

An autonomous lightweight actuated orthosis to support ankle plantar flexion

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Master of Science Thesis



AN AUTONOMOUS LIGHTWEIGHT ACTUATED ORTHOSIS TO SUPPORT ANKLE PLANTAR FLEXION

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Abstract

Walking – a process that happens automatically for most of us – is one of the biggest daily challenges for patients whose legs are impaired by health conditions like stroke, multiple sclerosis, or trauma. Even moderate impairments can cause decreased stability and mobility because of weakened muscles, spasticity, and lack of sensory feedback. The traditional solution is to add stiffness around a joint or lock a joint completely at a certain moment in the gait cycle with a passive structure called an orthosis. Although these kinds of solutions have proved to help patients increase their mobility, the extent to which they enable patients to participate in daily life situations that require mobility is limited.

This thesis aimed to design and realize an autonomous actuated ankle foot orthosis that – by injecting power around the ankle joint – can further increase mobility of patients. Ankle plantar flexion is a good target for adding this extra support because of the isolated boost of power that needs to be generated during push-off which patients often lack. The conceptual phase resulted in a mechanical design that uses an electric motor and ball-screw gear to create a linear actuator that exerts ankle power via a linkage mechanism. A spring was implemented between the actuator and load, by making the output linkage flexible (a leaf spring).

Adding actuation to an orthosis will inevitably increase its weight, and increased weight can potentially diminish the net effect of the device because the user has to accelerate the mass of the orthosis. To prevent this, the approach of this thesis was to find the optimal actuation system, which accounts for a significant part of the total weight of the orthosis. This was done by creating a dynamic model of the motor and gear and use it to optimize the parameters of the linkage mechanism and spring for the amount of push-off power supplied to the ankle. Doing this for all combinations of pre-selected motors and gears, the optimal drive components could be found by comparing the support to weight ratio of those combinations.

Given the optimized actuation system, the detailed designs of all other systems were created, prepared for manufacturing, produced, and assembled i.e.:

- A framework for real-time torque control which is responsible for regulating the actuator's behaviour and harnessing the energy necessary to feed the actuators.
- The sensors which feed back the state of the actuation system to the controller and all wiring needed to route the signals properly.
- The mechanical structure and human interface which connects the human foot and lower leg to the actuator. The mechanical structure transfers forces from the actuator to the human limbs in a safe and comfortable way.

The result was the fully functional device shown in fig. 1. Doing walking tests with humans was beyond the scope of this thesis, but preliminary tests to assess the device's performance showed that:

- The mass of the orthosis is 1.5 *kg* and mass of the backpack is 5.2 *kg*.
- The backpack contains a functional joint torque controller.
- The device is capable of autonomous operation.

With this set of specifications our ankle-foot orthosis is powered, autonomous and the lightest currently build. We hope that further research with our orthosis will increase the quality of live of patients currently living with a walking impairment.



Figure 1: Photo of the author wearing the powered ankle foot orthosis

Contents

1	Introduction	7
	1.1. Assumptions and Conventions	8
	1.2. Background	9
	1.3. Project Goals	10
	1.4. State of the art	12
	1.5. Design criteria and main requirements	13
	1.6. Conclusion	14
2	Conceptual Design	15
	2.1. Introduction	16
	2.2. Functional analysis	16
	2.3. Design choices	17
	2.4. Conclusion	21
3	Simulation	23
	3.1. Introduction	24
	3.2. Method	24
	3.3. Results	31
	3.4. Discussion	33
	3.5. Conclusion	34
4	Detail design	35
	4.1. Introduction	36
	4.2. Main structure and interface	36
	4.3. Linear actuator	38
	4.4. Lever arm	39
	4.5. Ankle hinge	44
	4.6. Conclusion	47

5.1. Introduction	50
5.2. Control Hardware	51
5.3. Control Software	55
5.4. Conclusion	58

6.1. Introduction	60
6.2. Manufacturing and Assembly	60
6.3. Experiments	62
6.4. Conclusion	68

A.1. Literature Survey	81
A.2. Parts list	81
A.3. Technical Drawings	81
A.4. Photos and videos	81

Introduction

1.1. Assumptions and Conventions	8
1.2. Background	9
1.3. Project Goals	10
1.4. State of the art	12
1.5. Design criteria and main requirements	13
1.6. Conclusion	14

This report describes the design, manufacturing and assessment of a state of the art orthotic device that supports lower extremities. To ensure that this project will provide meaningful contribution, it is important to address the following questions: Who are your customers? Why would they need such a device? What does it add to the devices that already exist?

The title of this report already reveals the purpose of the device, but at the start of the project, this was still open. In this chapter will answer the aforementioned questions and, in that process, derive a clear-cut design goal, the required product functions and design criteria to assess potential solutions.

1.1 Assumptions and Conventions

This report includes an analysis of human walking from a mechanical perspective. Human walking arises from a series of complex movements of the legs also referred to as the human gait cycle. To simplify the analysis in this report, walking will be considered as a motion in sagittal plane (see fig. 1.1), thus disregards all rotations in other planes. Although non-sagittal movement is important for maintaining stability and taking turns during walking, the main sources of propulsion come from sagittal motion.

In gait analysis, time is indicated in percentage of stride – the period of one complete cycle of the leg shown in fig. 1.2 – which starts at first contact of the foot with the ground (heel strike). Heel strike initiates the stance phase which can be sub-divided in loading response, mid stance and push off. After push-off, the leg swings forward ending the cycle when the foot contacts the ground the second time.

The rotations of the human body segments in the sagittal plane are expressed according to the sign and naming conventions in fig. 1.1. The joint angles q_{ji} and torques T_{ji} (for $i = 1, 2, 3$) are taken positive in counter-clockwise direction (angles are zero in standard anatomical position). The leg motions q_{ji} are also referred to with their medical terms: flexion/extension for the hip and knee and plantar/dorsal flexion for the ankle. The joint rotation $q_{ji}(t)$ and joint torque $T_{ji}(t)$ that are used in this report are filtered and averaged gait data that were obtained from Winter [1]. This data is from subjects walking at a normal speed of $\approx 1.2m/s$. Assuming that the wearer will exhibit a normal walking pattern with the orthosis, might not be realistic – especially for the future plans to test with patients. For the purpose of this optimization, however, the power levels and range of motion will be similar for moderate deviations from normal gait.

This chapter also addresses gait of people with a walking impairment. Most information on this subject originated from a literature survey done by the author [2]. The study included characterizing walking impairments, looking at their effects on the gait cycle and finally formulating recommendations for future orthotic devices. The information essential for this thesis is summed up in the following section. When additional information on this subject is desired, the full survey can be found in appendix A.1.

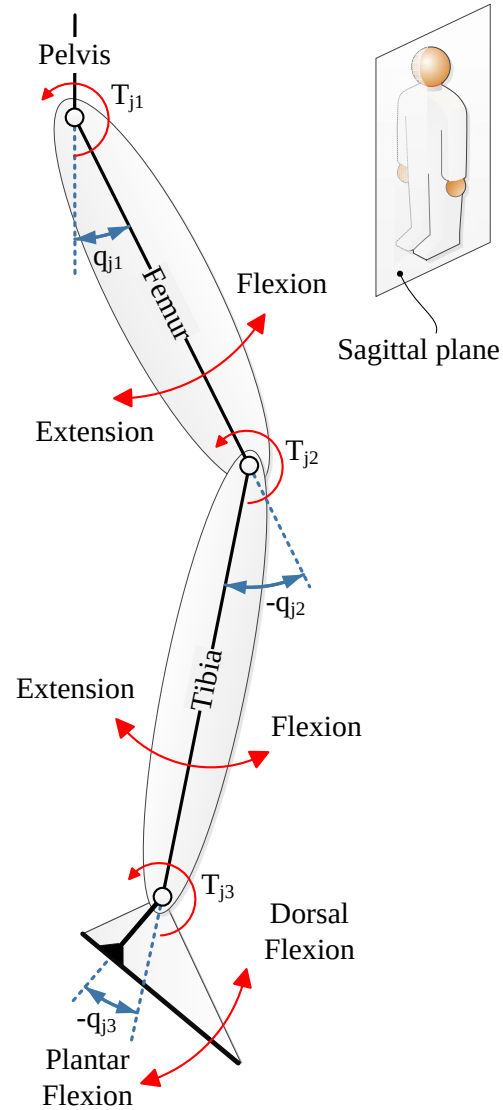


Figure 1.1: Representation of the leg in the sagittal plane including the naming and sign convention

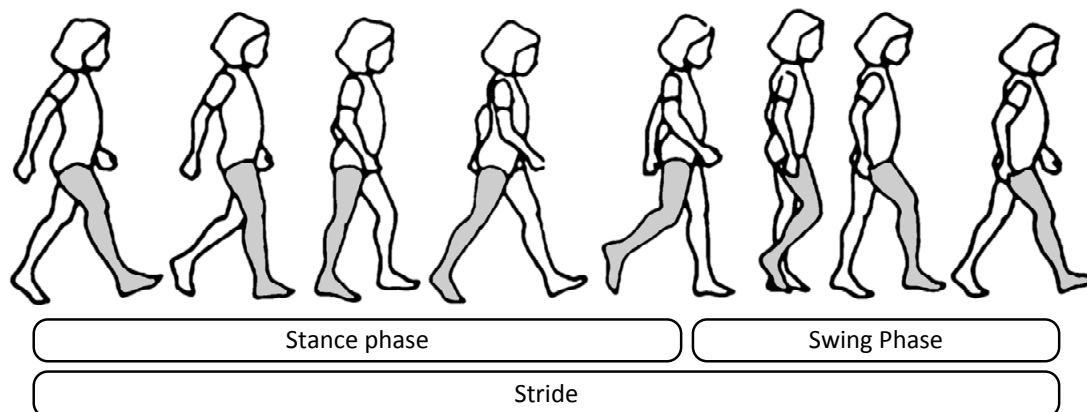


Figure 1.2: Normal gait cycle [3]

1.2 Background

Yearly many people suffer from health conditions that lead to walking impairments. These conditions include: stroke [4], neurological disease (e.g. multiple sclerosis [5]), muscle disease (e.g. Post-polio syndrome [6]) and trauma (e.g. spinal cord injuries [7]). From the mechanical point of view, health conditions can compromise body structures and functions in the following ways [2]:

- Muscle weaknesses or lack of muscle control.
- Lack of proprioception (sensory feedback of forces and motion of the body).
- Abnormal stiffness of the joint (spasticity).

These impairments manifest themselves on different regions of the leg and different degrees of symmetry for each patient. This means that every patient – while having the same illness – has a unique deviation from normal gait, which is often worsened by a variety of compensation tactics.

The most common countermeasures for these functional deficiencies are passive structures around ankle – ankle foot orthoses (AFO) – and/or knee joint – knee orthoses (KO) shown in fig. 1.3, that stabilize those joints during the entire or part of a stride.

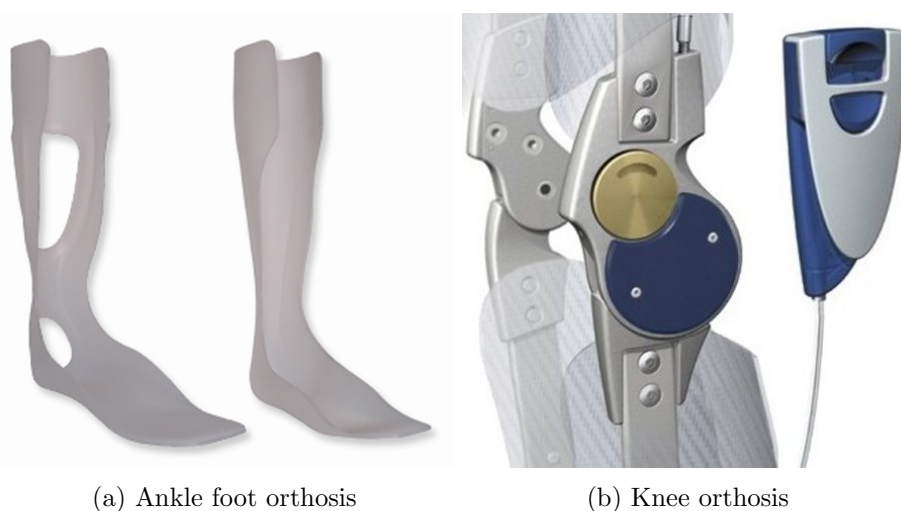


Figure 1.3: Orthotic devices [8]

These products are known to increase walking stability and share some of the load on the muscles. Research done by Bregman [9] suggests that these solutions do not enhance impaired gait e.g. increasing push-off power of the ankle. This limits the amount of mobility of a patient, hence his/her ability to participate in activities that require long walking endurance (e.g. walks in nature and professions where mobility is required). Recent developments in commercial orthotic devices focus on integration of electronics in the orthoses (e.g. Walkaid from Innovative Neuronics [10] and the C-Brace by Otto Bock [11]), however, none of these devices inject power.

If we want to further increase the mobility of patients, the possibilities of powered devices need further exploration.

1.3 Project Goals

The implications of impairments to the lower extremities greatly vary depending on the type and severity of the health condition. The target group for this project is people with moderate impairments, which is the biggest group of patients [2]. This means they are able to walk, but aren't able to do so in an efficient manner because of the aforementioned weaknesses, spasms or sensory issues in their legs. This group of patients could greatly benefit from a device that supports those weaknesses by injecting energy.

The final goal for this project: Creating an autonomous lightweight actuated orthosis that can enhance impaired gait and reduces metabolic cost of transport. In order to do so, the following sub-goals have to be reached:

1. Design of a lightweight actuated orthosis,
2. Manufacturing and assembling the design,
3. Designing and implementing a control algorithm,
4. Functional testing of the prototype,
5. Testing the prototype on healthy subjects,
6. Testing the prototype on patients with a moderate walking impairment.

The goals for this MsC report, do not fully encapsulate the goals mentioned above. To set realistic goals, considering constraints from time and resources, the first four sub-goals will be the minimal deliverables for this thesis.

The question that is answered at the start of the project is: what joint should be supported by the device. A good exploratory tool used for this purpose is data from sagittal plane gait analysis performed by Winter [1] shown in fig. 1.4. It shows the average joint angles, external torques and powers normalized to bodyweight for hip flexion/extension, knee flexion/extension and ankle plantar and dorsal flexion. Examining these plots, different types functions can be identified where the muscle are absorbing or injecting energy in a specific directions. A schematic breakdown of those functions in time is shown in fig. 1.5. At the ankle, we see a nicely isolated sequence of absorption in dorsalflexion direction and a large power injection peak in plantarflexion direction. This injection phase is called push-off and is one of the main sources of propulsion during walking. For a large group of patients [2], push-off is inadequate because of weakened calf muscles. Therefore we did decide to create a powered ankle foot orthosis (PAFO). Furthermore, designing a device for the ankle requires an actuation system highly optimized for weight – as a study Browning et al. [12] showed – which fits the strength of department nicely.

In contrast to the long term goal of this device, the first step in this project will be fitting a device on a healthy subject and showing that a net reduction in metabolic

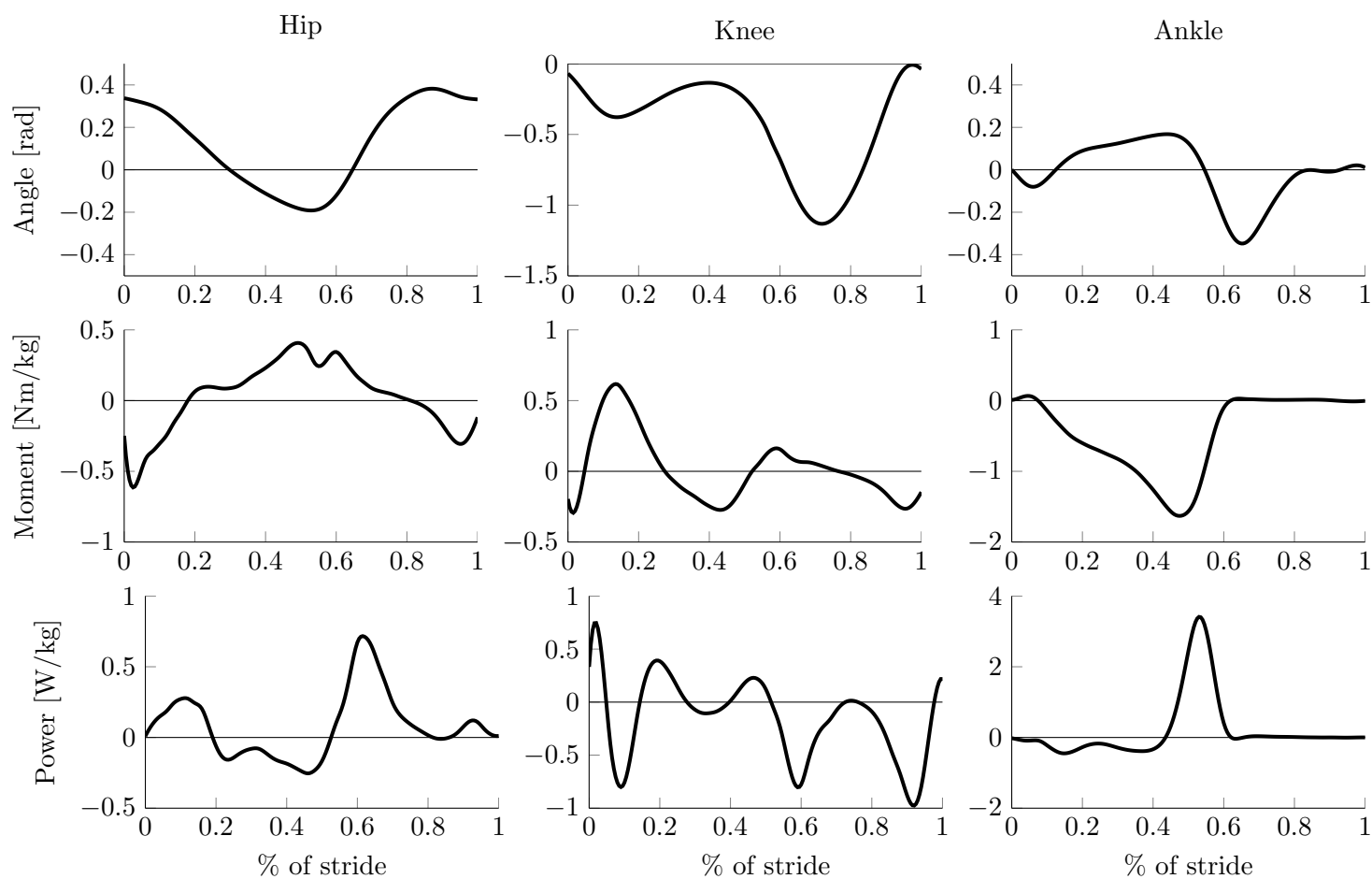


Figure 1.4: Filtered and averaged gait data of healthy people at normal walking speed (1.2 [m/s]) in the sagittal plane.

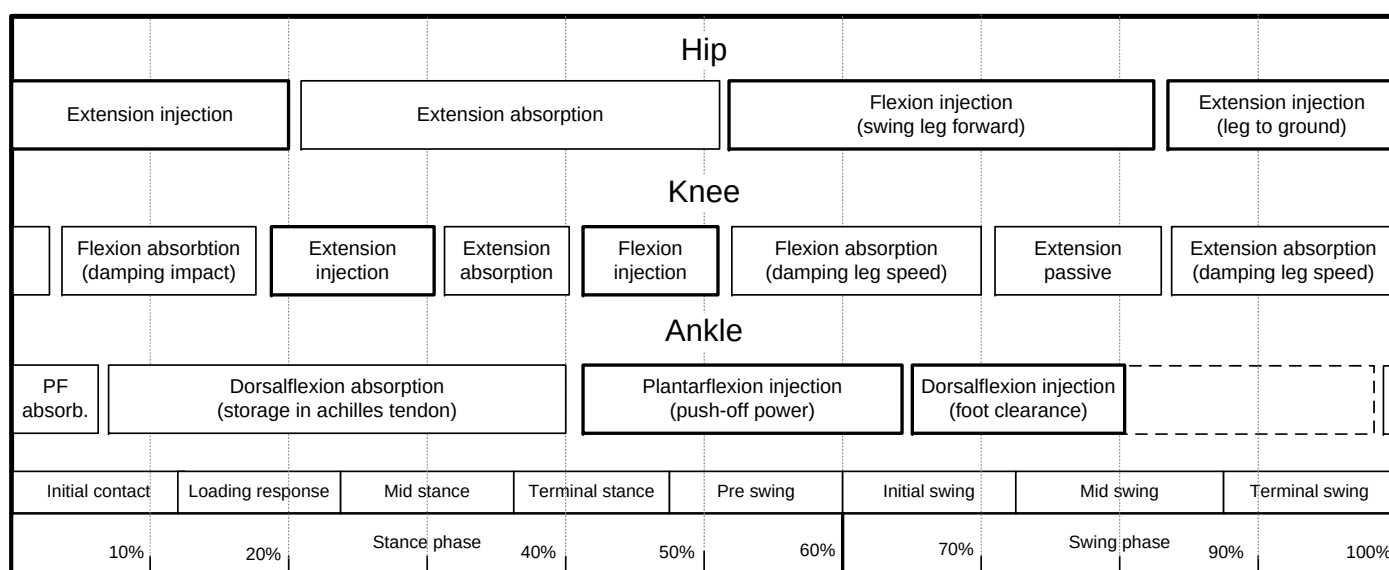


Figure 1.5: Sagittal plane joint functions in time

cost of transport can be achieved. As will be shown later, doing so will greatly ease the challenge of having patients with walking impairments benefit from such a device.

1.4 State of the art

Several researchers have put effort into creating and testing orthotic devices for the ankle. Figure 1.6 shows the devices capable of enhancing human walking by injecting plantarflexion energy.

