experiments were performed on a four-step staircase mounted on the gait lab walkway, shown in Fig. 3-44. Due to the short length of the staircase limited steady-state data is available – it is affected by end-conditions: stopping and starting at the top or bottom of the stairs. The stair climbs were part of a study on adaptive terrain control of which this author is a co-author, more details can be found in [115]. Heuristic measurements from an onboard inertial measurement unit are used to automatically identify flat ground, stairs or inclines. From this terrain identification the control parameters are modified for the appropriate task. In stair ascent and descent biological ankle joints traverse through a wider range of motion than that on level-ground. The leading toe will plantarflex in preparation for the ascending or descending toe. This provides stability and reduces impact compared to a passive device that lands on a flat foot. Finite-state machine impedance parameters are adjusted by the terrain adaptation controller to approximate the matching biological gait profile.

The data in Figs. 3-45 show the TF8 is able to replicate biological range of motion within the constraints of the accuracy of the implemented control system. The top row is BiOM, the middle row is TF8 and the bottom row is biological data. Comparision between top and middle Figs. 3-45 (b) clearly marks the difference between the limited range of motion BiOM and TF8 system. The BiOM is able to generate biomimetic torque at the joint, but it is not able to replicate natural kinematic motions.

Joint torque during stair ascent and descent are compared between TF8 and biological torque in Fig. 3-46. The solid line is mean and the spread is one standard deviation. The plots show that biological torque morphology can be recreated with the actuator, though it does not track entirely accurately. These differences may be due to controller accuracy, and potentially the motor driver power saturation discussed. It is clear from these figures that the system has the feasibility to replicate
biological joint kinematics and kinetics.

The metabolic effects of accurate kinematics at the joint are beyond the scope of this design thesis, but are likely to prove beneficial to the user, and the ability to replicate them are core to the thesis. When joint kinematics do not match biological it requires other portions of the body to compensate. This may appear as increased metabolic energy expenditure or increased effort from other parts of the body, causing for example, chronic back pain. Does providing biomimetic torque alone reduce metabolic expenditure or does the lack of range of motion require additional muscle recruitment on the contralateral side of the body? Thus, does providing more biomimetic motion functionality reduce the energetic expenditure of walking? These
Figure 3-45: Stair ascent with one human subject wearing a BiOM and a TF8 ankle prosthesis with a stair ascent controller, experiment described in [115]. The top row is BiOM data, the middle plot is measured on the TF8, and the bottom plot is biological data from [65]. (a) Joint torque, and (b) joint angle.
are questions worth asking when evaluating design decisions in powered prosthesis design.

### 3.4.5 Knee

Initial experiments have been performed with one male subject with unilateral above knee amputation. Fig. 3-47 shows first steps with a user wearing the combined knee and ankle configuration. Both TF8 actuators are identical hardware configurations. The walking controller is the same finite-state machine described above, and is running on both systems. The ankle determines walking state. The difference on the knee controller is that it does not identify its own state, but rather requests state updates from the ankle at 200Hz. The knee then transitions states based on these commands from the ankle. The tuning procedure is similar to that of the ankle where user preferences are used to determine impedance set-points of desired joint angle, stiffness and damping. Due to issues with bluetooth communications walking data has not yet been collected, but it is underway for future studies.
3.5 Actuator Performance Comparison

The measured results of eCOT for the prosthesis are compared to the simulation expected result. For the participant walking at 1.5m/s he measured eCOT of the prosthesis results is 0.053, using Equation (2.15) and the result from Fig. 3-42. This value is 10% lower than the simulated expectation of a 0.059 eCOT for a similar user mass person walking the same speed. This result is plotted on the surface and contour maps originally used to identify a design specification capable of achieving both knee and ankle design criteria, Fig. 3-48. The dotted black lines in the top plot show the as-built actuator configuration. The dotted lines in the bottom plot represent the expected electric COT, and the orange dotted line shows the measured result.

The performance metrics of the TF8 actuator are compared against other published ankle prostheses. The system specification and results are shown in Table 3.1,
Figure 3-48: The eCOT measured in the walking trials is overlaid with the original design specification energy map adjusted to simulate the 70kg participant walking at 1.5m/s. The dashed lines on the top plot show the system configuration tested. On the bottom plot, the dashed orange line is the 0.053 COT measured at the ankle prosthesis during walking trials, the black dashed line is simulated and results in a 0.059 COT.

and presented again, here, in Table 3.7 as a convenience to the reader. Direct comparison across prostheses is difficult due to differences in reporting. User mass normalized torque and power are helpful for biomechanics comparison, but are not accurate in describing maximum capabilities of the prosthesis hardware. Some devices have benchtop data available, while others only have walking data. This walking data may not show the full capability of the hardware. In some cases only range of motion is given, or control bandwidth is stated but without a torque magnitude specified. Similarly, mass measurements of the actuators are often not comprehensive, showing only mechanical hardware without batteries or electronics, or some mixed combination. Research-grade hardware is also of very different robustness than commercial-grade.
Table 3.7: Actuator Design Specification & Results

<table>
<thead>
<tr>
<th></th>
<th>Target $^+$</th>
<th>Ankle Result</th>
<th>Knee Result</th>
<th>Units</th>
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<tbody>
<tr>
<td>Range of Motion$^1$</td>
<td>45-0-65</td>
<td>35-0-75</td>
<td>0-110</td>
<td>Deg</td>
</tr>
<tr>
<td>Velocity</td>
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<td>6.8$^*$</td>
<td>6.8$^*$</td>
<td>rad/s</td>
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<td>175</td>
<td>175</td>
<td>Nm</td>
</tr>
<tr>
<td>Max Power</td>
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<td>400</td>
<td>400</td>
<td>W</td>
</tr>
<tr>
<td>Bandwidth Magnitude</td>
<td>82</td>
<td>82</td>
<td>82</td>
<td>Nm</td>
</tr>
<tr>
<td>Bandwidth Frequency</td>
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<td>6.2</td>
<td>6.2</td>
<td>Hz</td>
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<tr>
<td>Segment Mass$^2$</td>
<td>2.0</td>
<td>2.18</td>
<td>1.6</td>
<td>kg</td>
</tr>
</tbody>
</table>

$^+$ Actuator specification is determined based on a 90kg user walking at 2m/s.

$^1$ Range of motion as shown: dorsiflexion - neutral - plantar flexion. For a knee walking requires up to 72°, sit-to-stand and stair ascent require ≤ 95°, and 21-0-40 for an ankle [65], [89].

$^2$ The equivalent body segment mass is 2.9% of a 90kg person [68], [90].

* Calculated due to power electronics limitation, based on observed behavior. $\omega = \frac{48V \cdot \eta \cdot 100Kv \cdot 2\pi}{60 \cdot \frac{1}{N}}$.

Consumer products must be built to withstand lifecycle testing, and safety requirements posed by government and insurance regulating bodies. These restrictions often lead to heavier devices. In a similar vein, commercial vendors also have larger teams and more resources to put into designing hardware, than individual academic research projects led by one or more graduate students. Direct comparison is thus difficult to fairly administer, but some comparison metrics are nonetheless helpful in evaluating different designs and their potential impact to the field.

In Figs. 3.5 and 3.5 raw performance metrics are used to compare absolute maximums across devices with real stated mass on the horizontal axis. The first plot, Fig. 3.5 only includes ankle prostheses with all electronics and batteries integrated into their stated mass. To show a snapshot of the whole field the second plot, Fig. 3.5 includes all powered ankle prostheses the author has found published. Not all plots show all prostheses due to under-reported metrics. The legend shows small asterisks by the devices that do not include batteries in their mass, and in some cases they also do not include electronics. The metrics are collected from journal publications using each of the respective devices. In these plots the area enclosed by the dotted lines indicate the region that performs better than the base walking data used to define the design specification: 90kg user walking at 2m/s. The area enclosed by the shaded region is performing better than the one commercial device, the emPower. The purple asterisk shape is the TF8 configured as an ankle prosthesis, and the blue triangle is the emPower. The design goal to improve upon the commercial device, and biology is to build an actuator that is within both shaded regions.
Figure 3-49: Comparison of powered ankle prosthesis performance. The area enclosed by the dotted line indicates a region that out performs an equivalent biological ankle at the specified target walking speed of 2 m/s. The shaded box indicates performance greater than the only commercial device: the EmPower. The design objective is to be within the areas enclosed by both the dotted line and shaded box. The devices shown all include electronics and battery mass on-board. Comparison between prototype and commercial devices is not necessarily a direct evaluation; commercial products may have different design targets due to consumer preferences, and they generally become heavier to accommodate packaging and robustness requirements due to government or insurance regulation.
Figure 3-50: Comparison of powered ankle prosthesis performance. The area enclosed by the dotted line indicates a region that out performs an equivalent biological ankle at the specified target walking speed of 2 m/s. The shaded box indicates performance greater than the only commercial device: the EmPower. The design objective is to be within the areas enclosed by both the dotted line and shaded box. The MIT TF8 is shown with and without battery for comparison, but is nominally operated with battery mounted proximally on the socket. Comparison between prototype and commercial devices is not necessarily a direct evaluation; commercial products may have different design targets due to consumer preferences, and they generally become heavier to accommodate packaging and robustness requirements due to government or insurance regulation.

* Prosthesis mass does not include battery, nor possibly electronics.
3.6 Discussion

Benchtop and human walking trials demonstrate the TF8 RFSEA can achieve biological kinetics and kinematics of both knees and ankles. The simple and adaptable design reduces hardware maintenance complexity, and system mass. The measured mean electric COT of 0.053J/kg at 1.5m/s walking speed surpasses the simulation expectations and outperforms the Au, Herr [41] ankle that achieved 0.06J/kg at 1.25m/s. This verifies the simulation based multi-pass design optimization driven by a minimum electric energy and mass objective is a successful approach to mechanical design when coupled with bold machine design.

3.6.1 Design

The cantilever-beam reaction-force series elastic configuration enables convenient and mass efficient mechanical packaging. The moment-couple generates uniform strain along the length of the cantilever-beam spring, thus maximizing material utilization while minimizing mass. In contrast to a traditional SEA arrangement of coil springs stacked serially along the linear actuator axis [111], the spring arrangement in TF8 allows a build height reduction by wrapping the spring back along the length of the linear actuator. Traditional SEA configurations also rely on secondary linear bearings to support motion of the intermediary coupling between spring, motor and output. In the TF8 design presented here, the spring acts as a flexure to support displacements of both the spring and motor, removing the mass and volume of extra bearings and motion stages. This design improves upon the leaf-spring configuration [116] used, by replacing the universal joint constraints with the flexure behavior of the spring. Axial play due to ever-present manufacturing misalignments are managed with spring washers mounted in the perpendicularly arranged pivots. Using a large gap-radius motor with relatively high torque enables the removal of a commonly used intermediary belt-drive gear reduction. Simplification of the design reduces assembly complexity, and more importantly, mass. In a similar vein, integrating the torque-motor as a frameless configuration with ballnut mounted directly within the motor rotor reduces the redundant mass often required to serially couple a screw and nut to a motor shaft. Combining a ball-nut with a motor is commonly done, but integrating an outrunner motor rotor and the use of a yoke pivot is most similar to [63], but, rather than serially stacked discrete components, this new design integrates the nut directly into the rotor and the supporting yoke directly into the stator support. This integration allows the ball-screw to pass entirely through motor, increasing range of
motion while decreasing build height, and reducing overall mass and package volume.

The reaction force series elastic actuator (RFSEA) inverts the spring placement of
the SEA, placing the spring in series with the ground path of the motor, rather than
between the drivetrain load-path and output. Much of the behavior of a traditional
SEA remain. A transfer function comparison of the actuator performance is character-
zized by Paine [110] and Orekhov [67] showing output response equivalence between
the output of the RFSEA and the SEA. Two major drawbacks do exist: where the
sprung mass of the SEA is that of the drivetrain and rotor inertia, in the RFSEA,
the sprung mass includes the drivetrain, motor inertia and also the motor and spring
mass – thus reducing overall system natural frequency. Also of consequence is the
force in the spring can be higher than the output force, requiring substantial design
consideration of the spring element and its associated mounting hardware. In the TF8
design the moment on the spring is nearly double that at the output. Paine presents
an RFSEA actuator based on the 200W Maxon EC-4 Pole motor, a belt drive, and
ballscrew with rotary ballnut. The motor nut assembly mounted on steel coil springs,
and linear rails. Orekhov developed a RFSEA for humanoid robotic applications also
utilizing the Maxon EC-4Pole motor, belt drive and ballscrew coupled to a titanium
cantilever beam spring. In each of these cases, the primary reason for choosing the
RFSEA architecture is preferential packaging of the mechanical design. In the case
of the TF8, packaging is the primary driving factor to use a RFSEA configuration
to allow substantially enhanced energy storage in the spring, despite the trade-off in
reduced natural frequency.