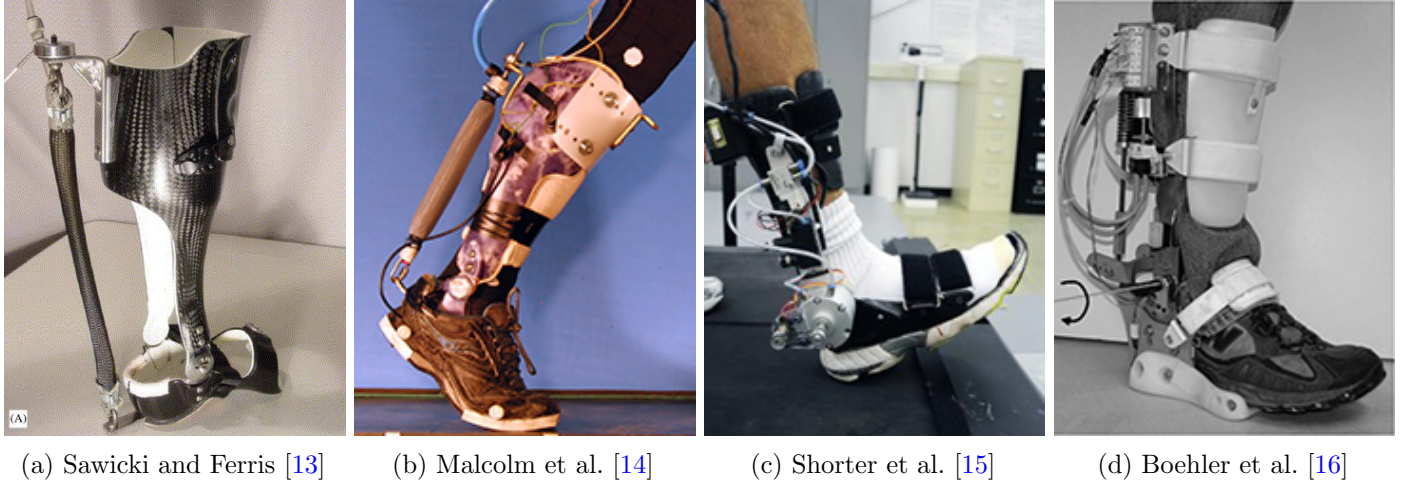


Figure 1.6: State of the art

The devices shown in fig. 1.6a and fig. 1.6b with a weight of 1.7 and 0.7 *kg* both work with a pneumatic muscle, powered from a fixed power source. The latter shows that, when applying properly timed power bursts, PAFOs can reduce the net metabolic cost of transport. When the goal is to impact patients daily lives, however, the step to creating autonomous devices is inevitable. Figure 1.6c shows a device of 1.9 kilograms that uses a pneumatic rotary motor and enables portability with a gas cylinder. The efficiency of such a pneumatic system is very limited [ref] and with rotary motors the mass is largely located at ankle height causing high rotary inertia w.r.t. the knee. Wiggin et al. [17] showed how devices can utilize passive components like springs to store energy from the absorption phase and reuse this energy during push-off. The 1.7 kilogram autonomous device in fig. 1.6d uses a combination of a spring and electric motor – along with optimization of the geometric parameters and spring stiffness of the device – to can provide support up to 40%. This method is called series elastic actuation and can greatly increase performance of a PAFO as will be discussed later in this report.

For the device described in this report, we intent to reach an even higher performance with two innovative approaches:

1. Usually a motor and gearbox is selected based on an initial estimation and then the system is optimized. Our approach is to use the properties of multiple of pre-selected a motors and gears in a dynamic model, optimize each combination for support, and choosing based on the trade-off between support and weight. This way, the properties and limitations of the drive combination are integrated in the optimization which leads to a increased total performance.
2. When including a series elastic element in the design, traditional springs (e.g. helical or spiral springs) are often considered. These kinds of components need

a lot of additional structure for proper support and guidance. Our design will integrate the series elasticity by making structures – that are needed anyway – compliant, and therefore omit additional weight.

Realizing and assessing a lighter and more powerful prototype will increase the group of patients that can benefit from PAFOs and challenge the possibilities of current day orthotic solutions.

1.5 Design criteria and main requirements

Given the background, design goals and competition; a framework of criteria can be set that will help with judging (and comparing) design concepts. The following criteria are relevant for the current design:

- **Inertia:** Inertial properties (mass and moment of inertia) of the device.
- **Comfort:** Wearing comfort for the user.
- **Cosmetics:** Aesthetics of the device.
- **Controllability:** Effort needed to properly control the device.
- **Safety:** Physical safety of the user.
- **Adjustment range:** Variety of people that can fit the device.
- **Structural integrity:** Strength and stiffness of the devices' structures.
- **Portability:** Mobility of the device.
- **Performance:** Amount of plantarflexion support.
- **Realizability:** Effort, cost and time needed to realize all components.

Some of these criteria yield one or more aspects that are quantifiable, also known as requirements. Most of these requirements will be found and revealed in subsequent chapters, but the most fundamental ones known from the start are listed below:

- A study by Meuleman et al. [18] shows that when a mass up to 2 *kg* is added to the foot or 6 *kg* to the pelvis, subjects show significant gait adaptation. However this research also states that: “these changes were all within the normal inter subject variability we considered these changes as negligible for application as rehabilitation robotics and assistive devices”. The prototype should be design well below this boundary.
- The first prototype will be suited for subjects with a mass up to 80 *kg*. The length of the subjects can range from 1.7 to 2.0 *m* with shoe size of 41 to 44 (European). This range will include a large group of healthy Dutch students.
- The prototype should facilitate walking speeds up to 1.2 *m/s* (normal speed from gait data).

1.6 Conclusion

This chapter has set the challenge, scope and requirements for the design of a state of the art autonomous powered ankle foot orthosis that supports ankle plantarflexion. This thesis will not only focus on the design, but also realization of the device and assessment of its performance.

First, chapter 2 will discuss how the conceptual design of the PAFO was chosen and motivates those choices. Chapter 3 will describe the optimization of the actuation system. Next, chapter 4 will describe the detailed mechanical design of the whole device. In chapter 5 the focus will be on the hardware needed to control the device and the software implementation and control framework. chapter 6 will describe how the design was manufactured and assembled as-well-as the assessment of the devices' performance. Finally, chapter 7 presents the main conclusion of this thesis. This report will end with a word of thanks to those who helped with this thesis in chapter 8.

The bibliography and a list of symbols can be found in chapter 9 and 10, respectively. The appendices were bundled on a compact disk that is included with this report in appendix A.

2.1. Introduction	.16
2.2. Functional analysis	.16
2.3. Design choices	.17
2.4. Conclusion	.21

This chapter will start with a functional analysis of the design goals. Given these functions that need to be included in the device, we can generate a wide range of possible solutions for each of those functions resulting from brainstorming sessions. Next, combining sub-solutions into concepts and assessing them with the previously set criteria, will result in choosing the most promising and realistic option and set a framework for simulations and detail design.

2.1 Introduction

The design goal formulated in the previous chapter is: Creating a lightweight autonomous actuated ankle foot orthosis that is capable of generating plantarflexion power. As shown in the state of the art section, this design problem can be solved in a lot of different ways. This chapter focusses on formulating a conceptual design that describes the combination of solutions that will be used to achieve the design goal. Because of the extensive experience with designing actuated and orthotic devices ([19], [20], [21], etc.) and the limited time frame, a semi-methodical approach was used to generate and choose concepts i.e. the scope of available solutions was limited to a subset with which the department had good experiences.

2.2 Functional analysis

To find which functions are needed in the device, it is useful to analyse the used scenario. This not only involves looking at the main functionality, but also how the user starts and ends this activity as shown in fig. 2.1. The used scenario also results in additional requirements for the device, like easy doing on and off the device and being able to sit in a chair with the device on. The supported walking block contains the main functionality. The sub-functions of this block were discussed in the previous chapter (see fig. 1.5).

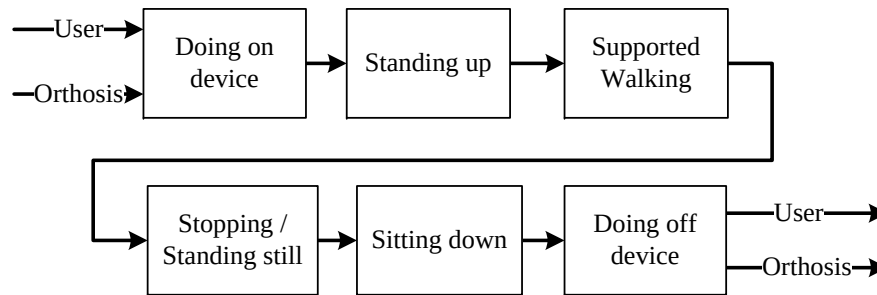


Figure 2.1: User scenario for the orthosis

Next, a first level functional decomposition was done to identify the functionalities the device needs to fulfil its goal. All these sub-functions can be grouped into systems (see fig. 2.2):

- Human interfacing system: In order to provide plantarflexion support for the device, the actuation system has to interact with the human limbs. The human interfacing system is responsible for transferring forces on the lower leg and foot.
- Structural system: This system has to add the necessary constraints to the individual components within and between systems.
- Actuation system: Next to the main function of this system, injecting mechanical power, the actuation system has also to be able to facilitate free motion, temporarily store energy and absorb impacts.
- Communication system: This system has to make sure all signals are properly transferred within and between system.

- Control system: The device needs a system that interprets its state and regulates its behaviour, called the control system. This system also has to provide power to the actuation systems.
- Sensory system: The control system needs information about the state of the device. The sensory system has to provide this information.

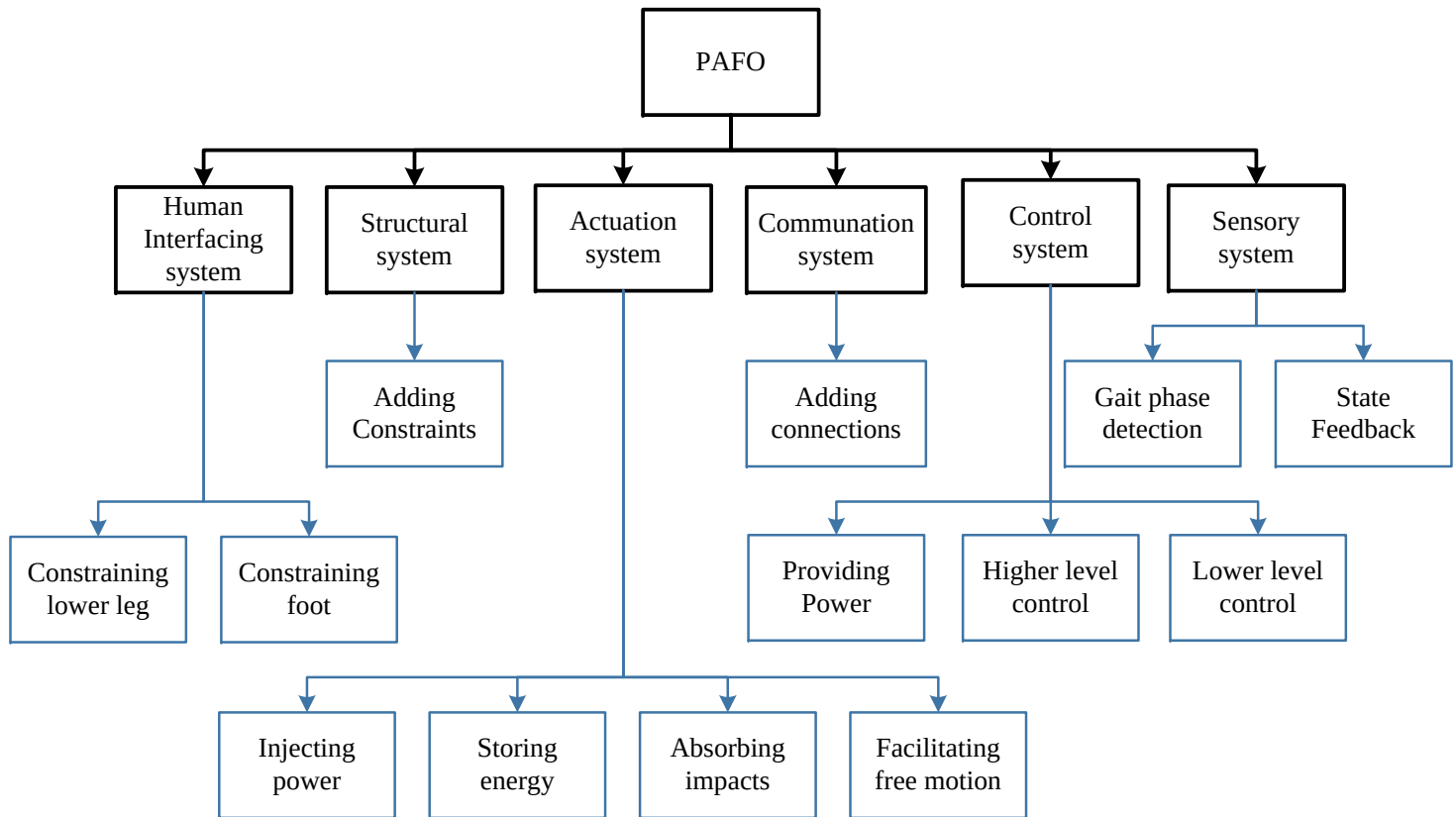


Figure 2.2: Functional decomposition

2.3 Design choices

To fulfil the sub-functions discussed in the previous section, we need sub-solutions. One sub-solution can fulfil multiple functions and one function can require several sub-solutions to be fulfilled. This section discusses the sub-solutions that were considered and motivates the choice for each function based on the design criteria listed in section 1.5. The combination of chosen sub-solutions forms a design concept.

As mentioned before, the design approach is semi-methodical, hence instead of choosing multiple concepts and comparing them, only one concept was chosen. Furthermore, not all sub-functions are approached on a conceptual level. These sub-solutions are discussed in the detail design chapters.

Structural system

This system is largely integrated with the human interfacing and actuation system and will therefore be discussed in those sub-sections.

Human interfacing system

Interfacing with the human limbs usually involved a trade-off between adjustment range, comfort and structural integrity. Table 2.1 shows the solutions chosen for this system (and the structural system).the first decision that needs bo be made is between a single or bilateral hinge of which the latter was chosen. Although this choice can cause a slight decrease in comfort a bi-literal hinge increases the structural integrity. The second choice, involves the solution for contraining the foot and lower leg. A double shell solution was chosen because it combines the comfort of ergonomic shells with the predictability of a hinge.

Adding constraints



Connecting & Constraining foot & lower leg



Table 2.1: Concept choice for the actuation system where the underlined text indicates the chosen solution

Actuation system

The actuation system plays a primary role in the PAFO design and the solutions selected for this system dictate some of the following choices. An overview of the concepts that are considered for the actuation system is shown in table 2.2. The first three rows concern power injection and facilitating free motion functionality. This functionality required an electric motor, and two transmissions (the first increases the output force and the second converts force to joint torque). The last row shows the options to store energy and absorbing impacts (series elasticity).

The options for the electric motor are a selection of standard or outrunning DC motors. The latter show promising results in another project, but was still in the testing phase

Injecting power & facilitating free motion

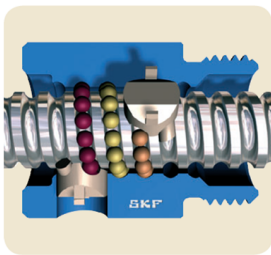


Standard DC motor



Outrunning DC motor

Injecting power & facilitating free motion



Ball-screw gear



Harmonic gear



Maxon planetary gear

Injecting power & facilitating free motion



Bowden cable



Linkage mechanism



Belt drive

Storing energy & absorbing impacts



Die-spring



Machined spring



Composite leaf-spring

Table 2.2: Concept choice for the actuation system where the underlined text indicates the chosen solution

at the time. The former option – the one that is chosen – has more of-the-shelf motor controller available and detailed information on the motors’ parameters is available.

The last three choices are highly interconnected. The combination of ball-spindle, linkage mechanism and composite leaf-spring provide an efficient and lightweight actuation system. The ball-screw gear converts the motor rotations in a linear motion with high efficiency. A linkage mechanism can now convert the force to a joint torque via a lever-arm. This is where the spring can be integrated, replacing the normally rigid lever-arm with a flexible element. Predictions show that composite materials where most likely needed to make the concept feasible.

Control system

The control system yields functionality to control the PAFO on lower and higher levels. The former regulates the input to the electric motor through a local control loop in the firmware. The latter is provides a way of implementing a higher level controller. An overview of the concept choices is shown in table 2.3.

To understand the options and choices of these functions, it is beneficial to understand the chronology of the design process: The concept choice of lower level controller is made after completing the simulations of the actuation system (see chapter 3) which resulted in a motor selection. The motor type and its power requirements narrows down the controller choice to the Maxon EPOS3. The Maxon EPOS3 motor controller communicates via EtherCAT, which again narrows down the options for higher level

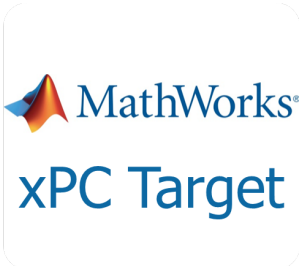
Lower level control		
		
Barrett PUCK	3Mixel	<u>Maxon EPOS3</u>
Higher level control		
		
Etherlab	<u>SOEM + E-box</u>	Mathworks xPC target

Table 2.3: Concept choice for the control system where the underlined text indicates the chosen solution

control to the three options shown in the second row of table 2.3. Contradictory to the mechanical design process, for the higher level control, multiple solutions are worked on in parallel. After testing and comparing, Simple Open EtherCAT Master [37] combined with E-box[38] (for details see chapter 5) is chosen for its runtime interface with Matlab and ease of portability.

The downside of choosing the EPOS3 controllers is their relatively large size and weight. Therefore, the whole control system has to be located in the a backpack. Providing power to the PAFOs is done via battery packs also located in the backpack.

Communication system

The main communication method is dictated by the motor controller choice, namely EtherCAT. Other input and output signals depends on the sensor choices described later.

Sensory system

The solutions for the sensory system are chosen pragmatically depending on the requirements that originated from the actuation design.

2.4 Conclusion

This chapter gives a brief overview of the functions needed for the PAFO and how these functions are grouped into systems. Next, the sub-solutions that can fulfil these functions and the choices that led to a final design concept is discussed.

In the following chapters, there will be a clear distinction between systems that will be designed with a pragmatistical approach (control, sensors and communication) and systems that will be thoroughly analysed and optimized (actuation, human interface and structure).

3.1. Introduction	.24
3.2. Method	.24
3.3. Results	.31
3.4. Discussion	.33
3.5. Conclusion	.34

This chapter will focus on optimizing the components of the actuation system that were chosen in previous chapter, hence an electric motor combined with ball screw gearing in series with a flexible lever-arm. The model will be used to optimize the amount of support that the actuation system can generate around the ankle. The optimization is based on the assumption that normal gait kinematics are preserved while walking with the device. The optimization is performed for each combination of the selected motors and ball screws.

3.1 Introduction

Selecting optimal drive components for a device that has to be attached to the human limbs – especially as proximal as the ankle – requires a good trade-off between performance and weight. The work of Wang [22], introduces a quasi-dynamic model of a motor, gearing and spring to optimize configurations on energy efficiency and compare their total mass. In Wang’s case the goal was to design an orthosis that provides enough power to support a fully paralysed user and device itself. In the current study, the human can generate (at least some) power around the ankle, hence partial support will suffice. In this design case, the main goal is to get a net benefit in energy expenditure when wearing the device, hence some support can be sacrificed in favour of being able to make a lighter orthosis which minimizing the users effort in dragging the device along. To find the combination of drive components (i.e. motor and gearbox) that give the optimal results for our application, we will introduce a dynamic model of the actuation system shown in fig. 3.1 and optimize the system for each combination of selected motors and gearboxes.

The general concept of the actuation system is already narrowed down to an electric rotary motor driving a ball-screw spindle. To transfer the actuator power to ankle power, the linear actuator is suspended between the shank and a flexible lever arm attached to the foot, which means it also functions as series elastic element. As is shown by Boehler et al. [16], integrating this series elasticity is not only an effective way of implementing torque/stiffness control, but can also significantly increase the amount of support power during push-off. Additionally, the spring reduces impacts between the actuation system and the users ankle.

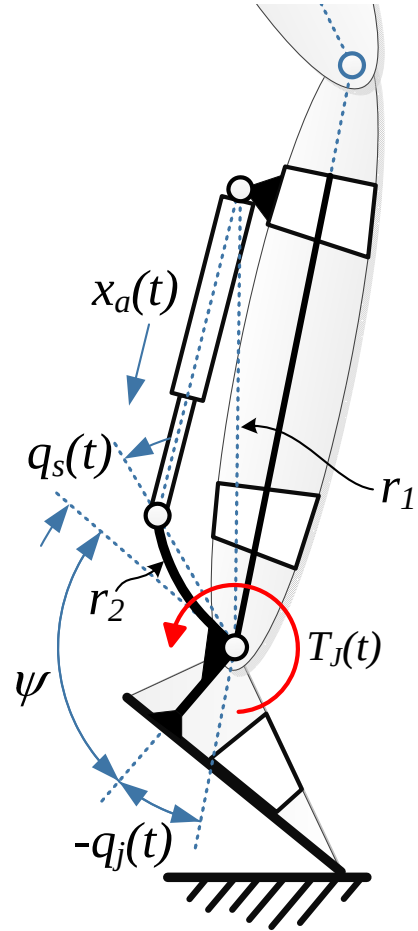


Figure 3.1: Actuation system, where the variables are: $q_j(t)$ the relative joint angle, $T_j(t)$ the external joint torque (positive in dorsiflexion direction), $q_s(t)$ the spring deflection angle, and $x_a(t)$ the actuator stroke. The geometric parameters: The initial lever-arm angle ψ , and lever-arm lengths r_1 and r_2 .