Motor

The relatively high-torque of the U10 motor allows a lower gear reduction than the
smaller diameter Maxon EC-4 Pole inner rotor motor commonly used in other pub-
lished hardware [41], [47], [60]. Reflected inertia is a critical parameter in both control
bandwidth as well as user safety in high impedance contact conditions. Larger radius
motors have an \( r^2 \) inertia increase yet are not subject to as large of a \( N^2 \) drivetrain
reflected inertia contribution. The trade-offs between reflected inertia of rotor and
drivetrain tends to nullify one another [71]. Since reduced gearing generally benefits
drive-train efficiency, and reduces operational speeds, larger torque motors with lower
reductions tend to have slightly improved performance in torque generating efficiency
and audible noise.

The larger inertia of the outrunner motor does, however, have a drawback related
to its larger inertia. The reaction torque from high acceleration motions can propagate
through the structural chain and to the user. This motor can generate up to 4 Nm of torque in normal operation, and while that reaction torque was initially considered in the loading on the spring, its effect on the user was not realized until parts were already under manufacture. This effect becomes apparent when no smoothing is used between transitions in impedance set points for the walking controller. Solutions to minimize the reaction yaw torque on the user include: introducing a ramp to the change in the desired joint position, smooth trajectory generation with minimum jerk, or including a low-pass filter to the desired torque signal. Setting a 30Hz low-pass butterworth filter on this torque signal showed effective improvement for reducing the slew-rate on the motor without the added complexity of handling transitions between impedance setpoints.

A secondary user experience driver towards larger torque motors is that smaller diameter motors operate at higher rpm, generating higher frequency audible noise. Removing this high frequency element can reduce the audible noise range to a more qualitatively acceptable frequency range. Humans experience the same sound pressure level as different volumes based on frequency. Though not quantified on the TF8, comments from users is that the sound seems less noticeable than other systems. That said, the test subjects are often in state of excitement regarding testing a robotic prosthesis. Their experience is often limited to only the time on the machine in the lab, and could likely change with long-term experience with the device. Some feedback from users of the emPower system is that everyone notices you when we walk through the office, or into a library. Those same users have said they think the TF8 sounds quieter and is at a more pleasing tone.

**Springs and Controls**

The moment-coupled cantilever-beam reaction-force spring arrangement excels in a few ways and has some drawbacks discussed below. The flat plate beam can be routed back along the length of the linear actuator, reducing overall system height when compared to helical springs. An equivalent metal coil spring capable of storing the 9J of energy requested, is estimated to be 100mm in diameter with 15mm thick rectangular cross-section, the mass of which is measured in kilograms rather than the 100 grams for the TF8 composite spring. The cantilever beam also acts as a precision flexural element providing its own translational constraint negating the need for additional bearings or guideways to constrain deflection motions. Finally, manufacturing of a flat plate cantilever beam is relatively straightforward, enabling multitudes of springs to be available to tune performance to different users or applications. The TF8 design
enables static reconfiguration: by releasing four bolts the spring can be swapped for another to match user mass and application. This functionality is similar to [117] but without a complex and massive taper locking attachment means, and without adjusting the cantilever length. Construction of the flat plate from fiber composite structures also enables aligning unidirectional tensile fibers, exploiting the primary benefit of composite structures – enabling composites to far out perform metallic alternatives for energy density, as shown in Table 3.3. With stress loads primarily in the outer millimeters of the spring, future iterations of the spring might include a foam-core composite to substantially reduce the mass of the spring.

There are two unfortunate conditions that occur with the load path configuration of this actuator: the first is inherent to the reaction-force configuration and the second could be mitigated with a second design iteration. The actuator remains fully controllable even with both design flaws, however, they do show up as small non-linearities in operation. The reaction force spring configuration creates a relatively large sprung mass of the motor, moment arm and spring. The motion trace image in Fig. 3-30(a) demonstrates the motor motion when loaded. The additional reflected sprung mass of the motor in addition to rotor inertia during high impedance motions reduces system natural frequency making control potentially difficult near that frequency. This is overcome with closed-loop torque control as it is on the system, where the closed-loop natural frequency shows up at 3.9 Hz, but system the reaches a 6Hz control bandwidth. It should be achievable to hit higher bandwidth with a more appropriate motor drive as explained in Section 3.6.1.

It is possible the natural frequency is evident when the user lightly loads the toe and brings the foot-spring into play in a middle-level inertia condition. The SEA is evaluated in a high impedance condition (stance), and in low-impedance (swing). However, it is not bench tested in this middle-level loaded condition where the overall system stiffness is affected by the additional series spring of the attached flex-foot and the leg mass of the user. In this case system natural frequency is lowered by the low stiffness of the output foot, and the larger mass of a human leg. This behavior is actually evident biologically: when one’s leg is partially suspended on a toe, the mass of the calf combined with the elasticity of the muscle tendon body also oscillates at an equivalent frequency. The difference though is that an unaffected leg can be volitionally controlled to stop, whereas the robot must be lifted from the ground. This is an area open to further investigation.

During numerical simulation the overall natural frequency of the actuator was underestimated. Where the design constraint had been set for a 13 Hz natural frequency
actuator, instead the design resulted in a 1.5 Hz natural frequency actuator. This occurred primarily due to a unit conversion issue in the code. Fixing the design code resulted in a estimated natural frequency of 2.2Hz. The result of 1.5Hz is likely due to not including overall structural stiffness in the estimates of series spring stiffness. This has so far proven adequate for experiments with multiple subjects walking on the ankle, but potentially poses some complications or improvements for very high performance control.

Generally, it is preferred to keep the system natural frequency outside of the operating regime of the actuator because small commands can drive the system unstable. The optimization results consistently drove the natural frequency close to this operating zone. The likely explanation is that at this point the system requires minimal input to produce large output. Normally this condition results in uncontrollable behavior. However, a model-based controller that accommodates this condition can actually exploit this minimal effort input and further improve energetic performance of the actuator. For example, applying a damped notch filter at this natural frequency to the input command can adjust the controller to account for the system behavior, and reduce electric effort.

The torque control for these studies has been manually tuned by hand, and could be substantially improved with more sophisticated compensation techniques. Inclusion of friction compensation, no-load current slew compensation, model-based feed-forward control, and more sophisticated lead-lag control would all lead to higher performance from the actuator.

Overall electric cost of transport matches well with predicted values. Results of a user walking at 1.5m/s showed an eCOT 10% lower than expected. An agreement of 10% between simulation and built hardware running a simple finite-state machine walking controller experimentally validates the simulation models. The improvement in operation over simulation is likely due to more conservative selection of drivetrain or power-electronic efficiency. It could also be overly conservative estimates of rotor inertia and load inertia. A large contribution to the energy budget goes to accelerating the motor back and forth. The calculations assume the inertia includes both the rotor inertia as well as the inertia of the next link. That is the mass of the foot, or in the case of the knee, the added mass from an equivalent leg segment. This is likely the cause of the overestimate because it is known that the power electronics are not effectively generating torque with applied current, due to incorrect commutation (more in Section 3.6.1). What this conservative estimate means is there is still room for improvement in the simulation and design. Likely mass can still be trimmed from
the motor – for instance a shorter height motor stack, or perhaps a slightly narrower
diameter motor would still be adequate to achieve the desired performance.

Regarding system electrical energy regeneration: there is opportunity to regenerate
electricity during controlled dorsiflexion, a negative work phase of gait. However,
the low velocity motion limits the energy transfer to the motor. Most of the energy is
stored in the spring – this is evident in the spring displacement and actuator energy
curves in Fig. 3-7. The majority of the power flow occurs at the transition from
late-stance push-off to early-swing when the motor must accelerate out of its way to
allow the toe to return to dorsiflexed state to prevent toe-drag during swing. This is
where rotor inertia, load inertia, and internal viscous damping have a strong effect
on energetics.

Though it would appear there is limited electrical cost during controlled dorsiflex-
ion, the energetic analyses in Section 2 do show that a parallel springs in the ankle
could be beneficial. The design trade-offs to include the parallel spring is that of
kinetic, kinematic and design complexity. For this effort to build a hardware plat-
form that enables a wide range of research it leads to a design that aims to reduce
mechanical complexity, while also enabling a larger range of motion. A parallel spring
actuator that is terrain adaptive, or shoe adaptive would require an adjustable neutral
set point for the parallel spring. This added complexity is not worth the added
mass when the system efficiency is already as high as it is; according to the simu-
lations it appeared the system as-built would be able to generate proper kinetics in
combination with full biological range of motion.

**Drivetrain and Linkage Geometry**

All of the components were designed to manufacturer specified fatigue limits that
would sustain a 90kg user walking/jogging at 2m/s, 5000 steps every day. The most
heavily loaded components and most likely to fail first are the motor bearings and
ballnut. The ballnut will actually only last one year with maximum loads on every
step of every day, while all other components have a minimum of four years life for
the same loading conditions. Though one year sounds short it is assumed that people
do not usually run every step of every day, nor that the life of a laboratory device
would experience such usage. This extra life analysis does also help with the potential
overload condition that does occur in the laboratory setting when poor signals or poor
controls lead to unexpected behavior. Estimates of cycle life are possible by applying
the load trajectories to estimate loading and travel length of the ballscrew, bearings.

Some place for potential concern in the design might be the pivot bearings which
only undergo oscillating motions rather than full rotations. Oscillating motion is a difficult condition for bearings because for roller element bearing grease tends to collect on the edges, while only a portion of the roller elements sustain load. In some cases plain bearings can be preferential due to the larger distributed area over which stress is applied in addition to the polishing nature of the motion. Ingress of abrasive material can however lead to accelerated wear. To evaluate this effect, two experiments were performed: in half the actuators needle bearings were used for the motor pivots while in the other two plain polymer bearings were used. After two years of working with the actuators in benchtop and user studies one of the needle-bearing actuators was disassembled. In this tear-down wear on the pivot shaft at the location of the needle rollers is visible. From this result it is recommended to use the plain polymer bushing for future design iterations.

An issue in the load-path design is the ballnut resides an offset distance from the motor support pivot, visible in Fig. 3-30. This offset allows a moment to be applied to the ball-nut under heavy compression loading – likely limiting the lifetime of the ball-nut due to the geometry of the internal ball returns. In tension the configuration remains naturally stable, but in compression there is instability about the pivot. This pivot condition shows up as backlash or deadband around the zero load condition. This can be accommodated by preloading the nut against the screw by slightly rotating the moment arm yoke when it is clamped to the spring. A less assembly-technique dependent solution includes a modified nut assembly where a second ballnut sits above the pivot axis. This modification is prototyped on one actuator and visible in Fig. 3-51. The second ballnut stabilizes the screw against any applied moments by generating a force couple between the two ballnuts on either side of the pivot. In future designs it would be worth investigating if a single ballnut mounted directly intersecting the pivot would be adequate or if both ballnuts are preferred. A single, centrally located ballnut would be lower mass, and cost, and thus preferable.

The total range of motion affects overall system height as the screw must sweep through the arc traversed by the output arm. Larger output moment arms benefit reduced screw loading, motor currents and better utilization of bus voltage; however, they also increase screw travel distance (build height) and motor velocity requirements. The total range of motion is limited at either of the joint range limits by adjustable hardstops. These are additively manufactured of tough polymer by means of selective laser sintering or jet fusion. Fig. 3-30(d) shows the total range of motion of the actuator configured as an ankle-foot prosthesis with dorsiflexion and plantar
flexion angles labeled.

Some deficiencies of the ballscrew is its low efficiency at low force. From an energetic perspective low efficiency at low load is fine because where there is little force there is little energy expenditure. However, for force control around a zero-point this is troublesome. The low efficiency limits the ability to use current-sensing for torque sensing about zero. In walking we are generally concerned with high power maneuvers, and zero-torque is not of concern. This may be why many system operate in a position control mode during swing phase: because, their low-torque controller do not behave well while unloaded. These results are insufficient for neurally controlled devices that must map to user intention. Neural control experiments often include a user manipulating their joints in free-space. This free-space motion being a dominant demonstration of neural control prefers a smooth operation around the zero torque point so as to not induce oscillatory or vibrational behavior that approaches the uncanny valley.
Mechatronics

The embedded system used in the TF8 should be upgraded in the future. The FlexSEA system is a nice low-power package for driving a motor, however it is not easily extensible, has limited documentation, and is not capable of effectively driving this specific motor. Further, the design tradeoffs for compact size and low-power operation make it difficult to use as a research platform. Wireless communication is performed with a Bluetooth 2.0 protocol that is low-power but unreliable. Combining poor communication protocols with no on-board storage is not adequate for a research system. In more than half of the user-testing experiments the Bluetooth signal would not communicate or frequently drop between the FlexSEA and any one of multiple laptop computers used for commanding the experiments and collecting data. This lack of communication makes it difficult to adjust control parameters for individual users and makes it impossible to collect walking data when there is no local storage. 