3.2 Method

The actuation support system depicted in fig. 3.1 is modelled as shown in the block diagram in fig. 3.2. The diagram introduces the inverse dynamic model of the drive components, linkage mechanism, load sharing function, and motor and gear constraints. The inputs for the model are: The translation function $x_a(t)$ ¹ – one of the optimization parameters, and the external joint angle $q_j(t)$ and torque $T_j(t)$ which are obtained

¹The subscripts used for parameters and variables denote the location in the actuation system, namely a=actuator, m=motor, s=spring, g=gear, and j=joint

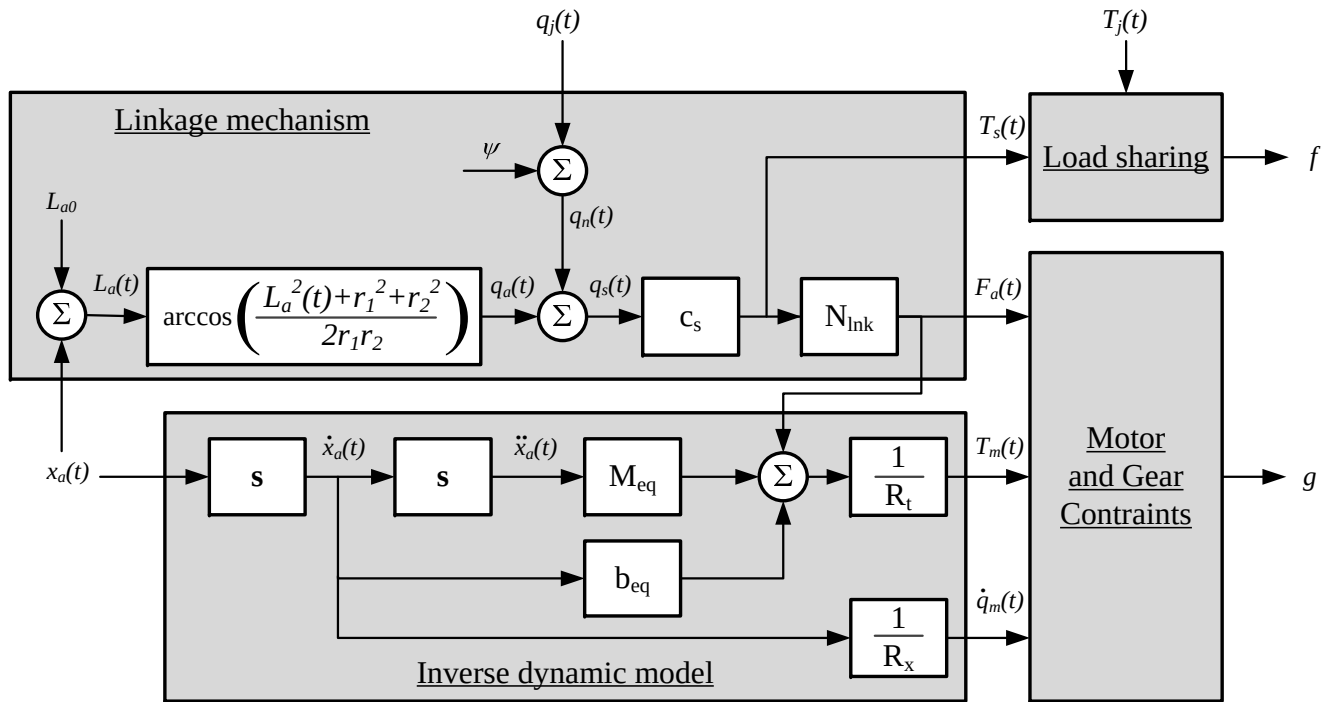


Figure 3.2: Block diagram of the mathematical model that describes the PAFO and its interaction with the user.

from gait data. The model outputs are: The support ratio f which is a measure of the residual power the user has to exert and the drive component constraints g which describe the limitations of the motor and ball-screw gear. Given these outputs, the support optimization can be written as the following minimization problem

$$\min_{\bar{z}} f(\bar{z}) \quad \text{s.t.} \quad g(\bar{z}) \leq 0 \quad (3.1)$$

where

$$\bar{z} = \{\bar{x}_m, r_1, r_2, \psi, c_s\} \quad (3.2)$$

are the chosen optimization parameters:

- The actuation parameter $\bar{x}_a$ is a vector of n target positions of the actuator, equally distributed over the one gait cycle. The actual position profile $x_a(t)$ is constructed by re-sampling $\bar{x}_m$ and applying an 3rd order Butterworth filter with a cut-off frequency of 0.3π radians per sample. The amount of points n is set to 16 which could construct a detailed enough function without significantly slowing down the optimization. For the travel, the actuator stroke $x_a(t)$ equals the motor stroke $x_m(t)$ (this is not the case for force).
- The behaviour of the linkage mechanism is determined by the parameters r_1 , r_2 , ψ and c_s which define the lengths from ankle joint to the top and bottom connections of actuator, the angle between ankle joint and lever arm (see fig. 3.1), and the torsional stiffness of the flexible lever arm with respect to the joint centre, respectively.

All other parameters of the actuation system shown in the diagram are given by the motor and gearbox specifications. The remainder of this section will focus on each of the individual blocks that will be used in a non-linear constrained optimization.

Inverse Dynamic model

As mentioned before, the actuator contains a rotational and translational system. The motor speed and torque can be converted to motion and force by means of the gear ratios

$$R_x = \frac{p_g}{2\pi} \quad \text{and} \quad R_t = \frac{2\pi\eta_g}{p_g} \quad (3.3)$$

respectively, where p_g is the pitch of the ball-screw spindle and η_g is the spindle efficiency. Using these conversions, the system can be represented by a simple translational inverse dynamic mass-spring-damper model, where the lumped mass of all drive components M_{eq} undergoes a linear translation $x_a(t)$ and is subject to external forces from friction F_d , the motor magnets F_m and interaction with the series spring F_a . The corresponding dynamic equilibrium equation of the system is

$$M_{eq}\ddot{x}_a = \Sigma F = F_m + F_a - F_d \quad (3.4)$$

where M_{eq} the equivalent mass in $[kg]$ is

$$M_{eq} = \frac{R_t}{R_x} (J_m + J_g) + m_{g1} \quad (3.5)$$

Where J_m and J_g are the inertias of motor and spindle, and m_{g1} the mass of the nut. Note that the spindle pitch quadratically decreases the inertial forces of the motor and spindle. The force from the motor magnets is

$$F_m = R_t T_m(t) \quad (3.6)$$

where $T_m(t)$ is the required motor torque. The damping force is

$$F_d = b_{eq}\dot{x}_a \quad (3.7)$$

where b_{eq} is the lumped damping coefficient, predominantly from bearing friction.

The interaction force with the spring F_a is obtained from a model of the linkage mechanism described in the next section.

Linkage kinematics

The series spring is modelled as a simple bending beam shown in fig. 3.3a. For $q_s < 20^\circ$ we will assume $r_2 = r'_2$ and $x_s = r_2 \cdot q_s$, which is a reasonable assumption of the deflection angles remain small. Now, the linkage mechanisms shown in fig. 3.3b can be identified which yields two triangles. The first is defined by the length of the actuator. The length of the actuator $L_a(t)$ is defined by the initial actuator length plus the actuator stroke function $x_a(t)$, hence

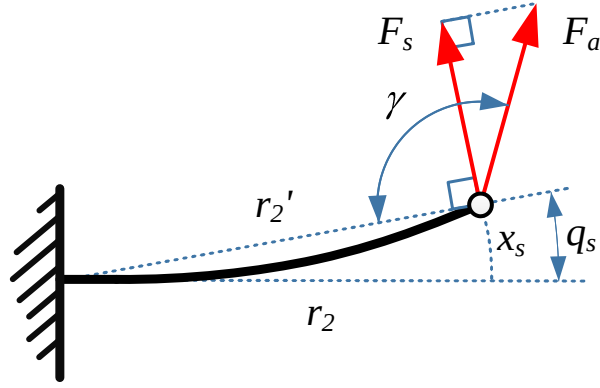
$$L_a(t) = L_0 + x_a(t) \quad (3.8)$$

where

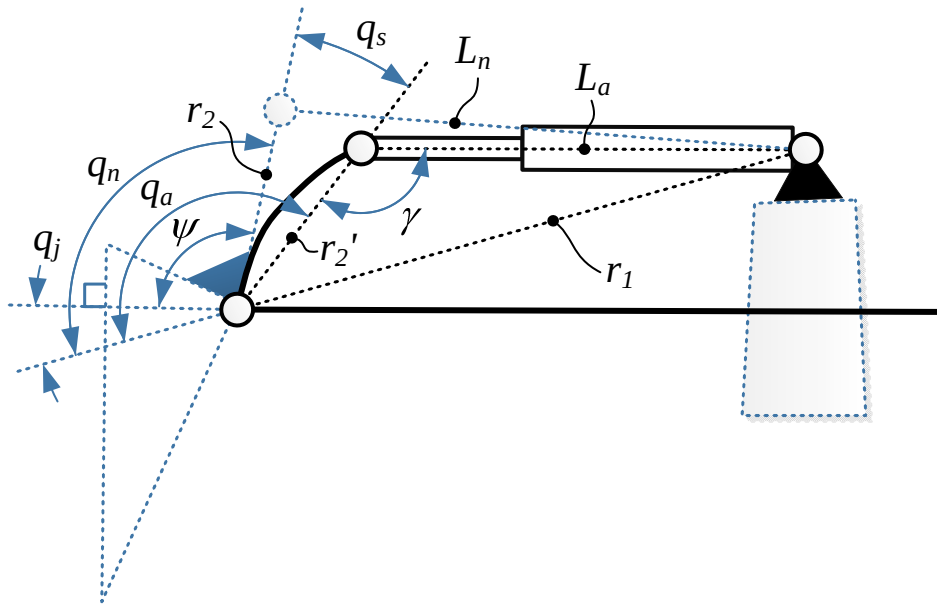
$$L_0 = \sqrt{r_1^2 + r_2^2 + 2r_1r_2 \cos(\psi)} \quad (3.9)$$

is the length of the actuator at $q_j(t) = 0$ without any spring deflection. Given the length of the actuator $L_a(t)$, the angle of the deformed lever is

$$q_a(t) = \arccos \left(-\frac{r_1^2 + r_2^2 - L_a^2(t)}{2r_1r_2} \right) \quad (3.10)$$



(a) Leaf spring



(b) Linkage mechanism

Figure 3.3: Model of the series elastic spring in a three bar linkage

The second triangle is defined the ankle motion which changes the neutral line of the spring

$$q_n(t) = \psi + q_j(t) \quad (3.11)$$

the actuator length needed to track this neutral spring position is

$$L_n(t) = \sqrt{r_1^2 + r_2^2 + 2r_1r_2 \cos(q_n(t))} \quad (3.12)$$

and corresponding neutral stroke $x_n(t) = L_n(t) - L_0$ which will be usefull variables for implementing zero impedance control. The deflection angle of the spring is obtained by

$$q_s(t) = q_a(t) - q_n(t) \quad (3.13)$$

the deformed spring exerts a torque around the ankle

$$T_s(t) = c_s \cdot q_s(t) \quad (3.14)$$

Where c_s is the torsional stiffness that the spring exhibits around the ankle joint. The actuator force F_a needed to deform the spring depends on lever arm length r_2 and angle $\gamma(t)$

$$F_a(t) = T_s(t) \frac{1}{r_2 \sin(\gamma(t))} = T_s(t) N_{lnk} \quad (3.15)$$

The linkage analyses outputs the support torque T_s for the load sharing equation and the resulting force F_a needed in the constraint equations in the next section.

Motor and gear constraints

Substitution of eq. (3.6) - eq. (3.15) in eq. (3.4) and rearranging gives the required motor torque

$$T_m(t) = \frac{1}{R_t} (M_{eq} \ddot{x}_a(t) + b_d \dot{x}_a(t) + F_a(t)) \quad (3.16)$$

In order to reach the torque, velocity and thus mechanical power required for providing support, the motor needs a current I_m , voltage U_m and electrical power P_{el} , which are linked through the equation below. Since the motor torque is proportional to the current

$$I_m(t) = \frac{T_m(t)}{K_t} \quad (3.17)$$

where K_t is the torque constant of the motor. The resulting voltage over the windings is

$$U_m(t) = R_m I_m(t) + \dot{q}_m(t) K_t \quad (3.18)$$

where R_m is the electrical resistance of the winding in Ω and $\dot{q}_m$ the motor speed in rad/s . The electrical power is

$$P_{el} = I_m(t) U_m(t) = \dot{q}_m T_m + I_m^2 R_m = P_{mech} + P_{cu} \quad (3.19)$$

Note that the first term equals the mechanical power P_{mech} and the second term is referred to as the copper losses P_{cu} in W . In efficient motors, these losses are minimal. This is why a the *motor constant* defined by

$$K_m = \frac{T_m}{\sqrt{P_{cu}}} = \frac{K_t}{\sqrt{R_m}} \quad (3.20)$$

is often used as a figure of merit to compare the motor efficiencies.

Now all mechanical and electrical loads are known, the limitations of the drive components can be mapped to inequality **constraints** for the optimization in the form

$$g(\bar{z}) \leq 0 \quad (3.21)$$

hence

$$g(x) = \begin{cases} \max(I_m(t)) - I_{max} & \text{Motor current} \\ \max(U_m(t)) - U_{max} & \text{Motor voltage} \\ \max(P_{el}(t)) - P_{max} & \text{Motor electrical power} \\ \max(\dot{q}_m(t)) - \omega_{max} & \text{Motor speed} \\ \max(F_a(t)) - F_{int} & \text{Spindle force} \\ \max(\dot{x}_a) - v_{max} & \text{Spindle velocity} \\ \max(\Delta L_a(t), \Delta L_n(t)) - L_{max} & \text{Spindle stroke} \end{cases} \quad (3.22)$$

the maximal allowable intermittent current I_{max} recommended by Maxon is

$$I_{max} = I_{nom} \sqrt{\frac{t_{cycle}}{t_{on}}} \quad (3.23)$$

where t_{on} is the time for $I_m > I_{nom}$ and t_{cycle} the process cycle. The total stroke of the actuator $\Delta L_i(t) = \max(L_i(t)) - \min(L_i(t))$ was limited for the practical reason that if the actuator is too long, it will not fit behind the users leg.

Load sharing

In the previous analysis we obtained the support torque $T_s(t)$ (eq. (3.14)) exerted around the ankle. Under the assumption that all support torque will linearly reduce the torque exerted by the user, the support ratio – the cost function of the optimization – can be formulated in either torque or power

$$f = \frac{\sum |T_j - T_s|}{\sum |T_j|} \quad \text{or} \quad f = \frac{\sum |P_j - P_s|}{\sum |P_j|} \quad (3.24)$$

The main goal of the device is to support the user at push-off. The support function that best describes this behaviour is an adapted version of the support power function

$$f = \frac{\sum |\max(P_j, 0) - P_s|}{\sum \max(P_j, 0)} \in [0, 1] \quad (3.25)$$

which rewards generation of positive power around the ankle thus push-off.

Additional metrics

The second important metric for making the trade-off was total mass of the assembly, which will make up a large part of the total weight of the orthosis.

$$m_{tot} = m_m + m_{g1} + m_{g2} \quad (3.26)$$

where m_m and m_{g1} are the mass of the motor and nut, and the mass of the spindle can be found by multiplying the specific mass of the spindle with the maximal stroke of the actuator

$$m_{g2} = \tilde{m}_{g2} \max(L_a(t), L_n(t)) - \min(L_a(t), L_n(t)) \quad (3.27)$$

A secondary metric is how much electrical energy was converted to mechanical energy compared to the copper losses in $[J]$

$$E_{mech} = \int_{t=0}^{t=t_{cycle}} P_{mech} dt \quad \text{and} \quad E_{cu} = \int_{t=0}^{t=t_{cycle}} P_{cu} dt \quad (3.28)$$

which sums up to the energy needed for one step with one leg. Given the desired battery life and the energy density of the battery, the required capacity and weight of the battery can be calculated. Since the battery will be located in the backpack, the battery weight will not be included in the trade-off.

Numerical implementation

The model of the electric motor and ball-screw gear consist of a dynamic model of stock products. Therefore, we need accurate specifications of these parts before implementing the model. Not all suppliers specify the properties needed to do a proper analysis, so on the short term we'll have to focus on the ones who do.

For subjects up to 80 *kg*, the external recorded power around the ankle peak to 260W during push-off. The aim of this design is to provide a minimal support of 30%, which means we will select motors of 80W and higher. The motors pre-selected for this analysis and their corresponding information are all from the Maxon Motor [23] delivery program. This choice is made based on the labs experience with these motors and the favourable delivery conditions. The type and relevant specifications of the selected motors are listed in table 3.1. For the ball-screw gears, a selection of SKF products [24] are made that fit the roughly estimated loading requirements. Specifications of the supplier are listen in table 3.2.

Property	Symbol	Unit	Motor type			
			RE35-90W	EC32-80W	EC4p22-90W	EC4p22-120W
Electrical power	P_{max}	[W]	90	80	90	120
Winding voltage	U_{max}	[V]	24	24	24	24
Torque constant	K_t	[mNm/A]	29.2	20.5	14.0	13.5
Winding resistance	R_m	[Ω]	0.583	1.390	0.570	0.341
Nominal current	I_{nom}	[A]	3.47	2.44	3.88	4.81
Total mass	m_m	[g]	360	270	125	175
Rotor inertia	J_m	[gcm ²]	33.5	20.0	5.54	8.91
Max speed	ω_{max}	[rpm]	12000	25000	25000	25000

Table 3.1: Specifications of the pre-selected Maxon motors. Note that RE motors are brushed and CE motors are brushless

Property	Symbol	Unit	Gear type			
			SH6x2	SD8x2.5	SD10x2	SD10x4
Pitch	p_g	[mm]	2	2.5	2	4
Max velocity	v_{max}	[mm/s]	277	260	166	332
Max load	F_{max}	[N]	1500	2600	3500	5400
Efficiency	η_g	[—]	0.95	0.95	0.95	0.95
Nut mass	m_{g1}	[g]	25	25	30	40
Spindle mass	$\tilde{m}_{g2}$	[g/m]	180	320	510	430
Spindle inertia	$\tilde{J}_g$	[gcm ² /m]	0.7	2.1	5.2	3.8

Table 3.2: Specifications of the pre-selected Gears

	Variable						
	$\bar{x}_a$			r_1	r_2	ψ	c_s
Unit	[m]			[m]	[m]	[deg]	[Nm/rad]
Lower bound	[-0.2	...	-0.2]	0.2	0.08	80	300
Initial value	[0	...	0]	0.3	0.1	90	1000
Upper bound	[0.2	...	0.2]	0.35	0.2	120	5000

Table 3.3: Initial conditions and upper and lower bounds of the optimization

Because both cost function and constraints are non-linear the minimization problem is solved with an *active-set* algorithm of the *fmincon* function in Matlab. The initial conditions and upper and lower bounds are set the values in table 3.3. The bounds are chosen to prevent infeasible or impractical constructions in the detail design (e.g. a relatively short lever arm r_2 with a very low stiffness c_s will require a spring that is impossible to construct).

3.3 Results

All 16 combinations of motors and gears are successfully optimized, getting an minimal cost ranging from 48 to 85 [%] and mass ranging from 206 to 443 [g]. Table 3.4 lists all optimization metrics and the corresponding parameters except for the target position vectors $\tilde{x}_m$ of which a typical example is shown in fig. 3.5. The trade-off between performance and weight is best visualized by plotting $1 - f$ against m_{tot} is shown in fig. 3.4, where the best combination are in the lower-right corner. Figure 3.6 shows a typical plot of the supportive power and the contributions of the actuator and spring. The power of the actuator and spring add up to the support power.

Motor	Gear	Result				Optimized parameters			
		$1 - f$	m_{tot}	E_{mech}	E_{cu}	r_1	r_2	ψ	c_s
		[%]	[g]	[J]	[J]	[mm]	[mm]	[deg]	[$\frac{Nm}{rad}$]
RE35-90W	SH6x2	63.5	403	22.1	7.5	350	122	120	300
RE35-90W	SD8x2.5	61.6	417	21.5	7.5	350	137	120	300
RE35-90W	SD10x2	57.7	441	19.6	7.2	350	123	120	300
RE35-90W	SD10x4	51.0	443	15.3	7.0	350	153	120	300
EC32-80W	SH6x2	42.6	313	13.4	8.4	350	130	120	300
EC32-80W	SD8x2.5	34.1	327	10.5	6.2	350	088	120	300
EC32-80W	SD10x2	35.8	351	11.9	8.8	350	197	120	300
EC32-80W	SD10x4	22.8	353	7.0	7.3	350	109	120	300
EC4p22-90W	SH6x2	47.5	168	15.1	8.7	282	199	120	300
EC4p22-90W	SD8x2.5	39.7	182	12.1	8.6	350	197	120	300
EC4p22-90W	SD10x2	39.9	206	13.4	8.8	350	197	120	300
EC4p22-90W	SD10x4	29.3	208	8.5	7.8	333	113	118	300
EC4p22-120W	SH6x2	63.0	218	20.5	8.1	350	152	120	300
EC4p22-120W	SD8x2.5	55.1	232	17.8	8.4	350	192	120	300
EC4p22-120W	SD10x2	52.2	256	17.7	8.0	349	190	111	300
EC4p22-120W	SD10x4	42.6	258	12.2	7.4	350	148	120	300

Table 3.4: Overview of the optimization metrics and optimal parameters

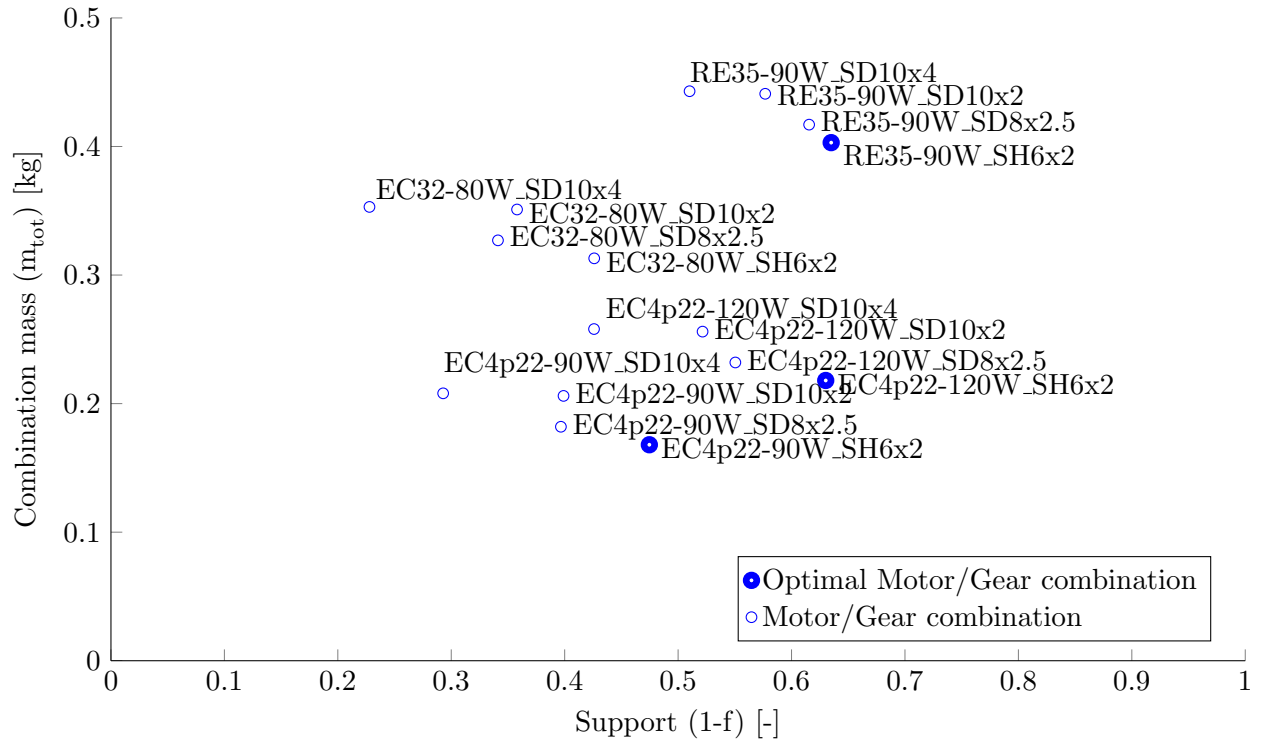


Figure 3.4: Plot of the total actuator mass m_{tot} versus the support ratio $1 - f$, where each circle $\circ$ represents a motor gearbox combination.

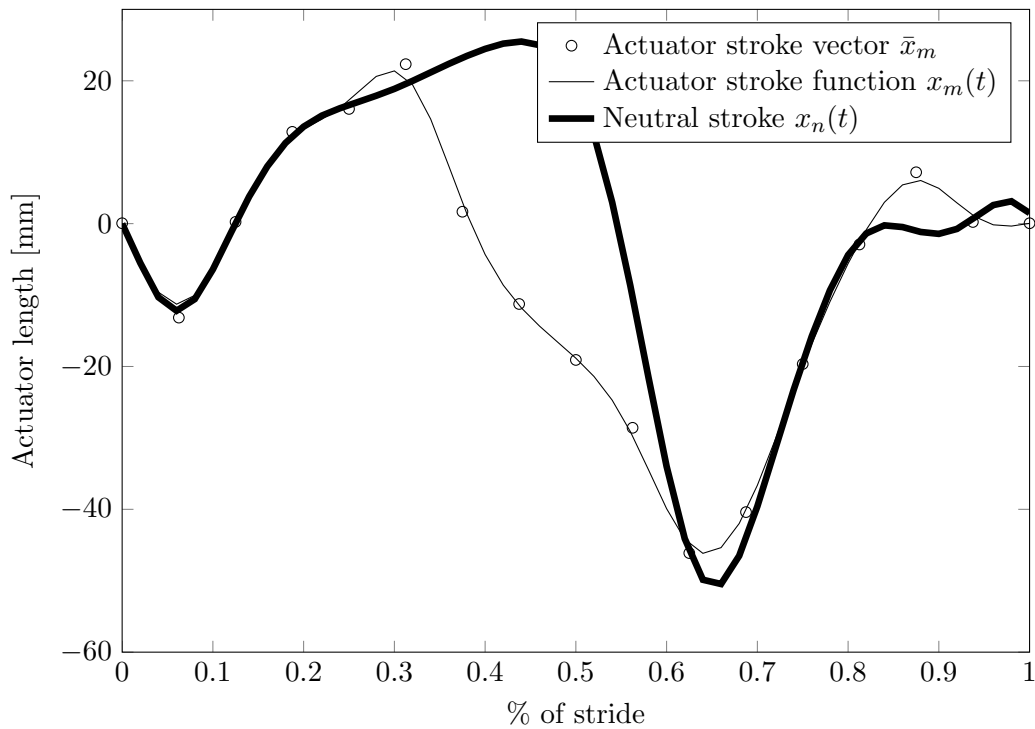


Figure 3.5: Actuator stroke and neutral stroke for the EC4p22-120W motor and SH6x2 ball-screw gear

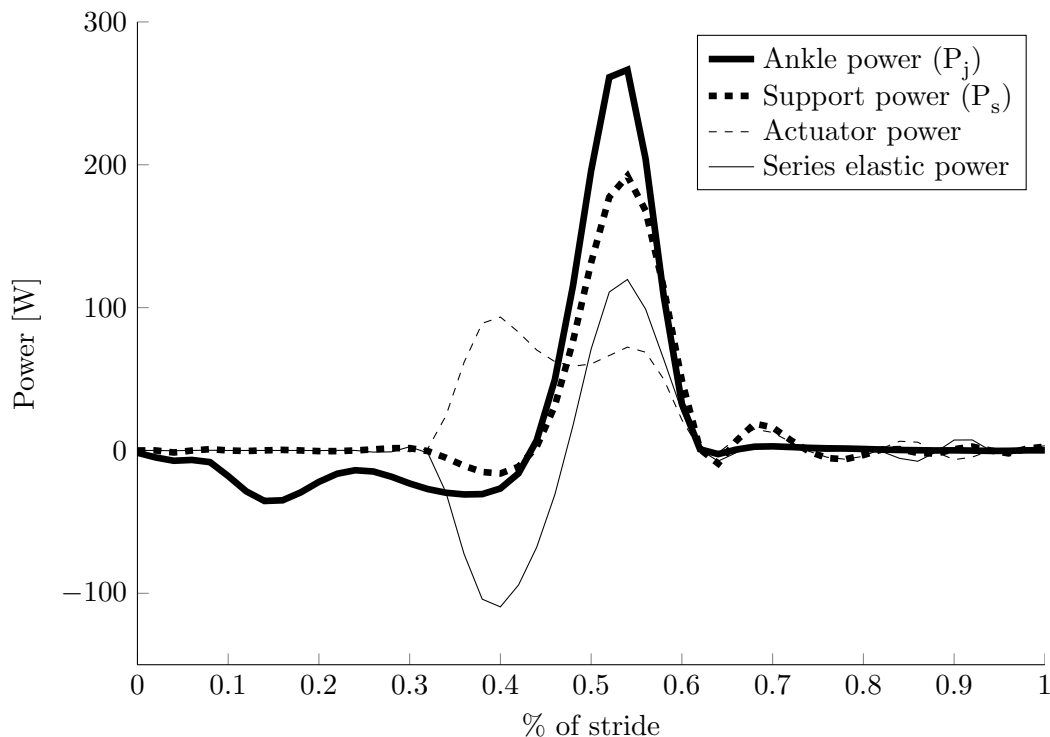


Figure 3.6: Plot of a typical power breakdown in time, with the external joint power P_j and support power P_s , and the power contributions of actuator and series elasticity

3.4 Discussion

Table 3.4 and fig. 3.4 present the trade-off between performance and weight, but not produce one clear “winning” combination. For the ball-screw gear, however, the spindle with the smallest diameter and pitch (SH6x2) outperforms the others. Next to having a smaller mass and inertia, these gears allow a higher feed velocity compared to their pitch, facilitating higher rotor speeds at smaller torques for which these type of motors can reach their maximum efficiency. Given the SH6x2 spindle, the RE35-90W, EC4p22-90W and EC4p22-120W motors are good candidates but each have some advantages and disadvantages:

- The RE35-90W shows the best performance (63.5%) without having the highest power rating, but is roughly twice as heavy as the other two.
- The EC4p22-90W is the lightest option but with lower support (47.5%). This is well above the required minimal support ratio of 30%, however this combination provides less safety margin, hence when taking into account some additional losses could be too weak.
- The EC4p22-120W has a performance similar to the RE35, but is much lighter. Although the power rating of this motor is higher than the others the copper losses are similar.

The considerations above did lead us to pick the EC4p22-120W motor with a SH6x2 ball-screw. This combination can theoretically generate 200 Watts of push-off power with a mass off 0.218 kilograms.

The strategy that arises from the optimization – to enable the system to generate 200 Watt push-off with a 120 Watt motor – is illustrated in fig. 3.5 and 3.6. Approximately 0.1 second before the start of the push-off the motor starts deforming the spring in order to store energy. When push-off initiates, the motor has to speed up and has to sacrifice some support, but then the spring starts releasing the stored energy, increasing the push-off power significantly.

The current results are valid for the heaviest subjects (80kg), but preliminary results of optimizations with lower subject weight showed that the configuration is should be robust to testing with lighter subjects. The only parameter that significantly changes with subject weight is the lever arm length r_2 , but when kept fixed, the performance does not deteriorate drastically.

3.5 Conclusion

This chapter introduced a dynamic model of a linear actuator capable of generating plantarflexion power, using an electric motor, ball-screw gear and a series elastic element integrated in a lever arm. By adding constraints that model the limitations of the drive components and a cost function that rewards a concentrated power burst at push-off, a standard minimization problem could be formulated. Running this optimization for combinations of 4 potential motors and 4 gears, resulted in a trade-off between performance and weight, from which – given some interpretation – the best option could be picked. The optimization showed that series elastic elements can drastically improve the power output of the device utilizing a special strategy during gait.

4.1. Introduction	.36
4.2. Main structure and interface	.36
4.3. Linear actuator	.38
4.4. Lever arm	.39
4.5. Ankle hinge	.44
4.6. Conclusion	.47

Now the actuation system is determined, and thus the forces and speeds that need be transferred to the human are known, we can start detailing the components that provide structure between them. Detail design include multiple design checks on strength, stiffness and component lifetime. To keep the device as light as possible, considerable effort has been put into minimizing the amount of material used as-well-as the choice of materials.

While in some cases, electronics – like sensors and cables – are considered in the final steps of the design (sometimes even afterwards); in a lightweight and compact design like this it is critical to fully integrate the sensors in an early phase, hence the sensory integration is also included in this chapter.

4.1 Introduction

Chapter 2 settled the overall concept of the orthosis and chapter 3 selected the best set of drive components, optimized the main dimensional design and series elastic properties of the actuation system. Integration of all components in a 3D model resulted in the overall mechanical design shown in fig. 4.1. The figure includes a short description of the main components. In the coming sections the components will be discussed as follows:

- 4.2 The structural design including the ergonomic shells (2,10) and their structural reinforcements (1,4,9)
- 4.3 Implementation of the drive components in a robust and safe housing with proper alignment and guidance (3).
- 4.4 Design and detailing of the flexible lever arm (6) - (8)
- 4.5 Robust and compact design of the ankle joint with integrated angle sensor (5).

All 3D modelling work is done in Solidworks, Simple design checks in Excel and finite element analysis in Ansys Workbench. For all design checks, a safety factor of 1.5 is used due to the highly dynamic loading conditions.

4.2 Main structure and interface

Requirements

The main structure has to transfer all interaction forces between the actuator and human or ground. The extremes are shown in table 4.1.

Limit	Calculation	Value	Unit
Maximal force in the actuator	$F_{a,max} = \max(F_a(t))$	500	[N]
Maximal support torque	$T_{s,max} = \max(T_s(t))$	78.54	[Nm]

Table 4.1: Requirements for the actuation design

The human interface has to at least provide a good fit for students with an average built with a mass up to 80 [kg].

Implementation

The concept chosen in chapter 2 includes ergonomic shells around the shank and foot – like in traditional orthosis – and add levers and hinges that can suspend the actuator. This choice limits the group of people that fit the device, but greatly increases the comfort of those who do. The shells are usually created from a plaster cast of the users leg, in this case of the authors legs. To make the shape more generic, features of the shape were neutralized and a double layer of padding was planned that can be removed when a subject has bigger legs.

The shells can be made from two materials: Plastic (Polypropylene) or Composites (Carbon fibre an epoxy). The main difference is that plastics need additional steel reinforcement while composite materials can be used for the full structure, providing that the inserts at the interface are robust enough and de-lamination of the composite can be prevented. Although using composites is a promising way of reducing the device's weight,

the cost and production time does not fit the current budget and planning, therefore making the use of plastics is the best choice for the moment. Moreover, choosing plastics leaves the possibility to make adaptations to the shell – removing and re-shaping – if the fit causes problems.

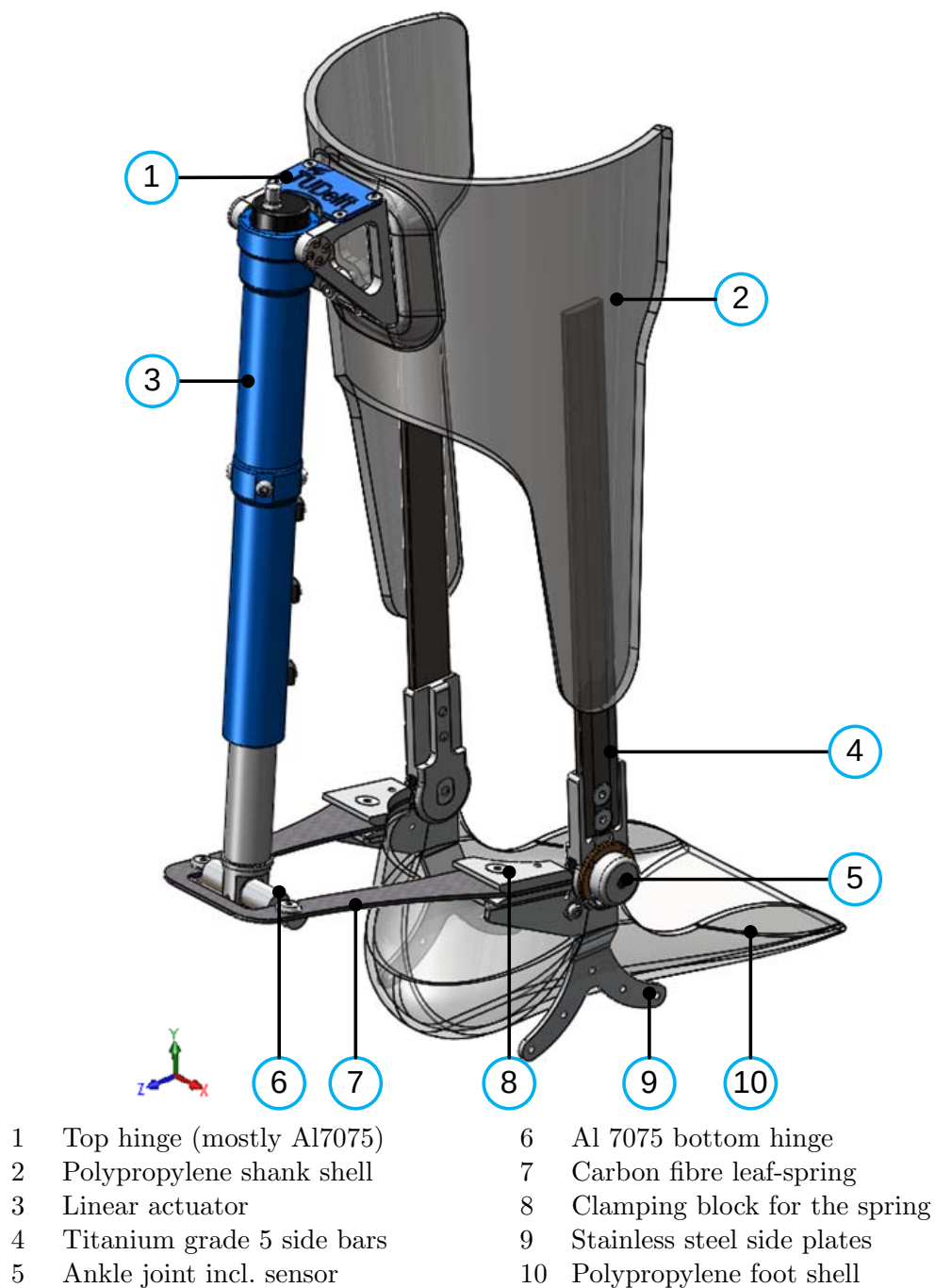


Figure 4.1: Overview of the right foot orthosis design

The plastic shells will be reinforced with titanium bars (4) at the shank and stainless steel plates (9) at the foot, connected via a threaded metal inserts. The top connection of the actuator to the shank (1) has to withstand high loads. To make a robust connection, a Aluminum plate is integrated in the plastic shell to which the hinge can be connected.

All design decisions concerning the human interface are made in collaboration with an orthopaedic workshop called Westland Orthopedie [25] – which has the expertise and facilities to create casts and shells and gave advice during several design meetings.

4.3 Linear actuator

Requirements

The relevant quantitative requirements needed for a detailed design are listed in table 4.2.

Limit	Calculation	Value	Unit
Maximal force in the actuator	$F_{a,max} = \max(F_a(t))$	500	[N]
Maximal torque of rotor	$T_{m,max} = \max(T_m(t))$	0.176	[Nm]
Maximal speed of the rotor	$n_{max} = \frac{\max(\dot{x}_m(t))60}{p_g}$	9972	[rpm]
Maximal stroke of the nut	$\Delta x_{max} = \Delta L_a$	75	[mm]

Table 4.2: Requirements for the actuation design

Implementation

The two core components of the linear actuator – the motor and gear – are optimized previously, but in order for these components to preform optimally, a proper implementation is essential. Figure 4.2 shows a cross-section of the actuator design, denoting all components.

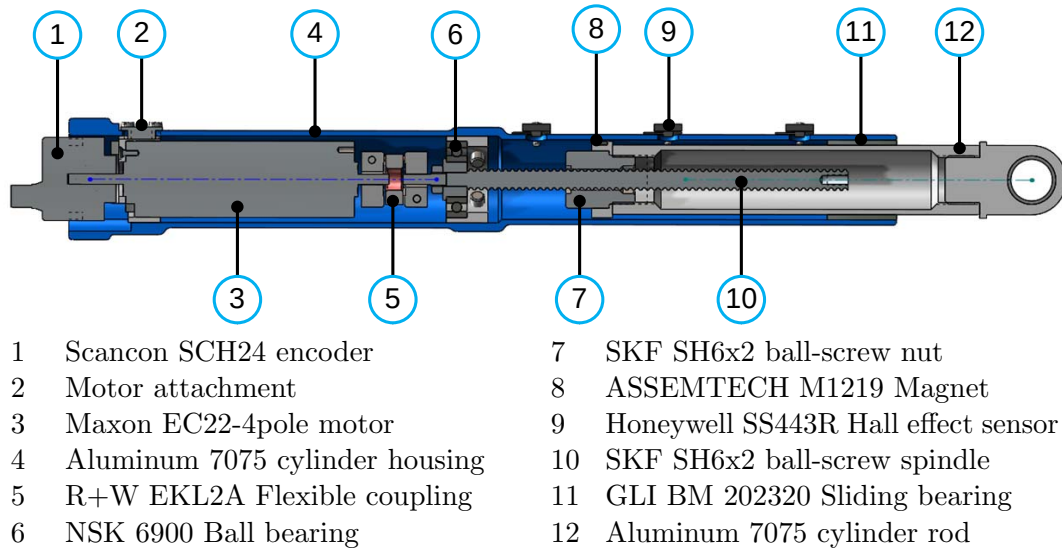


Figure 4.2: Cross-section of the linear actuator

The actuator consists of a Aluminum cylinder (4) that guides a hollow rod (12) through a PTFE coated sliding bearing (11). All drive components are safely mounted inside those components. The motor housing (3) is attached in the housing with an attachment part (2) at the back.

The motor shaft is attached to the spindle (10) through a flexible coupling (5). This coupling – type EKL2 of R+W Couplings [26] shown in fig. 4.3 – is specially designed to transfer high torques for its size and mass, but allow some axial movement (and small angular misalignments). The axial freedom is important to make sure that the bearings of the motor do not receive the full axial load $F_{a,max}$. Instead, the bigger radial ball bearing (6) will bear the axial load. The reason that a normal radial ball bearing is chosen for an axial load is that as long as the load is within the bearings' limits, friction is much lower than other bearing types [27] e.g. angular contact bearings and even axial bearings. Additionally, a single bearing can be used without facilities to add pre-load. Multiple options that fit the build-in dimensions were compared by means of a lifespan calculation, resulting in a choice for a NSK 6900 Ball bearing with a lifespan of $L_{10h} = 152h$ [28]. The outer bearing is fixed to the housing, while the inner ring is fixed to the spindle. The spindles rotations are converted to translations via a nut (7) attached to the rod (12)



Figure 4.3: EKL2 Flexible Coupling

To measure the stroke (x_s) of the actuator a Scancon type SCH24-2000-D-03-64-3-B incremental encoder (1) is attached to the back motor shaft. With a 2000 counts per turn, the resolution of the actuator stroke is 1000 counts per mm , which is more than required. The system is equipped with three hall effect sensors (9) attached to the housing: One near each endpoint and in the middle. These sensors can detect a magnet attached to the ball-screw nut. This way, the control system can do additional checks for safety purposes during operation. Next to the endpoint switch, a mechanical stop ensures that the cylinder does not exceed its normal range of motion.