A major issue with the design of the motor driver is the switching frequency is not well suited to the motor in use in TF8. The calculated power output of the TF8 should be close to 1kW mechanical power, but in experiments it has been a struggle to demonstrate even 400W of power. The torque availability seems to saturate due to ineffective motor commutation. Current is being sunk into the motor windings, but it is not effective at generating torque. After extensive testing it was found that the motor inductance was 5.5 times lower than expected; the motor inductance of 30.3\,\mu H is much too low for the 20kHz switching frequency of the motor driver. The motor simulation expected to reach 175Nm torques with 35A of motor current, however in practice 60A has hardly achieved these torque levels. Initial thoughts were the battery system was sagging voltage and its ability to supply current. Yet, motor current does track desired motor current. The issue at hand is that there is a large current ripple in the motor commutation. The motor phase inductance saturates due to the relatively long switching period and controlled current is not generating torque effectively. To understand how inductance affects the current ripple for a motor the current ripple on a buck converter can be used as a tool[118]:

\[
I_{rip} = \frac{V_i}{R} \left(1 - e^{\left(\frac{-\tau_{sw}}{\tau_c}\right)}\right)
\]

where, \(V_i\) is input voltage, \(R\) is winding resistance, \(\tau_c = \frac{L}{R}\), \(L\) is motor inductance, and \(\tau_{sw} = \frac{\text{duty}}{f}\), or duty cycle over switching frequency. The original motor specification claimed to have 0.168mH and so a switching frequency of 20kHz would have resulted in a ripple current of 6.5A and 2.5W power loss, assuming a 50% duty cycle. However,
with the actual motor inductance being 0.03mH, this current ripple reaches 35A and over 75W of power loss in the windings! At higher duty cycle this gets worse, and explains why torque saturation was observed at outputs above 110Nm at the joint. This mismatch in switching frequency helps explain why the more sophisticated control methods devised were not effective at improving actuator performance. The motor driver has not been effectively generating torque with the applied current and cannot be relied on to generate any specified torques.

A motor commutation issue had been noticeable early on as an audible noise during motor operation. Initially it was thought that there was a commutation mismatch due to a potential mismatch with the high-pole count motor and the motor encoder. High pole-count motors are efficient at generating torque but are difficult to control, as the spacing between poles is short and at high velocity this requires quick counting of the commutation encoder.

In attempts to get higher power out of the system it was found that there is audibly more noise emitted from the motor when operated at 48V instead of 24V. This commutation noise couples directly to the load cell rendering the force signal unusable. A median filter was implemented to reduce the noise induced spikes. Even so it rendered a 10x reduction on torque resolution as well as changed the reference voltage measured by the load cell.

To alleviate the issues with both motor commutation and communication reliability a new mechatronic platform is under development. The new platform focuses on making use of more off-the-shelf components that are well documented and supported as commercial products. At this time a promising motor driver is the Everest XCR from Ingenia that operates with either a CANOpen or EtherCAT interface. CANOpen is a low-power and highly robust communication protocol, though it does not support the high bandwidth of the EtherCAT. The motor driver can operate at 100kHz commutation, and uses field-oriented-control for motor commutation. The high-level controller is based on a BeagleBoneBlue with native support for CANOpen, wifi, a Cortex M4 processor for mid-level sensor updates, and dedicated QEI inputs.

In early experiments with the Everest XCR audible noise is effectively absent, load-cell noise reduced to enable 0.1% full-scale resolution, and joint torques as high as 215Nm have been generated. It appears the field-oriented-control is effective at generating useful torque with the applied motor current. The system is yet to be fully integrated and no closed loop control has been implemented.

To be noted, there is a trade-off to high frequency motor commutation and that is switching losses. The design choices for FlexSEA were to be a low-power embedded
system for mobile, wearable robots, it consumes about 0.5W in standby. The XCR consumes 2.2W in standby and the high switching frequency may further increase energy cost. MOSFETs experience power loss while conducting and also while switching on or off. Higher frequency switching increases these losses. Certainly compared to 35A motor current ripples being wasted due to incorrect switching frequency are more costly than switching losses. However, for future high efficiency designs it would bode well to include motor inductance and commutation frequency in the energetic analyses. Following in that vein, tracking battery resistance and capacitor resistance would also be important to understand the power dissipated in high current draws from the battery versus local capacitance near the motor.

### 3.6.2 Preliminary Walking Results

Walking with a finite-state machine shows the actuator generally tracks biological kinetics and kinematics but has room for improvement. The joint torque shows some phase lag and undershoot compared to mean biological data. The discrepancy in torque and power from biological profiles may be due to both differences in the estimation of a finite-state machine, as well the limited torque tracking capability of the low-level torque controller, due to poor motor commutation at the motor driver level. Even with the torque saturation, the actuator is shown capable of achieving biological kinetics and kinematics within an $R^2 \geq 0.85$. It is also able to produce 0.16J/kg of positive net-work at the ankle that aligns within one standard deviation of biological walking energy. Measures of $eCOT = 0.053$J/kg at 1.5m/s outperform the simulations by 10% and track within $R^2 = 0.97$. Differences are likely due to conservative estimates of rotor and reflected inertias and transmission efficiency. The measured $eCOT$ is rarely published but should be. It is a useful metric to compare overall system efficiency, and that is important for mobile, wearable robots. As comparison the Au et. al. ankle has a $eCOT$ of 0.06J at 1.25m/s walking speed. Thus the TF8 is within the ballpark, but outperforms other comparable systems that are published.

The state-machine settings and transition parameters were tuned to user comfort, but this does not necessarily align with biology. Joint torque and power aligns well with unaffected biological data, but joint range of motion does not match natural kinematics. One hypothesis is that because amputee subjects are often accustomed to stiff, passive, or joint limited powered ankle joints some people find large ankle range of motion unsettling or even unstable. Though data from one user is shown in
In this study, nine subjects have worn the device during initial testing. A non-scientific evaluation of preference seems to suggest people with more recent amputation tend towards a softer, larger range of motion ankle and virtual parallel spring, than those who have become more accustomed to stiffer joints. The users in the study presented here had preference for a stiff ankle joint, limiting range of motion. An alternative explanation, though, may be that a non-linear virtual spring should be used in the walking controller. A biological torque angle curve shows a hardening spring, but a softening spring might also be effective at giving stability at low angle while being extensible to a wider joint angle.

In an effort to improve biological torque tracking a non-linear spring is also evaluated for mid and late stance based on the torque-angle curve from the mass-scaled mean dataset of unaffected test participants. The actuator does show strong tracking performance in the loaded condition. In the unloading condition the user more rapidly releases load on the joint than does the biological dataset. Also clear in the Fig. 3-43 heel-strike measured torques are larger than expected with this state-machine controller. Though far more stable than a time-based torque replay, this method does not account for the shortened stride common in subjects with amputation. This shortened stride reduces the maximum dorsiflexion angle a person experiences during their stride and limits the ability to achieve the maximum angle transition state defined with this non-linear spring stiffness control method. A future method might make use of the non-linear spring lookup table but find a way to jump over to the push-off spring curve upon a change in joint direction at a zero-velocity point, rather than a maximum joint angle. Alternatively, an inverted softening torque rate might be desirable. Despite these challenges the ability to replicate arbitrary non-linear stiffness during mid and late-stance demonstrates a capacity to more closely replicate biological motions and forces.

In addition to flat-level ground walking, initial experiments with stair ascent and descent also show promise to achieving biologically relevant performance. In Fig. 3-45 it is clear the TF8 system tracks much more closely to biological torque and angle than the BiOM. The BiOM does well generating torque but has very limited range of motion. The TF8, limited by power electronics as discussed above, is not able to generate the second large torque peak at pushoff, but it is able to show a double hump torque profile. Additionally, the TF8 is able to track biological range of motion within the bounds of the controllers intentions. In these studies it is clear the actuator has the capability to perform the range of motion, and torques, but can limited by the fidelity of the controller and its ability to command desired trajectories. This is an
area that can be improved with further terrain adaptive controls.
Chapter 4

Future Work
4.1 Motor Modifications

As discussed in Section 3.6.1, there is a mismatch between the FlexSEA commutation frequency and the motor inductance of the U10 Plus KV100 motor. This discrepancy could be mitigated by either changing the motor driver, or changing the winding of the motor. Increasing the motor inductance by three fold would bring the electrical time constant of the motor within an operable range of the FlexSEA system, enabling more effective commutation. This change would also increase the motor torque constant by 1.7x, resulting in a $K_t = 0.162$ compared to the current $K_t = 0.095$ value. However, this increase in inductance also comes with a 3x increase in winding resistance. The resulting RSR for this new motor winding configuration would be RSR=70, compared to an original RSR=221, meaning torque generating efficiency would be lower for this modified motor. Using the simulation described in Chapters 2 and 3 to evaluate the difference in energetic performance between motors, assuming effective motor commutation, this modified motor would consume 3% more energy than the as-built system. The motor constraint evaluations of these simulations comparing the two actuators are in Fig. 4-1. To really evaluate the trade-off in torque generating efficiency and switching losses a more complete energetic model that includes MOSFET transition and conducting resistances should be employed.

The Fig. 4-1 examines what motor performance would look like with a 100kg person walking at 2m/s on (a) the as-built motor windings and (b) updated higher inductance windings. Much more torque is available, potentially bringing the actuator into an interesting performance zone. Both systems saturate in power, but the modified motor does so in fewer points and at lower current. However, in Fig. 4-2 the updated motor is again compared to the as built system. The new winding configuration increases torque capacity, but also costs about 3% more energy to operate due to the increased motor resistance. Comparing the difference in winding resistive losses against switching losses would help lead to a preferential configuration.
Figure 4-1: These plots are the motor trajectories and limits of a 100 kg person walking at 2 m/s comparing different windings. (a) Using the motor U10 Plus KV100 (Kt=0.095) winding configuration. (b) Using the motor wound with a KV=59 (Kt=0.162).
Figure 4-2: These plots are the ankle electric and joint power and energy simulation for a 100 kg person walking at 2 m/s. (a) Using the motor U10 Plus KV100 (Kt=0.095) winding configuration. (b) Using the motor wound with a KV=59 (Kt=0.162). The higher KV value motor is slightly more efficient, but more difficult to control due to its low inductance.
4.2 Torque Sensing

Sensor fusion would be quite appropriately applied to the force sensor to provide robust and clean signal. Combining multiple sources of force sensing would increase reliability and safety of the torque control system, and could even be expanded to verify reliability or safety handling of other onboard systems. Force or torque control is what enables controllable contact dynamics and smooth life-like control in free-space, but it is entirely dependent on the ability to sense torque at the output. If the signal is corrupt it can quickly drive a system into its rails, causing breakage or even injury to a user. At best the load-cell in TF8 can provide a realistic estimate of force in the screw with joint torque resolution as low as ±0.3 Nm, which is 0.1% of full-scale (FS). A load-cell signal that has a large amount of noise can reduce the force sensitivity around the zero point, or even reduce the torque control bandwidth by discretizing the control into very coarse-resolution. At worst, a loose or broken load-cell wire can lead the system to believe force is very different than reality. In this event, the system can run full power into an end-stop, or potentially damage a person if that person becomes the end-stop. For this reason it is important to consider alternative force sensing measures.

Reading force by measuring the difference between joint encoder and expected motor position works well, except that due to the stiff spring, even nearly imperceptible backlash in the drivetrain can lead to inaccurate force sensing. This backlash appears as spring deflection, thus affecting perceived force by the embedded system. Future work might account for these discrepancies by mapping directional dead-band into the sensing code. Small deviation from the idealized machined components also leads to a non-linear force difference between calculated and measured force (when compared to the load cell). These nonlinear deviations can be accounted for by a calibration routine that searches for empirical dimensions that closely match the measured force. Measured joint angle and force at the load cell are then mapped to joint angle as the actuator is moved through a portion of its range of motion. A search is performed to minimize the error between measured joint angle and calculated joint angle by adjusting the geometry in Fig. 3-26(a). The result is a joint torque resolution of ±3.0 Nm or 1.7% FS.

There are also alternative means of measuring force that could be combined to increase measurement reliability. A magnet and hall-effect sensor could be used to measure spring displacement. Alternatively, motor current can be used to measure screw force, but is only effective at higher loads. Ballscrew efficiency is low at light
load, meaning the motor does not see screw force induced torque until the stiction, viscous damping and inertia of the balls in the ballnut are overcome. This means current sensing is not effective below roughly 10Nm of applied torque on the actuator, a force under which the motor does not see the output, resulting in $\pm 5.7\%$FS resolution at best. Despite this low torque resolution, high torque conditions can be measured and can be used to verify other torque sensing signals to improve reliability and safety, such as in the event of a loss of other strain-gauge or joint angle sensors.