4.4 Lever arm

requirements

The specific quantitative requirements necessary to design the spring, are listed in table 4.3.

Limit	Calculation	Value	Unit
Maximal force in the actuator	$F_{a,max} = \max(F_a(t))$	500	[N]
Maximal load angle spring	$\gamma_{max} = \max(\gamma(t))$	114.7	[deg]
Minimal load angle spring	$\gamma_{min} = \min(\gamma(t))$	86.2	[deg]
Required spring stiffness	c_s	300	[Nm/rad]

Table 4.3: Requirements for the spring design

The exact behaviour of the spring is difficult to predict in advance. therefore it is beneficial to fabricate a series of springs with several stiffness's and lengths, hence the production technique used to create the springs should suited for making multiple variants of the spring in a cost-efficient way.

Analysis and Implementation

The spring model introduced in chapter 3 assumed a linear leaf-spring with a rotational stiffness of c_s around the ankle joint. Given that $x_s = r_2 q_s$ and $T_s = r_2 F_s$ the translational stiffness of the spring is

$$k_s = \frac{c_s}{r_2^2} \quad (4.1)$$

which results in 12.98 N/mm which is a more convenient measure when designing the spring. Given a deformation of 38.52 mm (at 500 N) on the endpoint, the resulting stress in the spring will put high demands on the material. Research in the auto mobile industry [29] – where similar requirements apply – compares spring materials by looking at the specific strain energy

$$\bar{S} = \frac{1}{2} \frac{\sigma^2}{\rho E} \quad (4.2)$$

where σ is the bending strength, ρ the material density and E the modulus of elasticity. Figure 4.4 compares the specific strain energy of the available spring materials: Spring steel, Titanium, Glass Reinforced Plastic (GRP) and Carbon Reinforced Plastic (CRP). The comparison shows that E-Glass has the highest energy density, however CRPs have a better resistance against fatigue.

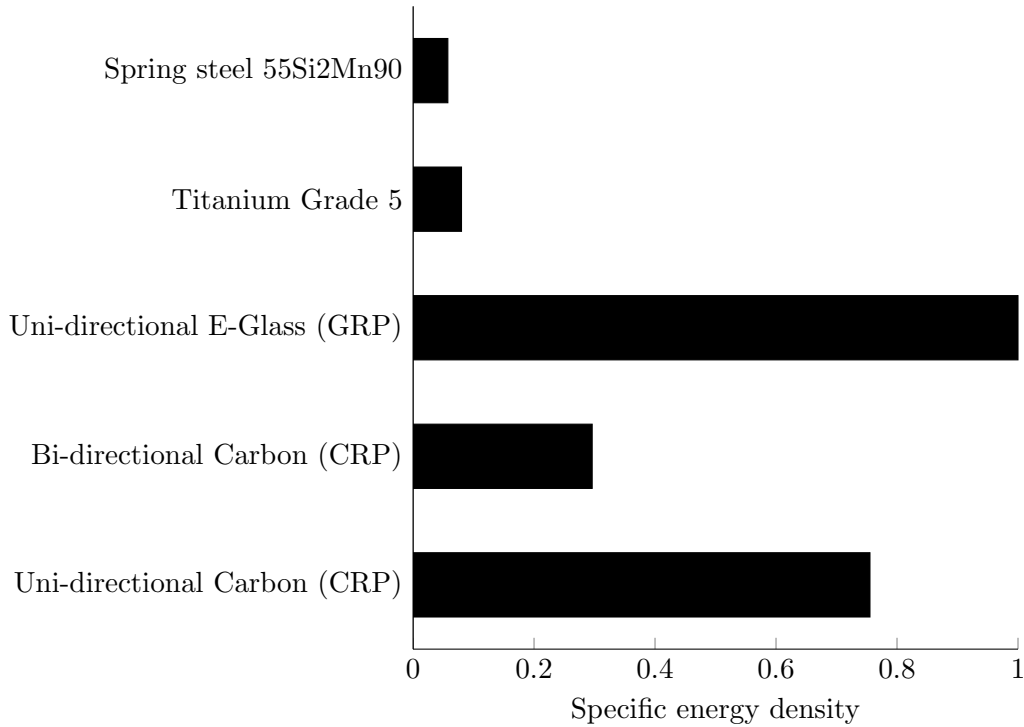


Figure 4.4: Specific energy densities of spring materials, normalized to the maximal value, based on the ultimate tensile strength.

An additional consideration to choose for CRPs is that more fabrication methods are available for these materials. Bearing in mind that the springs' production should be fast and cost efficient, the decision was made to make CRP plates of uniform thickness and cut a set of springs from it using a water- jet cutter. This way of production is cost efficient and flexible, because no mold is needed. The sacrifice made here, is in the shape of the spring. For an optimal spring design, the thickness of the spring t is usually tapered

(linearly or quadratically) to equally distribute the strain over the springs' length. In this case h is constant and only the width of the spring w can be varied over length. The direction of the carbon fibres – uni or bi directional – can be varied per layer (each layer is approximately 0.2 mm). The choice for this design is made to apply all unidirectional layers along the line of deflection between two single bidirectional layers for some structure in the non-loading direction. The material properties of unidirectional carbon are assumed to be dominant and linear isotropic: Tensile strength σ_t of 1850 MPa and modules of elasticity E of 130 GPa (obtained from the supplier).

To design a spring with the proper stiffness, the goal is to determine the desired thickness h (amount of layers) and width profile $w(x)$, where x is a point on the springs length. The lever arm length r_2 is 152 [mm], but this is not the effective length of the spring, hence this would require a leaf-spring attached in the centre of the joint, which is not feasible i.e. some length is reserved for the hinge and clamping mechanism. A realistic dimension of the spring length r_{eff} can be obtained from the implementation shown in fig. 4.5. The measure from the start of the tangential contact radius r_c to the centre of actuator attachment is 100.4 mm.

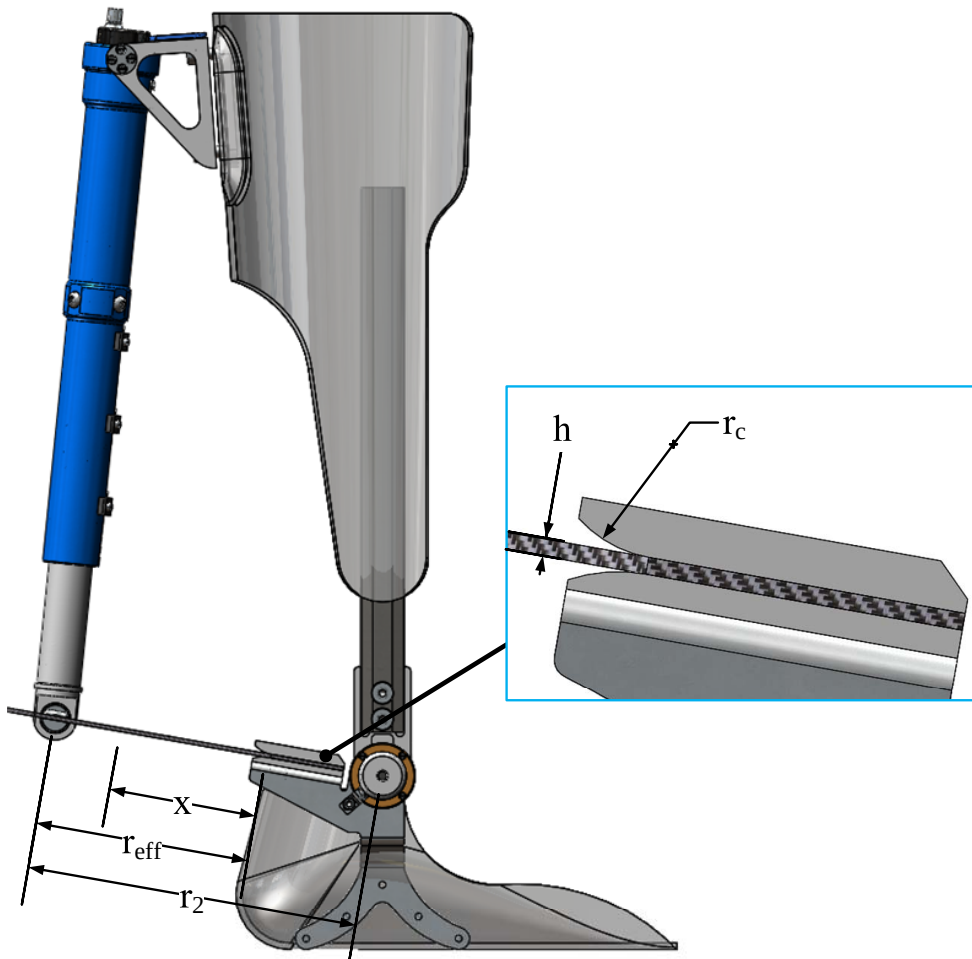


Figure 4.5: Illustration of the spring implementation, where the r_2 is the total lever arm length, r_{eff} the effective length of spring from the start of contact radius r_c to the actuator attachment, and t the spring thickness.

Finding the parameters h and $w(x)$ is done by taking the following steps:

1. An initial tool for finding a feasible spring stiffness as lower bound for the actuator optimization (chapter 3) is a simple finite element model of a beam with a rectangular cross-section in Matlab with a fixed width and linearly tapered thickness. Optimizing the parameters with this tool, resulted in a indication of the thickness of 3 to 1 mm and a total width of 60 mm .
2. Next, an adapted version of the springs 3D model (in SolidWorks), with features to apply the proper supports and loads, was linked to ANSYS workbench. Figure 4.6 shows Ansys model in including the fixed support (blue area) and load (red area). The load of $F_{a,max}$ was applied under an angle γ_{max} , to simulate the maximum force at the maximal angle.
3. The width profile $w(x)$ is the parameter that can be varied along the length. The goal is to distribute the stress over the length of the spring. Of the multiple variants simulated, the shape in fig. 4.7 showed good results
4. Finally, the thickness is tweaked to get the desired stiffness of 12.98 N/mm . The approach is to simulate a range of thicknesses from 1 to 3 mm and track the node that corresponds to the actuator attachment. Figure 4.8 shows the deformed spring that resulted from one of the simulations. The force-travel plots of the thicknesses with a stiffness closest to the desired stiffness are shown in fig. 4.9. The approximate linear stiffness is determined by fitting a linear curve on the Ansys results.

The above steps revealed that a leaf-spring of 1.8 mm gives an approximate stiffness of 13.03 N/mm , close to the desired one. The maximal resulting Von-Misses stress for this thickness is 1218.2 MPa , which is within safety margin.

In the modelling and design of the spring, some rough assumptions were made. Exact properties and performance of the spring will have to be determined experimentally (see chapter 6). To take into account that the actual properties of the device will deviate from the calculated ones, the spring will be manufactured in three thicknesses: 1.6, 1.8 and 2.0 mm . This will give more flexibility in the experimental phase.

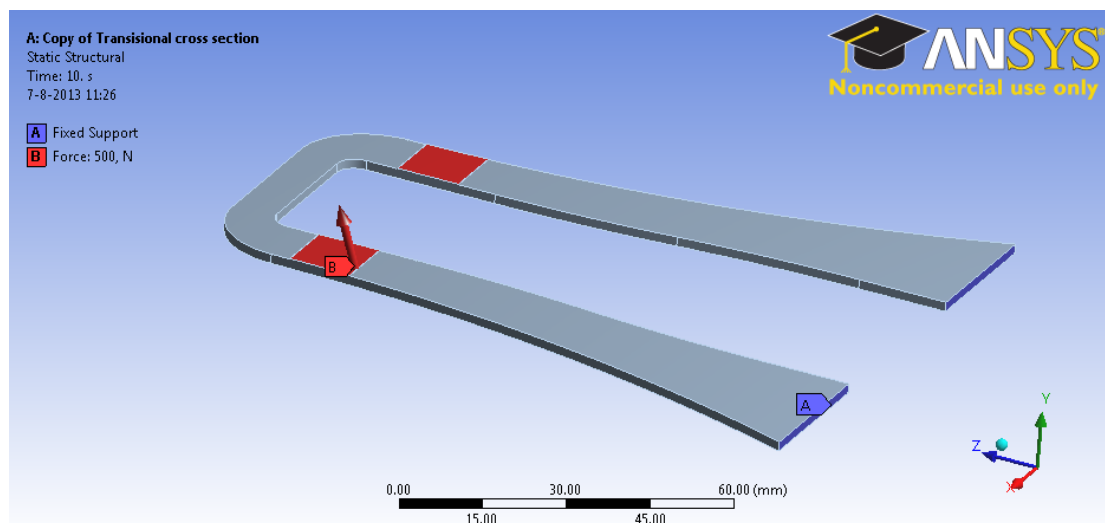


Figure 4.6: Ansys model import, with fixed support and forces applied

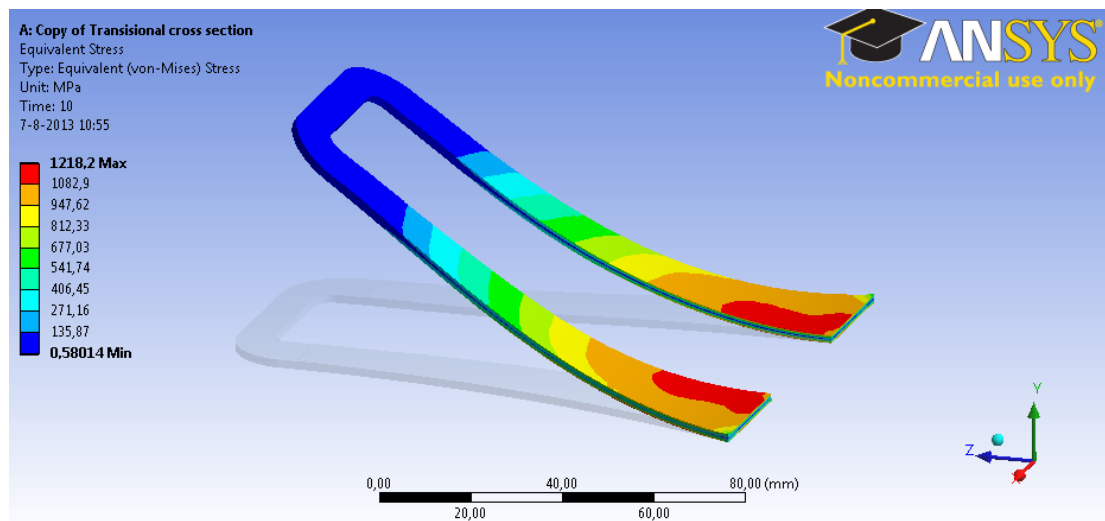


Figure 4.7: Screenshot from Ansys workbench of the von-Mises stress

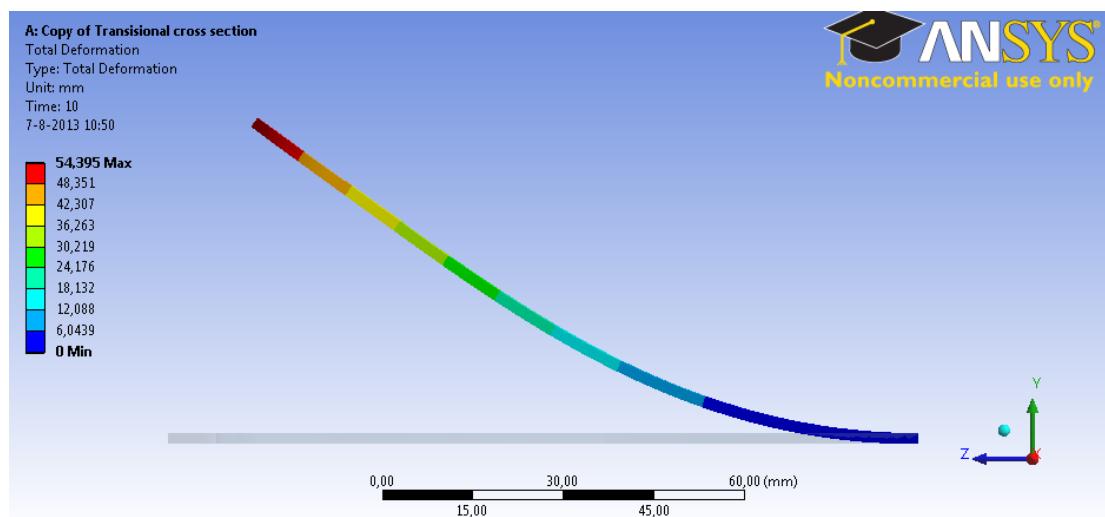


Figure 4.8: Screenshot from Ansys workbench of the deformed spring.

All decisions concerning design and fabrication of the composite leaf-springs are made in collaboration with Refitech [30], a company with the extensive expertise in fabrication of fibre reinforced plastics.

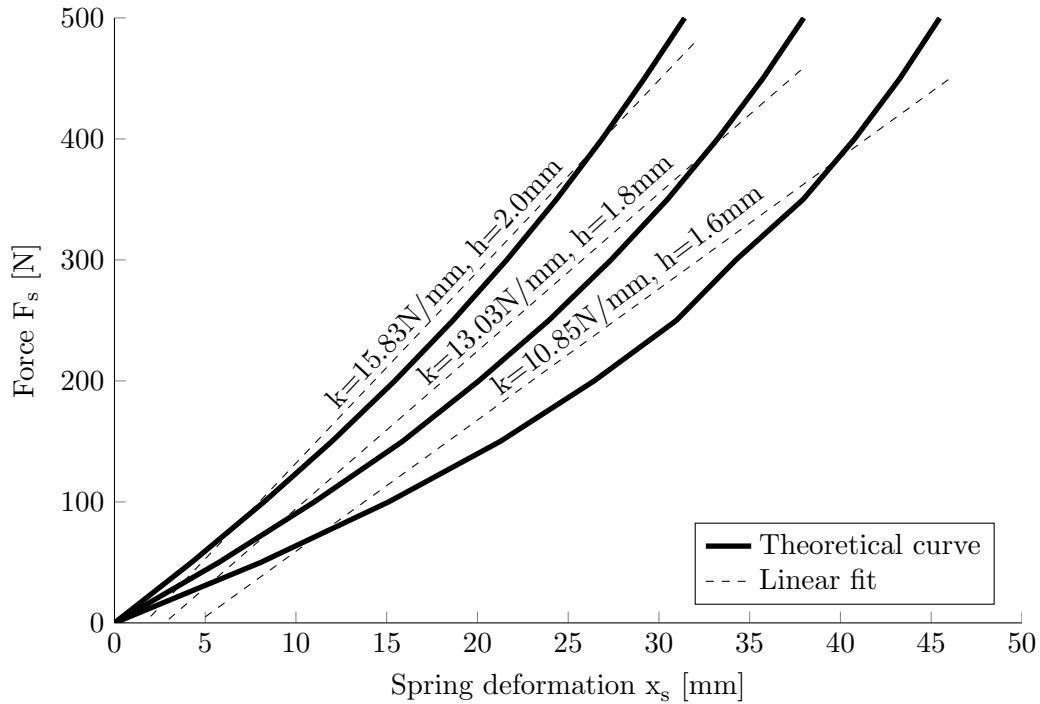


Figure 4.9: Simulated stiffness of the leaf-spring for three thicknesses (1.6, 1.8 and 2.0 mm).

4.5 Ankle hinge

requirements

The requirements needed to design the ankle joint are listed in table 4.4. Note that the angle limits of the device equal the maximum angles during normal walking plus a margin of 5 *deg*

Limit	Calculation	Value	Unit
Maximal speed of the ankle joint	$\dot{q}_{j,max} = \max(\dot{q}_j(t))$	40.7	[rpm]
Ankle dorsalflexion limit	$q_{j,max} = \max(q_j(t))$	10+5	[deg]
Ankle plantarflexion limit	$q_{j,min} = \min(q_j(t))$	-20-5	[deg]

Table 4.4: Requirements for the ankle joint, at normal waling speed.

The ankle hinge has to limit the amount of plantar and dorsal flexion to prevent injuries to the user.

Implementation

The ankle hinges connect the foot and shank structure giving the ankle one degree of freedom: Plantar and dorsal flexion. There are two mirrored hinges on both sides of the foot, but the outside one (left side of the left foot and right side of the right foot) is equipped with an encoder that measures the joint angle. A cross-section of the outer hinge is shown in fig. 4.11. The foot side plate (10) is suspended – in a slot in the housing (1) – radially by a press-fitted brass bearings (8) sliding over a fixed shaft (7) and axially by two plastic plates (9).

Components (2) - (6) are part of the joint angle sensor. Measuring the joint angle requires an encoder, preferably an absolute type to prevent recalibration each time you power-on the device. Furthermore, the build height should be as low as possible to reduce the risk of hitting the sensor to an external object while walking. Searching and fitting products from multiple suppliers resulted in finding an encoder that consists of a small PCB with separate magnet (see Figure 4.10) from RLS (Renishaw) [31], ideal for integration purposes. The exact type is RMB20IC13BC which has a single turn 13bit resolution and a SSI interface. Figure 4.11 shows that the PCB (3) is fixed to the ankle joint housing (1) and the magnet (4) is glued to Aluminum sensor housing (5) that can rotate in the slider rings (2) and connects to the foot side plates through a bracket (6).