4.3 Prescriptive Process for Clinical Tuning of Actuators

If the TF8 were a product prescribed in a clinic the user experience would be simple for both the user and clinician. The wide joint angle range of motion, and torque sensing could be used to improve automatic alignment for different user and shoe conditions much more so than traditional passive or commercial devices. To determine the correct spring a prosthetist would weigh the prescribee and evaluate their activity level. These two numbers would be used to cross-reference the target spring stiffness based on energy maps of Figs. 3-9 and 3-10. Similar maps based on preferred walking speed would also be similarly cross-referenced. A collection of springs would be in stock in increments of roughly 25kN/m from 100-500kN/m. It may make sense to combine springs in parallel meaning there are 100, 200, 300, 400 kN/m springs and then a collection of 25kN/m springs that can be added together for fine tuning.

The correct spring would be attached to the device and an automated control tuning procedure could be initiated either on the benchtop or while worn by the person. The person would be fitted by the prosthetist to best match their body and shoe configuration. Then the system would be turned-on.

An adaptive controller (in development, [119]) would further enable the true benefits of this configurable design. After only a troublesome few steps, the adaptive controller would converge on ideal gains for the individual actuator. Alternatively, the process may prefer the person stamping the ankle to provide impulse responses for the actuator controller to do on-the-fly system identification and gain tuning. Early simulations by a fellow student in the Biomechatronics Group have already shown the feasibility of an adaptive controller to converge on critically damped stability even with wildly varying model parameters of the TF8 system. The results of this convergence are shown to take only a few iterations in Fig. 4-3.
Figure 4-3: Results of an adaptive controller gain tuning on the actuator. The figure on the right shows the simulated controller as currently implemented on TF8, in blue, and an adaptive controller providing critically damped behavior within modest times. Borrowed with permission [119].
Once the torque control gains are adjusted the walking control gains can be tuned. Currently this is a manual process, but it is entirely feasible to connect the walking controller with a motion lab or additional sensors to close the loop between control parameters and walking performance. Walking control tuning could be down with human-in-the-loop control parameter optimization. Having the adjustable joint angle and torque measurements also enables automatic alignment for different shoe heel heights. This way the tuning procedure for standing up and walking on the device is nearly entirely automated. The simplicity and automated tuning should reduce time spent by the prosthetist making adjustments, allowing more time to be spent ensuring more difficult to adjust parameters such as socket fit can be coordinated to ensure a happy patient.

4.4 TFTiny

To address the tall clearance-height issue and to enable a wider range of smaller users to make use of the device, a smaller version, the TFTiny is in development. The height reduction is achieved by reducing the loading conditions, range of motion, and energy efficiency requirements. This new design is intended to be a lighter weight and shorter build height for smaller test-subjects. The push back on the energy requirements is to recognize its use as more of a research platform than a commercial product, at this point. The reason is there should not be a limit to achievable control bandwidth for energy reasons, when in the laboratory, it is best to see what is possible. The system parameters are derived from the same simulation and 3D models as presented in this dissertation and the TF8 system.
Figure 4-4: TFtiny is a reduced height version of the TF8. The height reduction is achieved by reducing the loading condition and energy efficiency requirements. It is derived from the same simulation and 3d models as TF8.

4.5 Rotary Actuator

For optimal energetic performance small modification to the architecture could be done to further tune the actuator to the application. The flexibility to replace springs to match user mass widens the functional space of the actuator. However, having even more flexibility in drivetrain options would lead to more efficient and more personalized prostheses.

The two main faults of the linkage-style actuator as used in the TF8 can be overcome with a rotary actuator. The first fundamental limitation of a linear actuator operating as a rotary output, besides range of motion and its coupling to stack height, is the limited torque accessibility at the output. That is: the linkage geometry must be tuned to the expected output trajectories in order to align the power-stroke of the actuator with the expected load requirements. A major limitation of this approach is there is highly discretized availability of reduction ratios for linear actuation. The ball-screw industry, besides being difficult to work with on a fast-turn prototype basis, only manufacturers limited sizes of ballscrews. This screw scarcity severely limits configurability of a design. Some reduction ratio tuning is possible with modification of the linkage geometry, but with that come additional trade-offs such as screw force or screw travel that lead to build height requirements that are non-ideal. For example,
in the design of the TF8 a 5mm lead screw is used. The adjacent sizes are 2, 3, 4 and 10 mm. The ideal for the knee may be l=7mm, but it is not an available size. Further, the load capacity and external dimensions of each of these leads, even within the same part-number category can also vary wildly, making adjustability extremely difficult.

If it were possible to have continuous variation of drivetrain reduction in an actuator it would provide further design authority to tune actuator properties toward biologically specific requirements. The TF8 actuator enables a simple flat-plate spring to be tunable and user swappable. The actuator also enables some flexibility in adjustment for varying lead ballscrews to adjust overall reduction ratio. However, the nature of manufacturer dependency makes ballscrew gear ratio selection highly discretized, and so, limited in adjustability to user requirements. Even across small changes in screw lead within the same manufacturer series there are differences in geometry that can conflict with other discrete components such as bearing sizes, that make swapping screw leads difficult in any generalized sense.

What would be ideal to tuning an actuator performance to an individual is an actuator that can have nearly continuous variability in gearing reductions with minimal variation to packaging geometry. The cycloidal drive is a drivetrain that potentially offers nearly continuous tunability between overall gear-ratios of 1:1 to 100:1 with the same or better efficiency than a ballscrew offers [120]. It has been suggested that constant efficiencies as high ballscrews in their prime operating conditions (>90%) are achievable (with proper design) irrespective of output load [107]. Once a standardized pin arrangement has been determined any number of gear reductions could be implemented within the same package. This would provide the designer flexibility to adjust not only spring stiffness but gear reduction as well, fully tuning a system to an individual user. Of course, this topology is not without difficulty. There are very high precision requirements to provide a near zero-backlash drivetrain, while not also binding, which would result in very low efficiency, or even an expensive brick. Further, due to the variation in lobe height, there is in fact a roughly 8% variation in gear ratio ripple throughout the rotation of the cycloidal disk [121].

Following the same simulation methods described in the preceding chapters combined with methods from [121], it is possible to generate expected loading conditions and component specifications for any combination of trajectories and associated actuators that make use of the cycloidal drivetrain. The Figs. 4-5 and 4-6 show an actuator based on the parameters of a 90kg user walking 2m/s. The bearings have been sized based on expected loading conditions, and part availability, though the structure has not yet been stress analyzed.
Figure 4-5: The cross section of a rotary actuator with a cycloidal gear reduction integrated into a motor housing.