Figure 4.10: Joint encoder

To protect the user from the device over-flexing the joint, mechanical end-stops are integrated in the hinge as shown in fig. 4.12. Angled edges (A) and (B) *foot side plate* will hit the *hinge housing* at the respective maximum plantar and dorsal flexion angles. Finally, the last design check involved the total range of motion. The main assembly is incrementally moved through its whole range of motion while it is checked for interferences as shown in fig. 4.13.

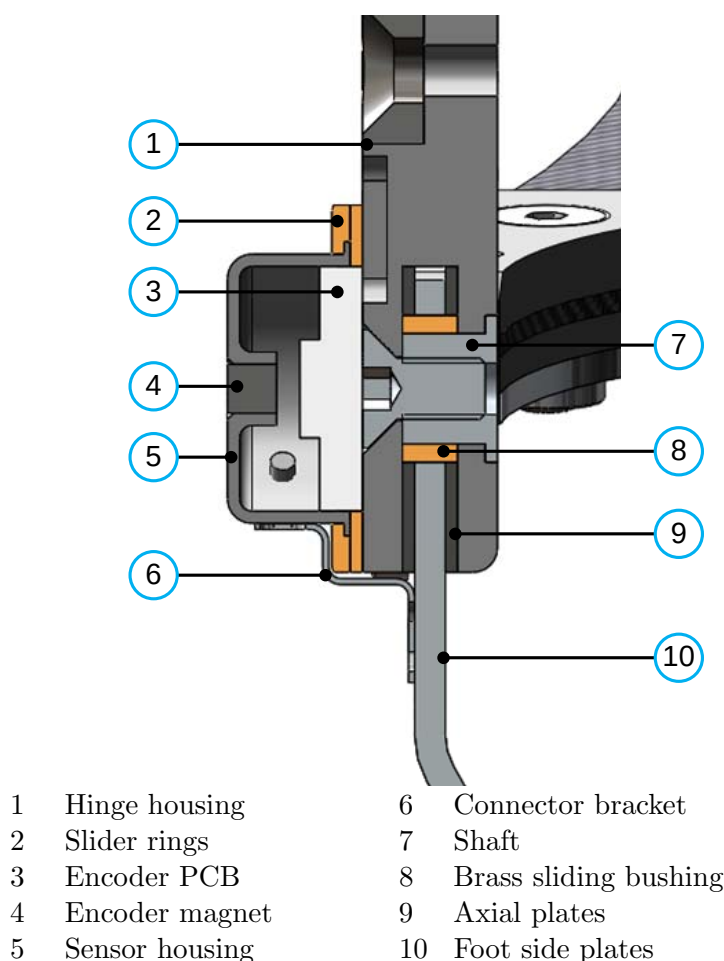


Figure 4.11: Frontal cross-section of the ankle hinge

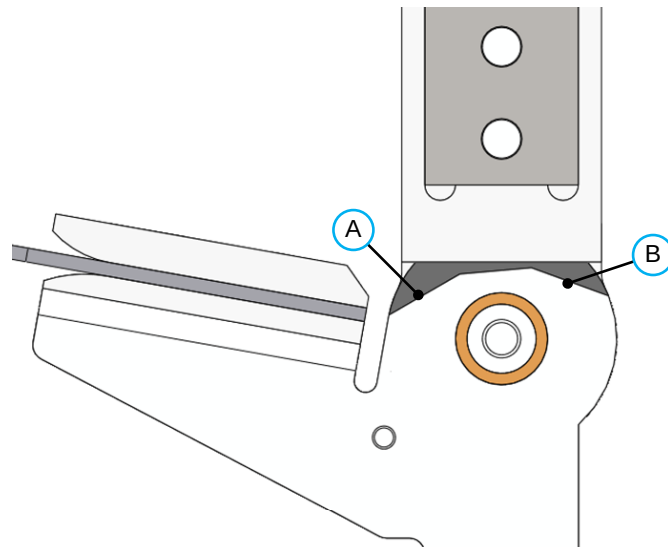


Figure 4.12: Sagittal cross-section of the ankle hinge.

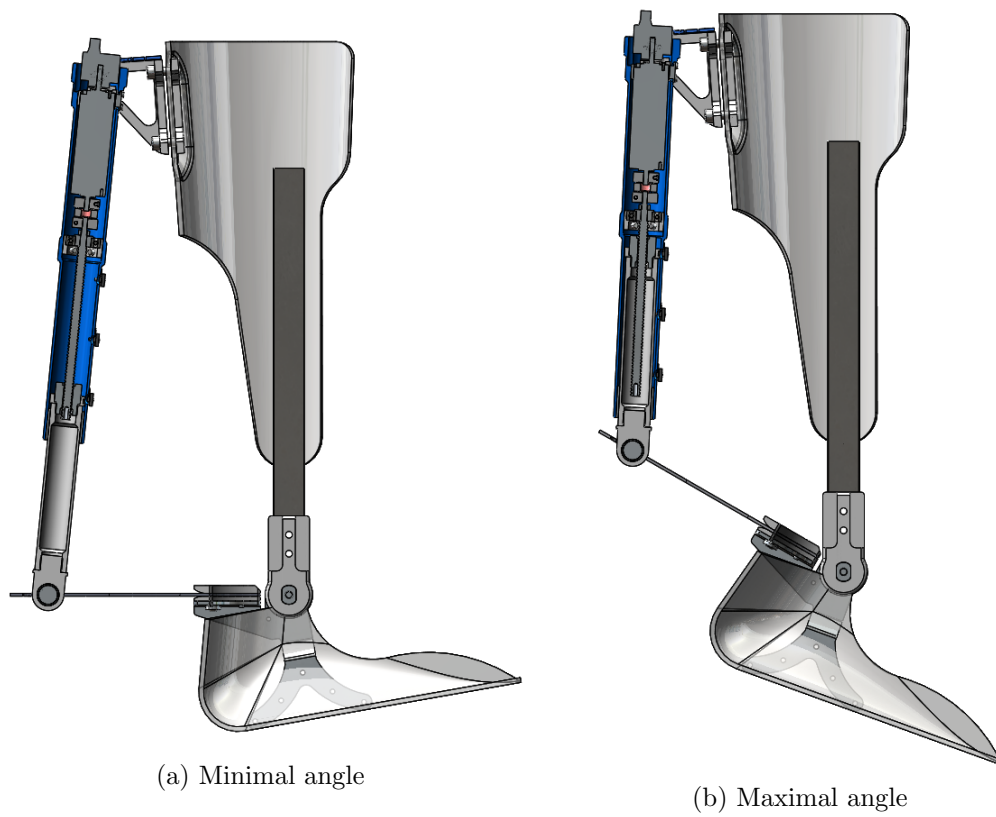


Figure 4.13: Range of motion check

4.6 Conclusion

This chapter reports all the details of the orthosis design and the resulting ready to produce 3D model. The ultimate goal is to construct a safe and lightweight device that can be comfortably fit around the foot and lower leg.

To ensure the users safety several design checks are done on the components maintaining a safety factor 1.5. The rotating parts of the actuator we fully covered by the structure.

To keep the added mass of the device to a minimum, the amount of material used for each custom part is kept minimal. For the spring mechanism, composite materials are chosen to reduce the mass further. To get some insight in what assemblies and component groups add most mass to the device, the mass of each part is obtained from the mass properties in SolidWorks. The mass contribution of the assemblies and component types are shown in fig. 4.14a and 4.14a. Note that most weight is used for the structure of the device. The shells and reinforcements of the shank and foot are large contributors. Applying composite materials for these components in the future, could be one of the main reductions in mass.

The total mass adds up to a theoretical 1454 g with the centre of mass at $(x, y, z) = (0.47, 144.58, 57.48)$ mm in the local coordinate frame (see fig. 4.1 for axis) of the shank and measured from the ankle joint.

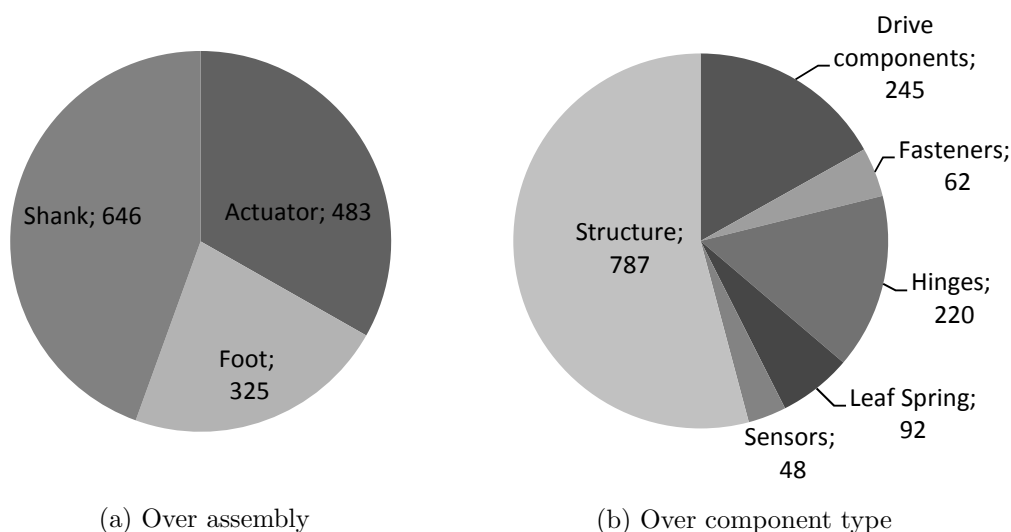


Figure 4.14: Mass distribution of the design. Total mass is 1454 g

5.1. Introduction	.50
5.2. Control Hardware	.51
5.3. Control Software	.55
5.4. Conclusion	.58

The mechanical design of the device is now finished. From previous chapters already reveal a general strategy on how to control the exoskeleton. The specifics, however, play a crucial role in achieving successfully functioning device.

This chapter will show an overview of chosen hardware implementation and motivate some of these choices. Next, a detailed section on the software implementation will focus on the software used to control the device.

5.1 Introduction

As determined in the concept phase, the orthosis will be controlled from electronics housed in a backpack. The overview of the selected electronics in fig. 5.1 shows how all active components in the orthosis (bottom row) are connected to the computer and power source.

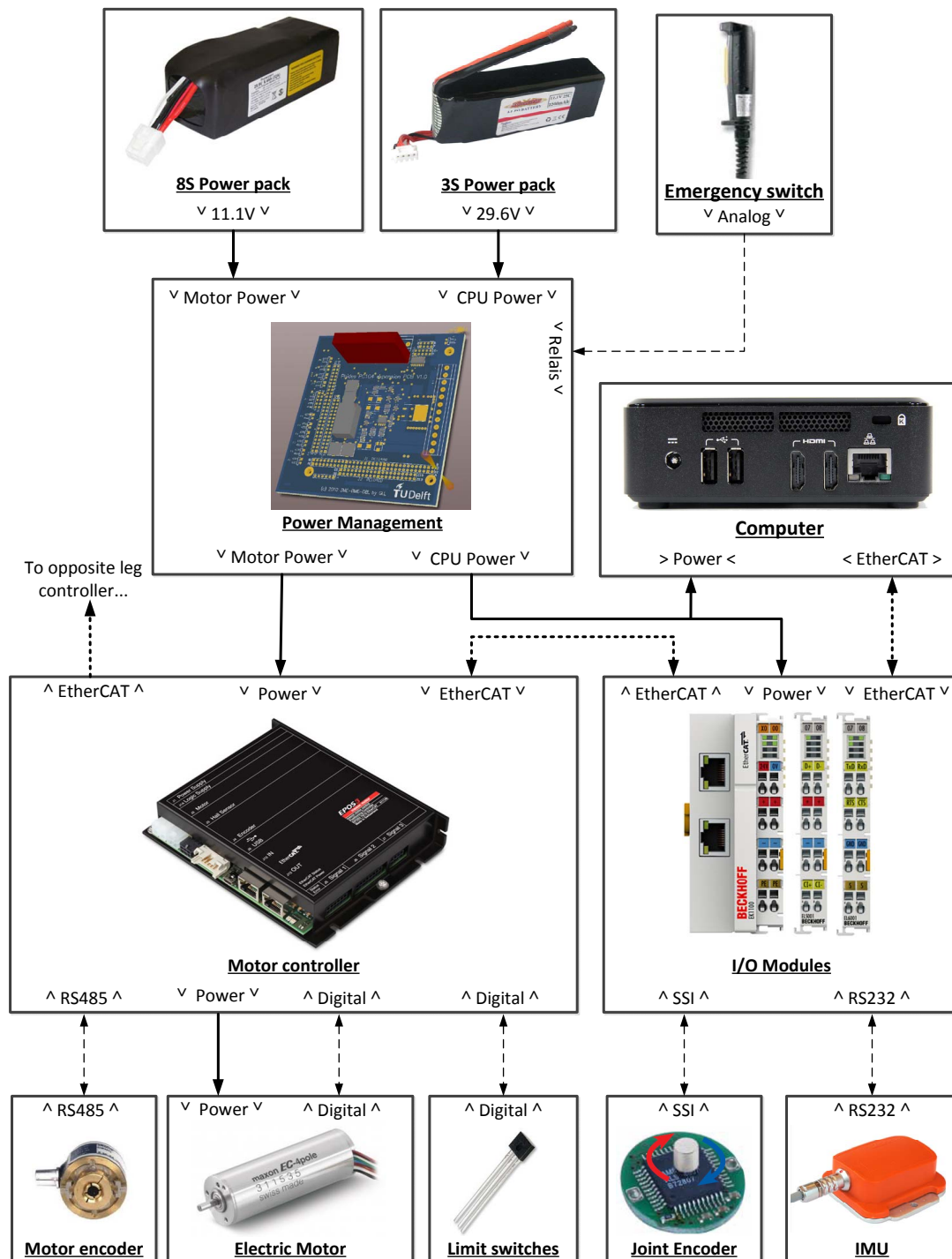


Figure 5.1: Schematic overview of the hardware and general signal routing

A fast and flexible system called “EtherCAT” [32] is chosen for data communication (see dotted linesfig. 5.1) between the motor controller, I/O module and Computer . This system supports real-time operation and provides the flexibility by allowing daisy chaining of components.

Section 5.2 will describe all selected hardware and justify their choice. Section 5.3 will focus on the software architecture and controller implementation.

5.2 Control Hardware

Requirements

The control hardware should be able to process all incoming signals, do computations based on these signals and regulate the electric motors. All relevant requirements for selection of the hardware are listed in table 5.1.

Limit	Calculation	Value	Unit
Maximal motor current	$i_{m,max} = \max(i_m(t))$	13.04	[A]
Motor voltage	$V_{m,max} = V_{cu}$	24	[V]
Energy consumption	$E_{el} = E_{mech} + E_{cu}$	28.6	[J]
Battery operation time	t_{OP}	2	[h]
Maximum backpack mass	$m_{b,max}$	5	[kg]

Table 5.1: Requirements for the control

The hardware should not interfere with the user, nor should it cause harm to the user. Furthermore, hardware itself should be protected from damage. Therefore, all components should fit a small envelope and the electrical connections should be fully shielded or covered. The fact that the user has to carry a reasonable amount of weight in his/her back is unavoidable, but the fit should enable the user to wear the backpack for 1 hour without any discomfort.

Computer

An *Intel NUC Kit DC3217IYE* is selected as the main computing unit. Specifications of the device are listen in table 5.2 and show that this is a very compact and lightweight device with relatively good specifications. An important advantage of this device is that it has a Intel 82579V chipset for LAN, which has good compatibility with EtherCAT.

Description	Value
Processor	Core i3 (i3-3217U@1.8Ghz)
Memory	4 GB DDR3
HardDisk	128 GB SSD
Connections	Gigabit LAN (RJ45), WLAN 802.11b/g/n, 2x HDMI, 3x USB2.0
Input Voltage	19 [V] $\pm$ 10 %
Mass	1 [kg]
Dimensions (LxWxH)	117x112x39 [mm]

Table 5.2: Specifications of the NUC

Motor control

The motor selected in chapter 3 is: *Maxon EC22-4pole 120W 24V*. Because of the double pole pair and brushless commutation, controlling this motor requires a specialized motor controller. Additionally, the controller has to be EtherCAT compatible and meet the power requirements. From multiple feasible options, the *EPOS3 70/10 EtherCAT positioning control unit* of Maxon motor [23] is selected, because of its good compatibility with the motor and acceptable lead time. The relevant specifications are listed in table 5.3.

Description	Value
Operating Voltage range	11-70 [V]
Max. continuous/peak current	10/20 [A]
Required inputs	RS485 Motor encoder, Hall sensors
Extra inputs	11 Digital ¹ , 2 Analog
Extra outputs	5 Digital, 1 Analog
Voltage supply	3 x +5 [V] DC
Mass	442 [kg]
Dimensions (LxWxH)	150 x 120 x 29 [mm]

Table 5.3: Specifications of the Maxon EPOS3

This device has its own firmware that handles the lower level control. Of the several control modes available on the device, positioning mode is the one needed on our device as will be discussed in section 5.3. Two controllers are needed, one for each leg.

Input / Output Modules

The I/O functionalities integrated in the motor controller, are not completely sufficient to connect all devices. The respective SSI² and RS232 signals from the joint encoder and inertial measurement unit (IMU), are connected to the system. Beckhoff [33], one of the founders of the EtherCAT system, provides a expandable I/O system. The bases of this system is an EK1100 Coupler, which connection to the EtherCAT system on one side and to a stack of terminals – individual components with a I/O functionality that can be put into series (see fig. 5.1) – on the other. In this case, a dual channel SSI input terminal (EL5002) and single channel RS232 input terminal (EL6001) were added to the stack, hence completing the connections of all components to the system.

Description	Value
Input Voltage	24 [V] (-15%/+20%)
Mass	105 [kg]
Dimensions (WxLxB)	44 x 100 x 68 [mm]
Additional Modules	EL5002, EL6001 ³

Table 5.4: Specifications of the Beckhoff EK1100

¹with SSI encoder functionality

²Although the EPOS3 controller is capable of reading SSI encoder information, initial testing pointed out that this option was not available for positioning control mode

³Each module adds 12 mm width (W) to the EK1100

Power source

To select a motor battery with sufficient capacity, an estimate of the total energy consumed by the orthosis in 2 hours of walking is needed. The energy for one step with one leg obtained from simulation is E_{el} , hence the total energy is calculated with

$$E_b = 2 \frac{t_{op}}{t_{cycle}} E_{el} \quad (5.1)$$

resulting in required capacity 411840 J or 114.4 Wh capacity. The battery voltage is advised to be higher than the motor voltage of 24 V . Lithium polymer (Lipo) batteries have good capacity and discharge rates. The available Lipo packs are composed in series to increase the output voltage. In this case, a type with eight batteries in series (8S) was chosen which has a 29.6 V output.

Given a minimal capacity of 3865 mAh for 2 hours, the battery acquired for the motor is a *ZIPPY Compact 5000mAh 8S 25C Lipo Pack* [34] shown in fig. 5.2, which weighs 937 g . Some extra capacity will be needed for powering the controller and compensating for losses. The 25C specification in the battery type means that it can discharge at a rate of 25 times the capacity, hence 125 A , far above the required 13.04 A .



Figure 5.2: ZIPPY Compact 5000mAh 8S 25C Lipo Pack

The NUC and Beckhoff EK1100 are fed with a second battery, to prevent voltage-drops and other artefacts caused by the motor/controller interactions to compromise the higher level control stability. Experiences with similar set-ups show that both devices consume up to 1 A each during normal operation at 20.6 V , hence 41.2 Wh . The battery pack chosen for this was a *ZIPPY Compact 5800mAh 3S 25C Lipo Pack* (11.1 V) of 433 g , hence 64.3 Wh . This battery does not meet the 2 hour operation time requirement, however, this was the highest capacity for 3S batteries. Higher series batteries are not supported by the power management board discussed in the next section. To compensate for this inconvenience, electronics to enable swapping batteries without powering-off are included.

Power management

Maxon documentation mentions about EPOS controllers: "4-quadrant amplifiers are able to feed back brake energy into the supply and therefore work like a generator. Thus a long braking process can cause the supply voltage to rise due to the fed back energy." This effect can cause damage to Lipo batteries when directly connected to the controller, and in some cases cause them to explode. To guaranty safety of the batteries and user, separate power management electronics is necessary. The custom developed power management electronics, includes the following features:

- Shunt regulation, to prevent the generating motor (e.g. during braking action) to overload the batteries. The excess energy is transferred into heat.
- Two solid state relays to switch on/off motor and computer power separately.
- Control panel with seven digital input buttons. Four are reserved as power switches and three are available to create interrupts to the computer (via the motor

controller I/O). The panel also includes LEDs for the following messages: Motor/computer enabled indication, motor/computer power low indication. Some additional LEDs are available for custom system messages.

- Voltage converter to provide the proper voltage to the computer and I/O modules (20.6 V).
- Hot-swap board for the computer battery, enabling easy swapping of batteries without having to power-off the device.

Back Pack

The previous chapters discuss all the components needed for control of the orthosis. The requirements state that those parts need covert sufficiently to prevent damage to the electronics and user. Hard shell backpacks are designed to protect inventory in high speed, high exposure situations (e.g. motor cycling, dirt biking or snowboarding). In this case, the *peoples delite executive* series of Boblbee [35] shown in fig. 5.3 is chosen because of the double compartment system: The main compartment (A) can harbour the electronics, while the separate bag at the bottom (B) can yield the battery packs. This way, switching the batteries is fast and easy. The backpack itself weighs 1.6 kg.



Figure 5.3: Backpack yielding the control electronics

Now, the whole control system is integrated in one compact mobile unit, interfacing with the software on the NUC for testing becomes inconvenient, since including the a keyboard, monitor and mouse wouldn't be very practical. Instead an Wireless SSH

connection is established between though a private network between a laptop and the NUC, which make is easy to control the device while a user is wearing it.

5.3 Control Software

EtherCAT

Control of the actuation system requires a fast and reliable means of communication between multiple the computer, I/O and motor controllers of the right and left leg. EtherCAT [36], communication protocol base on Ethernet technology, is very suited for this purpose. As mentioned in the previous chapter, all devices are compatible with EtherCAT communication, and can therefore by easily coupled together by daisy chaining them with standard Ethernet cables as shown in fig. 5.4. In this case the Intel NUC acts as a Master device and the Beckhoff terminals as-well-as right and left leg EPOS3 controllers are slaves.

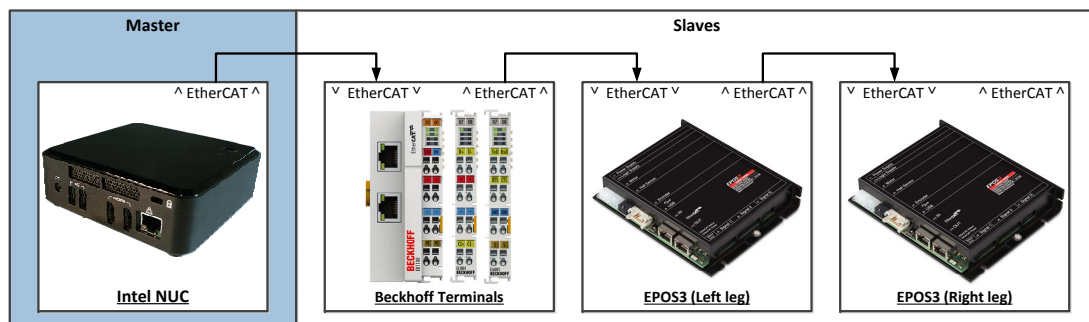


Figure 5.4: Schematic of the EtherCAT master-slave system

Master set-up

The Master device was equipped with the following software:

- Ubuntu 12.04 LTS, a widely used Linux distribution, is chosen for the operating system on the NUC. By extending Linux with Xenomai and the Adeos patch the system is able to execute code in real-time.
- Matlab version 2011b for Linux including simulink. Simulink provides a convenient way to create a controller using much built-in functionality and graphic programming. After creating the model it can be translated into real-time compatible c-code using `gcc-4.6` (GNU Compiler Collection) via Simulink *code generation*.
- Simple Open EtherCAT Master 1.3.0 [37] (SOEM). A piece of compact but powerful software that can turn any computer with a RJ-45 port into a EtherCAT Master device.
- E-box software from the Eindhoven University of Technology repository [38] creates an interface between the SOEM, the compiled code and Simulink using external mode.

When starting the model, the master detects all slaves attached to the network and request one of the states according to the schematic in fig. 5.5. If the timing and safety

requirements for that specific device is met, the slave will go to that state. Each state allows specific types of communication and gives access to different settings and variables of the slaves. In the initializing the BUS in the *init* state, the slaves go to *Pre-operational* state. In this state, the mapping of slaves' process variables is determined by setting up the sync manager (e.g. the EPOS controllers require a different mapping for different control mode). Next, the slave can be set to *safe-operational* state in which the slaves' static parameters (regulator gains, motor properties, etc.) can be set. Finally, the slaves can reach *operational* state, where process data (e.g. regulator set-points, actual position, etc.) can be send and received . Pre-, safe- and operational mode allow acyclic (mailbox) communication (SDO), and operational mode allows cyclic communication (PDO). The latter enables real-time communication between master and slaves.

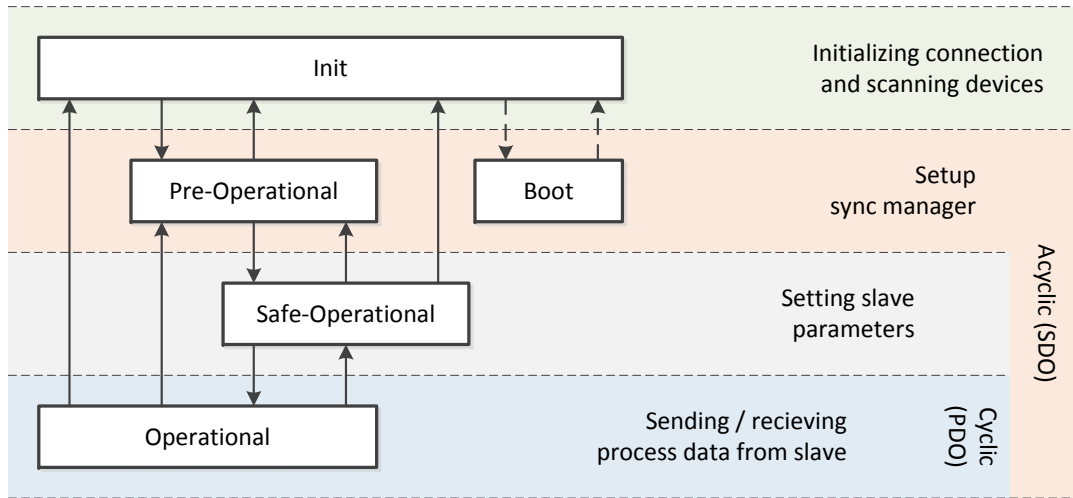


Figure 5.5: Schematic of the EtherCAT state machine [33]

Using the combination of hard and software described above, does not automatically work after installation. Getting it to work, required many iterations of solving compatibility issues, debugging and writing custom c-code. Although the amount of text dedicated this subject does not justify the amount of work that went in to it, further details of this process will not be described in this report. The author and supervisor do intent to make the software available for others.

Controller implementation

The controller of the orthosis described in this section is meant to provide a basic framework for torque control. Finding the optimal controller is beyond the scope of this thesis.

The basic idea of the controller is that the desired torque $T_{s,des}$ is tracked by the deflection of the spring. Recalling the kinematic equations from chapter 3 it is know that $T_s = c_s \cdot q_s$ assuming a linear spring. The spring design in chapter 4 shows that the spring has non-linear behaviour to an extent in which it is not reasonable to approximate it with a linear stiffness. Therefore, the spring is modelled is a third order polynomial $F_s(x_s)$ function obtained from fitting the data from force/travel experiments explained in the next chapter. The support torque now becomes

$$T_s(q_s) = r_2 F_s(x_s) \quad \text{where} \quad x_s = q_s r_2 \quad (5.2)$$

Since q_s depends on the current length of the actuator $L_a(x_a)$ and the length of the actuator given that the spring is not being deflected $L_n(q_j)$ (neutral length), the support torque can be written as a function of the stroke deviation from the neutral line Δx_n and joint angle

$$T_s(\Delta x_n, q_j) \quad \text{where} \quad \Delta x_n(x_a, q_j) = L_a(x_a) - L_n(q_j) \quad (5.3)$$

Since in the normal range of usage $T_s(\Delta x_n, q_j)$ is monotonic, there exists an function $\Delta x_n(T_s, q_j)$ which a good target for the controller, since the goal is to determine the stroke deviation given a desired torque, hence $\Delta x_{n,des}(T_{s,des}, q_j)$.

As mentioned earlier, the maxon EPOS3 controllers have several control modes including cyclic synchronous positioning (CSP) and cyclic synchronous torque (CST) mode. Initially the CSP mode was chosen, but due to many errors (a lot of the time of unknown origin) and trouble with tuning the gains, the switch is made to CST mode. The firmware guide of the controller indicated that CST mode basically controls the current over the motor windings, since it relates to the torque setpoint linearly. To avoid confusion with the overall (joint) torque controller the EPOS3 controller will be referred as a current controller.

The layout of the controller implemented in Simulink for one leg is shown in fig. 5.6. The outputs of the I/O module and controller are the raw signals of the joint angle $q_{j,raw}$ in absolute counts and rotor rotations $q_{m,raw}$ in relative quad counts which can be converted to radians $q_{j,abs}$ and millimetres $x_{m,rel}$, respectively. The former only has to be calibrated once by finding a reference angle $q_{j,ref}$ (since it's an absolute sensor), but the latter needs calibration every time the device is powered on. Since the absolute joint angle is available, it can be used to determine the neutral stroke from function $x_n(q_j)$ which will be experimentally determined in the next chapter. If at any moment in time the orthosis is held such a position that there is no spring deflection, thus $x_a = x_n$, that value of x_n held (hence the **H** in the summation point) added to $x_{m,ref}$ which gives the absolute x_a . When the device is in motion and the spring is deformed, thus $x_a \neq x_n$, then the stroke deviation Δx_n can be obtained by subtracting x_n from x_a again.

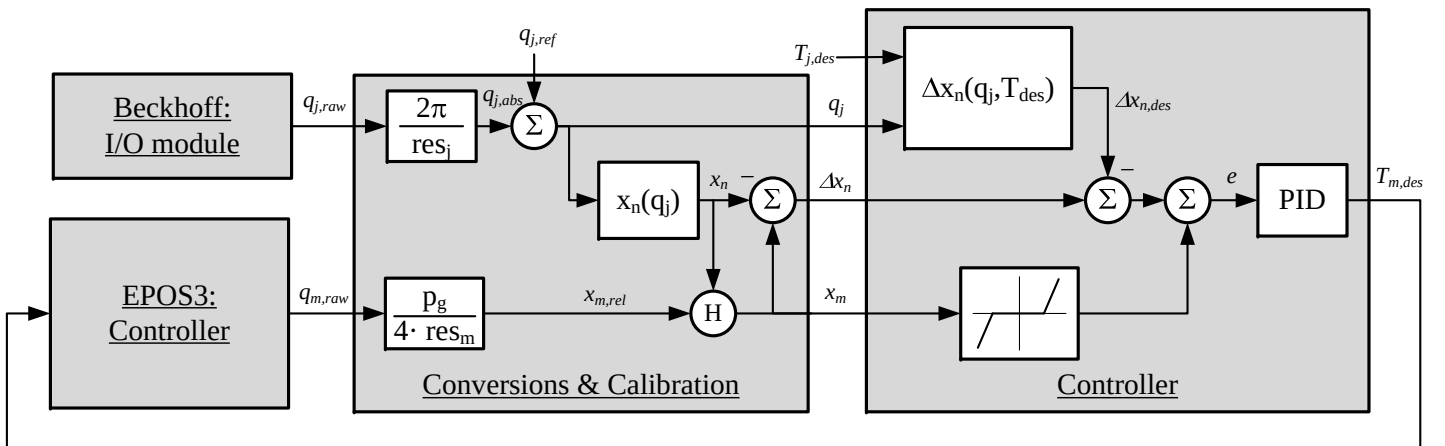


Figure 5.6: Layout of the joint torque controller

From the joint angle and the desired torque, the desired stroke deviation can be obtained from the function $\Delta x_{n,des}(T_{des}, q_j)$. Now, error is given by

$$e = \Delta x_n - \Delta x_{n,des} + x_{break} \quad (5.4)$$

where x_{break} is 0 unless x_a approaches the physical limits of actuator in which case it starts counteracting (breaking) to avoid hitting the physical end-stops with a large impact. The error signal enters a standard discrete PID controller block which outputs the desired motor torque $T_{m,des}$ to the current controller. The parameters of the PID controller will be discussed in the next chapter.

The EPOS3 controller in CST mode, needs a set of parameters which include the motor and motor-encoder properties and the P and D gain of the current control. The former is a subset of the motor properties found in chapter 3, the latter will also be discussed in the next chapter.

5.4 Conclusion

This chapter introduces a mobile hardware and software platform to perform real-time experiments with a powered ankle foot orthosis. The specifications of the chosen motor controllers, I/O module, and computer are presented, and the battery packs including management to power those components were determined. After showing how all those components are integrated into a hardshell backpack, the software needed to establish EtherCAT communication and create a link to Simulink introduces. Finally, an implementation of a torque controller including end-stop protection is discussed.

6.1. Introduction	.60
6.2. Manufacturing and Assembly	.60
6.3. Experiments	.62
6.4. Conclusion	.68

Now the whole design is finalized, we can order stock components and create technical drawings to fabricate the custom components at specialized companies. After arrival of all hardware; assembling, wiring and powering the device are last steps before the testing can start.

This chapter reveals the realized design and discusses the initial performance tests as-well-as the basic comfort and strength assessment of the device.

6.1 Introduction

The design process resulted in a light weight high-performance device, but – up to now – this is all theoretical. To validate this theory, all hardware is ordered and produced and assembled to make a fully functional prototype which was subjected to a performance assessment. First, a few tests – necessary for getting the devices operational – are performed: Experimentally determining the stiffness of the spring and the kinematics of neutral stroke x_n of the actuator. Based on this information, the controller discussed in the previous chapter are finalized and implemented. The controller gains for the lower level (current gains) and higher level (positioning gains) are tuned to obtain fast and stable control. Finally, some simple position and torque following experiments are done to test the bandwidth of the system.

6.2 Manufacturing and Assembly

In chapter 4 a fully detailed 3D model including all attachment material and sensors is created. To translate the model into actual components the following steps are necessary:

1. Create an overview of all custom and of-the-shelf parts in the model (see the list in appendix appendix A.2). All of-the-shelf components are ordered, taking into account the products lead times. Custom parts needed additional work (see next step).
2. For the custom parts, technical drawings are created, including all fits and tolerances (see appendix A.3). Because the design contains groups of specialized products that needed work from several suppliers, the manufacturing process need proper coordination to get the right parts at the right location at the right time.
3. When all parts arrived, the device could be assembled including securing components with glue and some final adjustments to a few components.

Figure 6.1 shows some of the intermediate steps of the manufacturing process. The ergonomic shell is created from an mould of the authors leg. These type of plastic shell structures aren't very accurate geometrically, but the actuation parameters r_1 , r_2 and ψ need reasonable accuracy. To solve this issue, some custom alignments tools are used as shown in fig. 6.1a.

A plastic sheet was vacuum formed around the mould while the alignment tools guaranteed the right placement of the interfacing areas as shown in fig. 6.1b. After completing a first set of the components for the right foot, a test version of the device shown in fig. 6.1c is made. With this version, a simple fitting and movement test is done on 10 students from the department whose length and weight fell in the aimed range. This resulted in some adjustments (e.g. neutralizing some shapes and adding some offsets) for the final model for both sides.

After receiving the final version of all components and fully assembling and wiring the device including the electronic hardware and backpack, the device was ready to test. The measured weight of the orthosis is 1.48 *kg* and the weight of the backpack including the controller, computer, batteries and accessories is 5.2 *kg*.



(a) Mould including alignment tools



(b) Plastic vacuum forming



(c) Test version

Figure 6.1: Picture of several stages of production

Figure 6.2 shows a photo of the fixed test set-up used to implement control framework discussed in chapter 5 and do performance tests – starting from the low risk test (e.g. sensor calibration) and continuing to higher risk tests (e.g. bandwidth).

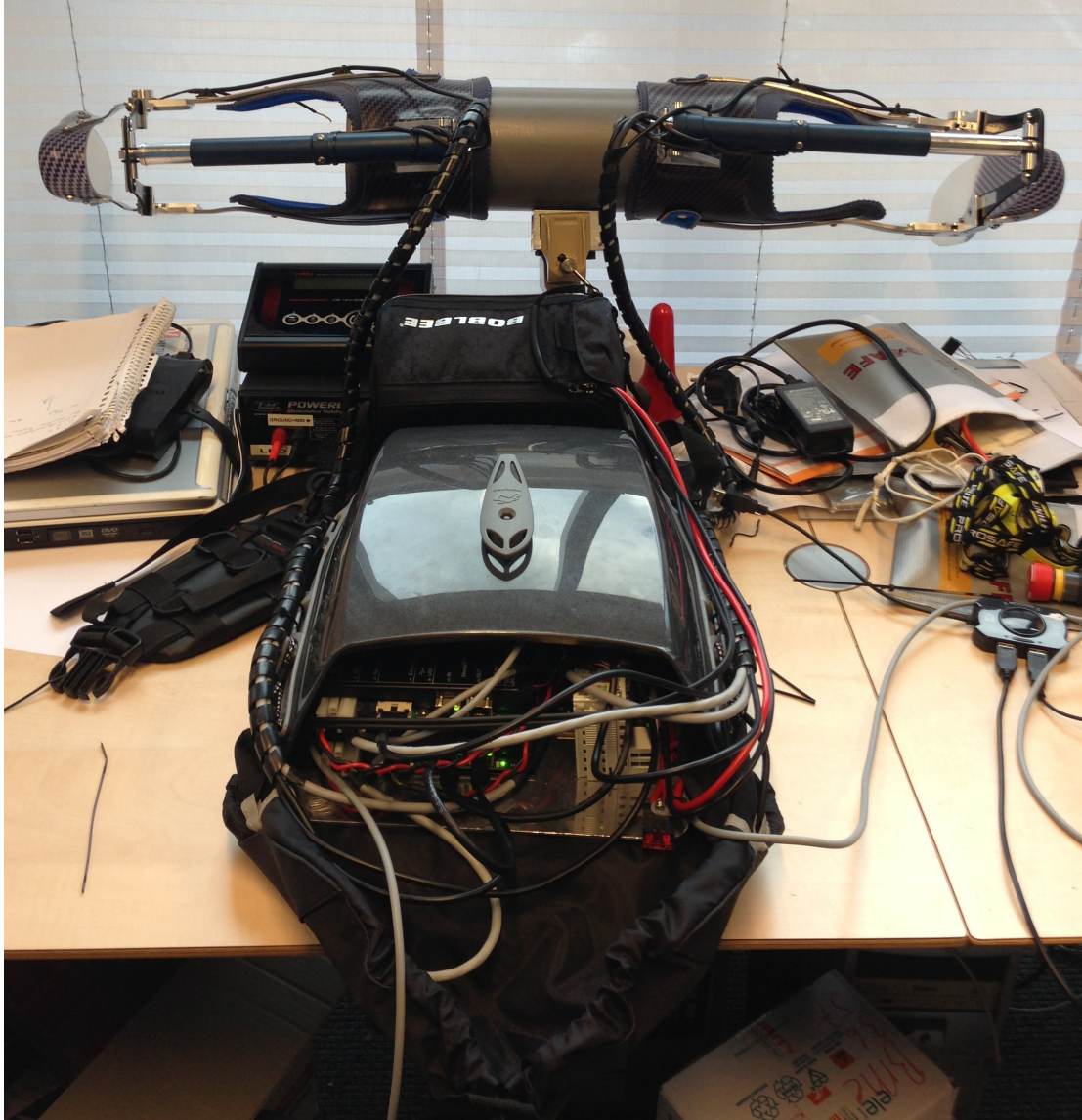


Figure 6.2: Fully wired test set-up

6.3 Experiments

The fully functional set-up described in the previous section, provided an easy test platform because of the Simulink interface. Two types of experiments will be discussed in this section:

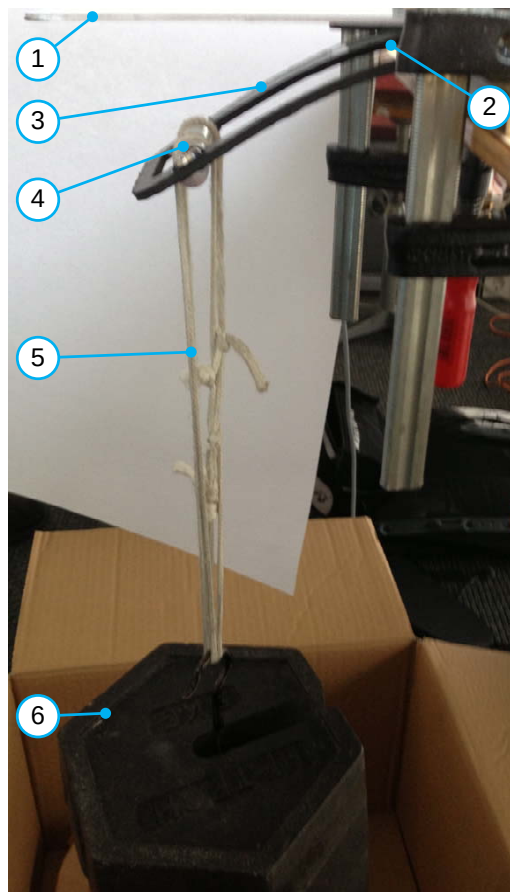
- Identification: Some of the parameter values determined in earlier chapters might deviate from their predicted values. Identification experiments are needed to check these values and use those values in the controller implementation.

- Tuning: Finding the right gains for the internal current controller of the EPOS3 and the torque controller in Simulink.
- Performance assessment: Earlier chapters also made claims on the performance of the device (following behaviour etc.). The claims will also be verified with data from experiments.

These experiments will reveal if the device meets its requirements and therefore conclude this thesis.

Spring stiffness

The spring design discussed in section 4.4, determined the stiffness of the spring using a finite element approach. Because the behaviour of the composite materials is hard to predict, the springs are made in three thicknesses. An additional uncertainty is the spring thickness itself because the production method – compressing layers of fibres under heat in an autoclave – has limited accuracy. Therefore the experimental set-up shown in fig. 6.3 is used to assess the deflection of the spring given a certain load.



- | | | | |
|---|-----------------------|---|----------------------|
| 1 | Stiff reference plate | 4 | Endpoint Shaft |
| 2 | Base clamp | 5 | Dyneema cable |
| 3 | Leaf spring | 6 | Platform for weights |

Figure 6.3: Experimental set-up for measuring the spring stiffness. The spring deflection is measured by increasing the load on the spring (6) and measuring the distance between the reference plate (1) and endpoint shaft (4) with a digital calliper.

Figure 6.4, shows the experimental data points and the 3rd order polynomial fit for each spring including the theoretical curves from 4.9. Note that the thickest (2.0 mm) and thinnest (1.6 mm) spring are stiffer and weaker than predicted, but the middle thickness (1.8 mm) almost exactly agrees with the theoretical curve. What causes these differences is hard to determine, but part of the cause could be the aforementioned production accuracy.