Figure 4-6: The cycloidal drive offers an interesting opportunity to be adjustable between 1:1 to 1:100 gear reductions in a single stage. This graphic is looking at the tooth profiles of the pins and cycloidal element.
4.6 Recommendations for Comparison

The field of powered prostheses could do a better job of publishing standard metrics to enable easy comparison between built systems. Engineering drawings should be presented along with benchtop and preliminary clinical results. Explicitly stating measured dimensions of the prostheses such as in Fig. 3-29 or Fig. 3-31, and their associated masses, such as in Tables 3.4 to 3.6, would be helpful for readers to understand exactly what user populations could benefit from the presented hardware. Additionally, showing range of motion such as Fig. 3-30(d) would be helpful to the reader. Benchtop experiments demonstrating nominal and peak joint torque, and power should be provided in a locked output, and in a free output state. Explicit definition of the magnitude of closed-loop torque control bandwidth should be stated. Without a magnitude specification there is no clear understanding of the capability of an actuator to actually achieve a desired torque at a specified frequency. Preliminary walking data, perhaps with a standardized finite-state machine controller, would also be useful to show real-world performance. Finally, publishing the eCOT, even from preliminary results, would enable energetic efficiency comparison across device architectures. Other metrics, that have also not been published here, but would be helpful to gain understanding of user acceptance of prostheses would be noise level measured at one meter distance, and qualitative responses from users answering a Likert-type survey.
Chapter 5

Conclusion
The TF8 ankle and knee powered prosthesis actuator is the first robotic prosthetic leg that can be personalized to an individual user and task. It is also the first robotic actuator shown to achieve both ankle and knee biological performance, and to do so with minimal electric energy consumption. The flat-plate composite spring that defines the energetic performance of the actuator is simple and inexpensive to manufacturer, making it possible to stock large variations of springs to provide ample adjustment for users. The TF8 actuator shows promise toward enabling high-activity performance with new intrinsic and volitional control paradigms. By integrating the ball-nut directly into the motor rotor and utilizing the spring as both energy storage and motion constraint it has been possible to reduce the design complexity and build the actuator into a standalone actuation unit. It is capable of producing repeated 175Nm torques, biological range of motion and power, and has a mass of 1.4 kg - 2.2 kg, depending on its configuration. The high resolution torque fidelity enables smooth free-space motion, in addition to high-power ballistic maneuvers such as in ankle push-off. The TF8 could potentially be used for other humanoid, quadruped or robotic applications by replacing the structural elements with application specific hardware and appropriate springs. Assembled as an ankle, the device is a light weight research-grade powered prosthesis expressing the highest torque and power density of any systems the author is aware of at this point in time. Assembled as a knee, the device similarly outperforms all other knee or knee-ankle systems.

The system is not without design flaws though: the nominal design natural frequency is lower than initially expected, there are non-linearities around zero load due to some backlash in the ball-nut arrangement with respect to the motor support pivot, as well as a reaction torque at high motor acceleration. Neither of these design issues have proven a problem in initial testing with subjects and manually tuned PID torque control. The highest priority place to focus improvement of the actuator would be reducing the overall build height. As a knee it is smaller and lighter weight than any other system, but as an ankle with the inclusion of a Vari-flex foot, the system height is just at the height of the emPower. This height has been usable with all of our test participants, however, it has been just on the edge of acceptability for at least one subject. This design issue is being addressed with the build of a new, shorter prosthesis: the TFtiny based on the same simulation code and 3D models as the TF8.

The process of searching for a minimum electric energy consumption per gait cycle configuration of design components proves effective at identifying a hardware specification. Discrete availability of hardware components limits the ability to con-
continuously search across the space and thus forces additional compromise in the design. The kinematically clamped analysis is a good starting point, however, future design attempts should make use of a more sophisticated actuator model that includes controller effort and power electronics losses. The process of disqualifying designs based on behavior that fails search constraints may be limiting when controller effort could allow generally better agreed behavior across the wider trajectory with torque, velocity or motor current and voltage saturation at only a few data points. Perhaps using an $R^2$ constraint with an applied motor saturation would provide a wider range of design options and resulting performance. Further, inclusion of a dynamic system model with control effort could potentially give better understanding of final system response. Finally, inclusion of more motor and electric dynamics is necessary. Modeling of motor commutation and switching frequency losses at the MOSFETs, battery resistive losses, battery discharge capacity, and capacitor resistive losses all would help increase fidelity of the electric energetic analyses, and may even sway architecture choices.

Despite the discrepancies between design objective and final results, the TF8 still outperforms all other published research-grade actuators in terms of mass, peak-torque, joint-power, range-of-motion, velocity, and energy efficiency. Compared to the commercial emPower, the system as built underperforms at high power. This is due to the discussed power electronics limitations of high current ripple, and ineffective torque generation from applied phase-current. When the new motor driver and embedded platform are operational the system is sure to show much higher power capabilities that outperform all known systems. For optimal energetic performance small modification to the architecture can be done to tune the joint to unique applications; swapping the ballscrew to a lead that more closely matches the application would more fully utilize the motor power range and energetic efficiency. The TF8 architecture as-built shows it is possible to achieve the desired performance metrics of knees and ankles while also reducing the component count of a powered prosthesis to just a handful of machined component. Reducing the complexity of managing different hardware configurations seems a worthy design effort for the research environment, as well as a commercial product line.

The design of TF8 attempts to not only build a high performance, cost-effective actuator capable of performing multiple functionalities, with high electric efficiency, it also strives to include a bold aesthetic statement for the design of hardware. At the end of the day we are building hardware to replace the function of lost body-parts. Those people who have the opportunity to make use of this hardware should
be inspired not only by the technology but by our effort to push the limits of how disability is perceived.
Parting Thoughts

Finally, I will leave the reader with some thoughts that I find to be true when building hardware:

- Never trust a robot; keep your fingers clear; at prototype level a loose wire can cause disaster.

- Design the hard-stops; the robot will always find the hard-stops, it is up to you to determine if they are a problem or not.

- Before you get into controls and walking, focus on safety features in hardware and software; whenever possible, find elegant means to handle unknown conditions in an attempt to prevent surprisingly ugly results.

- Always use an E-stop during low-level testing.

- Always put away the tools and replace broken bits; you never know who will need that part when there is no time to spare.

- Maintain your own set of hex-keys and digital calipers.

- Keep a minimum safety stock of all components so that you never run out of what you need; never let a two cent screw set a project back a week.

- Build a build-kit before you start to assemble your hardware.

- Two of the hardest parts of building robots are wiring and managing manufacturers.

- Never underestimate the value of relationships when building hardware. Your suppliers are crucial to the success of your designs.

- It is usually better to send out parts to the professionals so you can spend your time on other critical items, but, sometimes you have got to cut chips yourself to beat the deadlines and make things work.
Bibliography


[28] Fillauer, MotionFoot © MX.