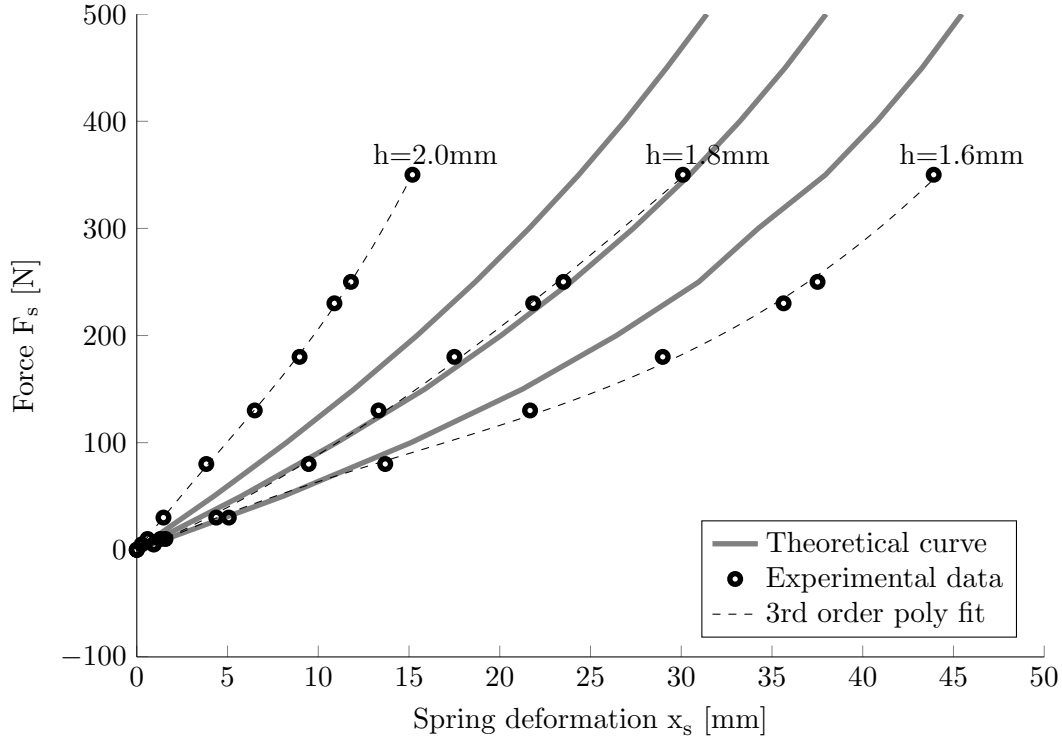


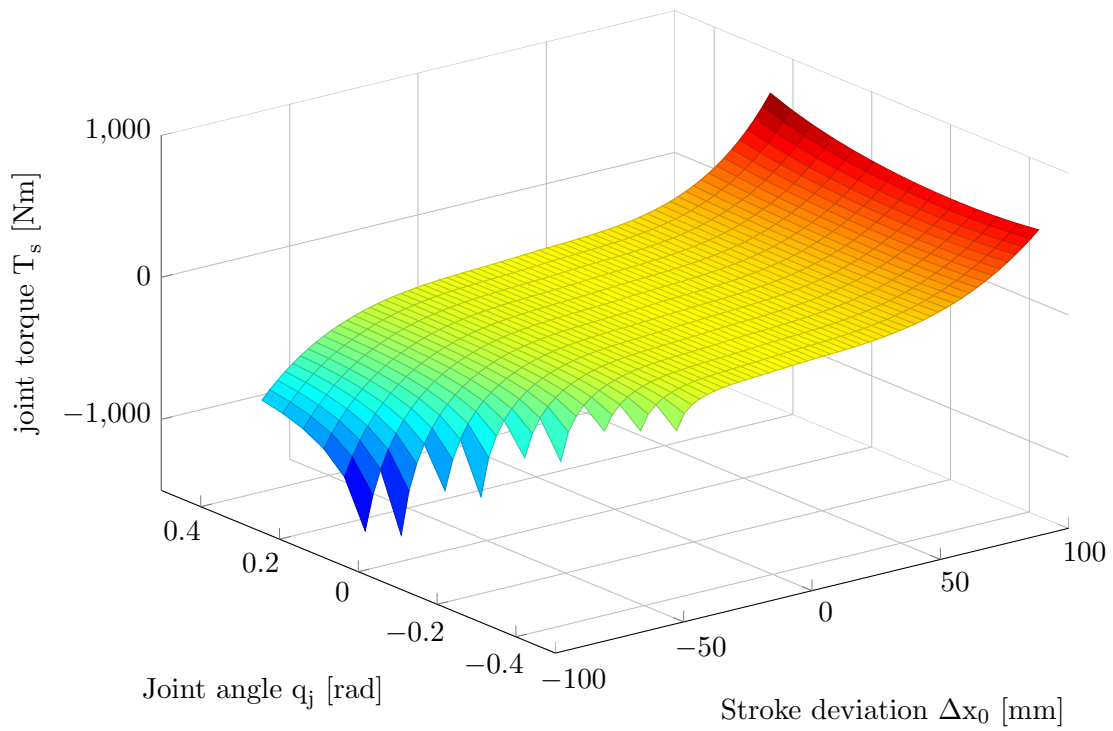
Figure 6.4: Test results of the stiffness experiments of the three types springs.

Now, the experimentally determined polynomials $\tilde{F}_s(x_s)$ for each spring could be used to complete the torque controller (see $F_s(x_s)$ in eq. (3.14)). The resulting support torque function $T_s(\Delta x_n, q_j)$ for $\Delta x_n \in [-100, 100]$ mm and $q_j \in [-25, 15]$ deg is shown in fig. 6.5a. The inverse function $\Delta x_n(T_s, q_j)$ for $T_s \in [-100, 100]$ Nm is shown in fig. 6.5b.

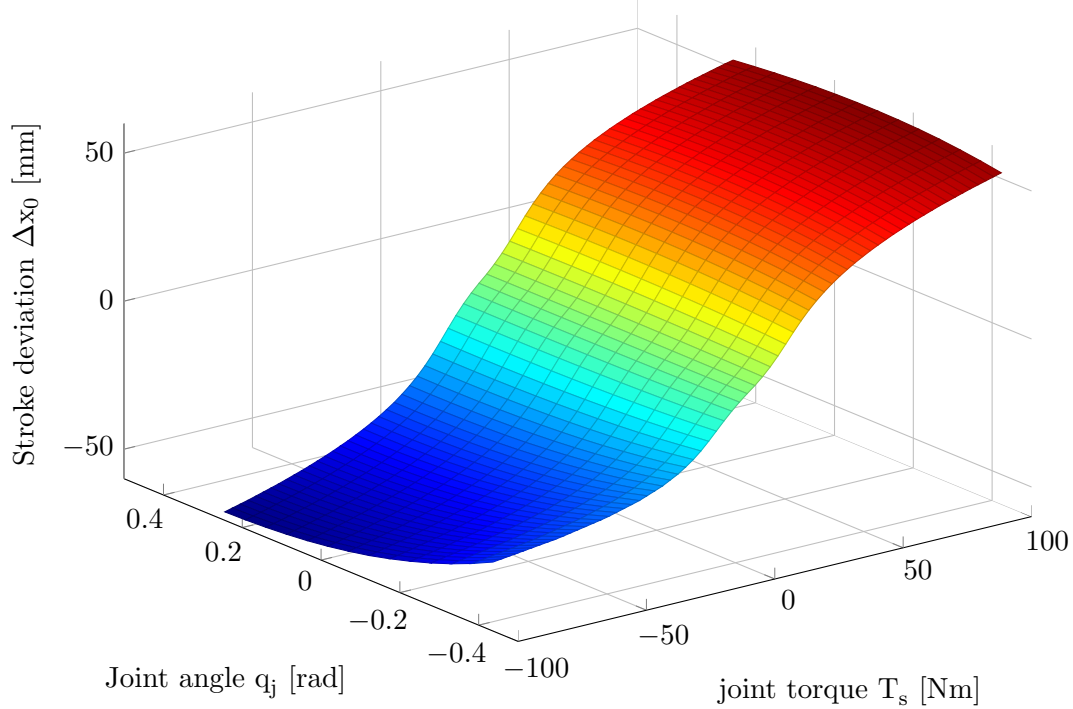
Neutral stroke kinematics

The second parameter that has to be experimentally determined is the neutral stroke of the actuator x_n needed for calibration of the controller. The method used to obtain this parameters was manually moving the actuator through its full range of motion while recording the actuator and joint positions. During this test, the spring needs to stay as neutral as possible. Since the friction and mass around the ankle joint was low with respect to the spring stiffness, this could be done with reasonable accuracy without special countermeasures. Data from tests of the left and right PAFO is shown in fig. 6.6a and fig. 6.6b including the third order polynomial fit and theoretical curve. Note that the theoretical curve and fit polynomial closely follow each other, hence the geometry of the device is as predicted.

Given the experimentally determined polynomial $\tilde{x}_n$, the torque controller could now be implemented.

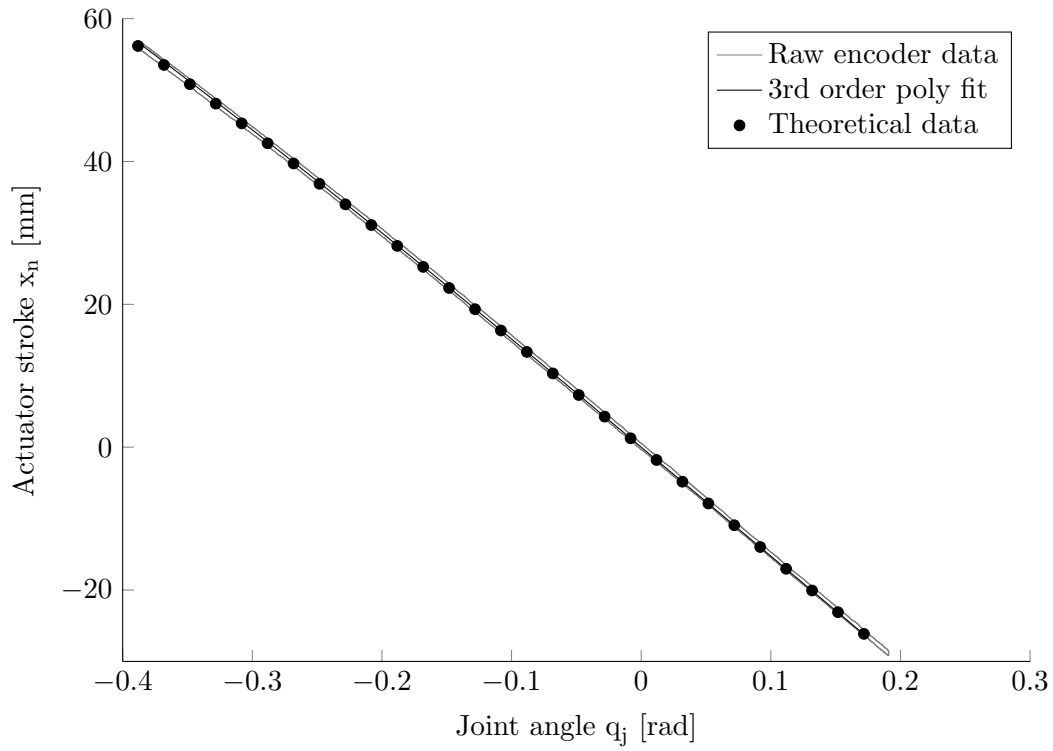


(a) Resulting support torque $T_s(\Delta x_n, q_j)$

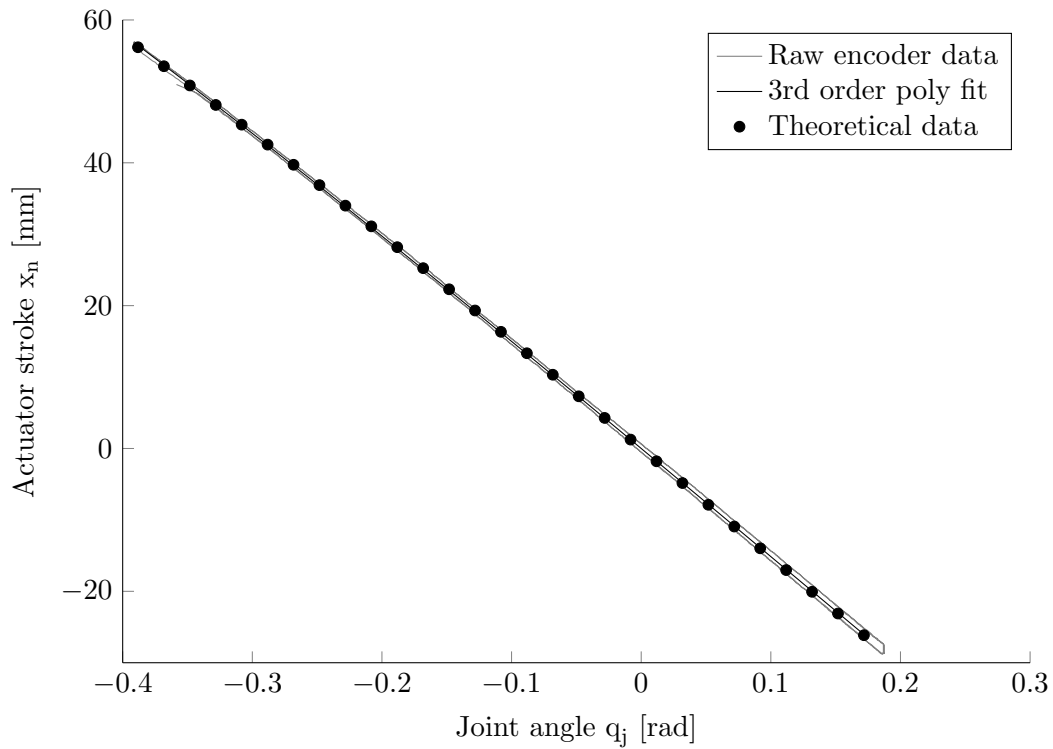


(b) Desired deviation from neutral stroke $\Delta x_n(T_s, q_j)$

Figure 6.5: Relations between the motor stroke (x_m) ad function of the joint angle (q_j) and support torque (T_s) for spring thickness $t = 1.6\text{mm}$.



(a) left leg ($\tilde{x}_n = 25.9q_j^3 - 5.9q_j^2 - 152.2q_j$)



(b) Right leg ($\tilde{x}_n = 17.0q_j^3 - 7.4q_j^2 - 150.8q_j$)

Figure 6.6: Motor stroke (x_n) as function of the joint angle (q_j) for zero torque in the spring.

Regulator tuning

The inner control loop – which is embedded in the maxon EPOS3 controller – regulates the current over the windings. Maxon provides software called *EPOS Studio* to tune the P and I gains based on a step response and store them in the controller. This method is used to obtain the gains for both PAFOs while in the fixed set-up, as shown in fig. 6.2.

The outer control loop concerns the joint torque controller discussed in section 5.3. It regulates the error e defined in eq. (5.4) using a Simulink discrete PID controller, hence

$$u(z) = \left(K_p + K_i T_s \frac{1}{z-1} + K_d T_s \frac{c_f}{1 + c_f T_s \frac{1}{z-1}} \right) e(z) \quad (6.1)$$

where K_p , K_i and K_d are the proportional, integral and derivative gains, respectively, T_s the sample time and c_f filter coefficient for the error derivative.

To get an indication of the proper gains for this controller, an experimental method by Ziegler and Nichols [39] is used. This method is based on finding the *ultimate gain* K_u , the gain that results in a step response oscillation of constant frequency, and T_u the period of the oscillation. The gains found for our system based on a 5 mm step response are $K_u = 100$ and $T_u = 0.1572$ s. Now, the controller gains can be found as proportions of the ultimate parameters [40]. For a controller with “Some overshoot” these are:

$$K_p = \frac{K_u}{3}, \quad K_i = \frac{2K_p}{T_u}, \quad K_d = \frac{K_p T_u}{3} \quad (6.2)$$

Tuning the gains of the torque controller requires an additional set-up where the ankle joint is rigid and the actuator only interacts with the spring. Because of the limited time frame, the choice is to use the available fixed set-up (see fig. 6.2) that allows free motion of the ankle to get an idea of the following behaviour of the system. The controller is running at a sample time of 1 ms and the filter coefficient is chosen 25. The resulting gains are used for all following tests.

Bandwidth testing

To test the performance of the controller, the crossover frequency f_c , Phase margin PM and gain margin GM are determined from a frequency response experiment on the actuator length. Figure 6.7 shows the Bode plot of a frequency response from a stroke set-point $x_{m,set}$ to the actual stroke x_m using a chirp signal with an amplitude of 2 mm and a frequency increasing from 0.1 to 30 Hz. The bandwidth of the system is 5.67 Hz with a phase margin of 49.52 deg and gain margin of 14 dB. A video of this test can be found on the CD in appendix A.4. Preliminary test using more aggressive gains showed better bandwidth with higher resonance peaks, hence there is room for improvement.

Additional tests

In addition to formal performance tests, some preliminary motion and walking test are done. These tests indicated that for free ankle movements with the PAFO in zero torque mode ($T_{j,des} = 0$) the controller performs nicely. For walking in zero torque mode, however, the controller started oscillating at loading response (right after heel strike). Therefore, these tests didn't produce any presentable data, but video material of these trails is available on the photos and videos section in appendix A.4. The videos show that the device can operate fully autonomously.

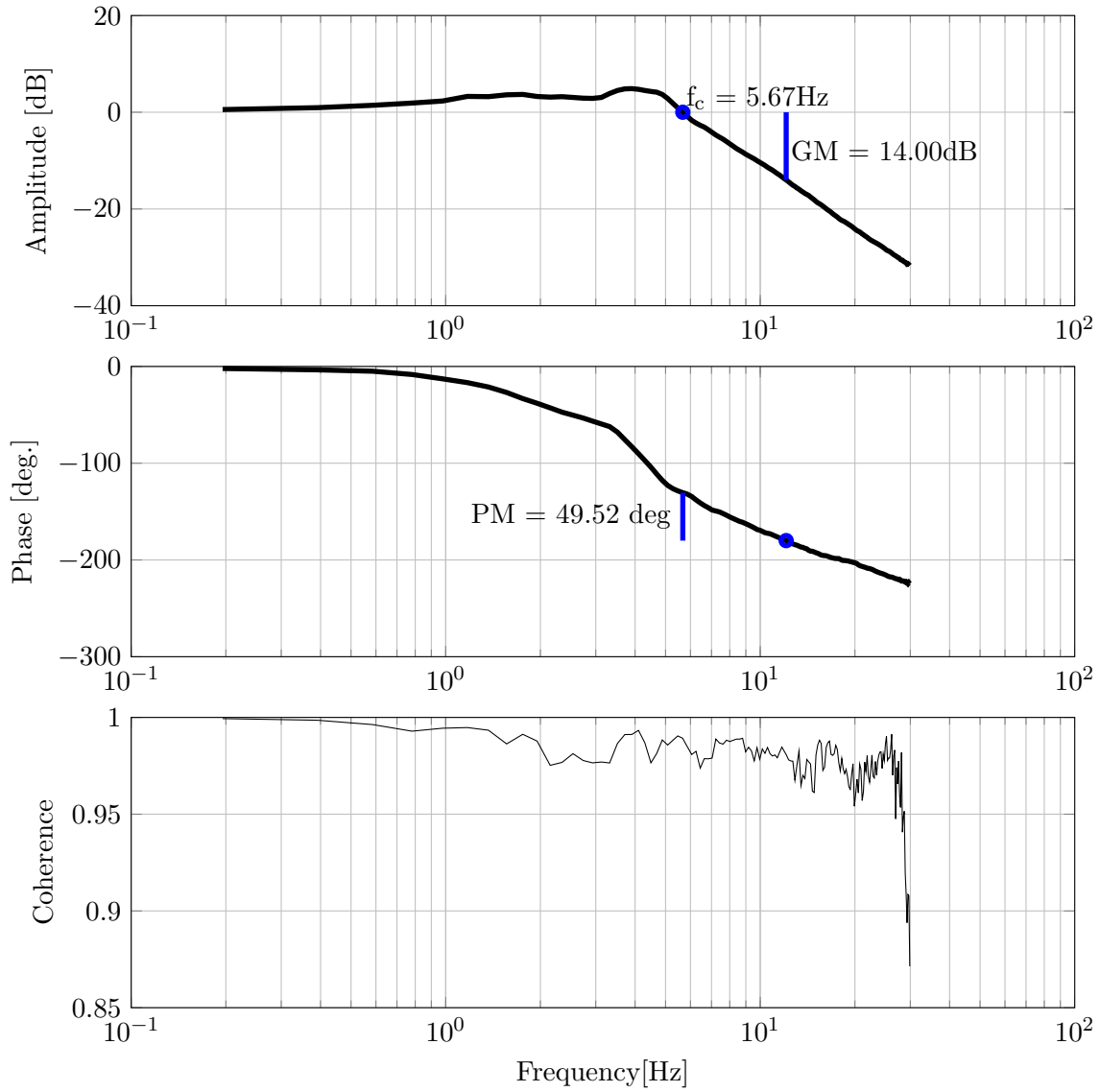


Figure 6.7: Bode diagram of the frequency response functions of the motor stroke

6.4 Conclusion

This chapter discusses the process from a detailed design to a fully functional prototype. Next, the some identification experiments are presented that resulted in completing the joint torque controller. Finally, the results of initial performance tests are revealed and discussed.

These final steps result in a full functional lightweight autonomous actuated orthosis. The controller performance in preliminary test show that the controller needs additional work to provide a stable and fast control. Fortunately, lots of options are still available to improve the control parameters. One of the options is revisiting the rule-based gain tuning approach and more thoroughly investigate the best method of finding the gains for this type of system. A alternative approach is to add feed-forward elements to the control framework. Since the human gait pattern is cyclic, hence predictable, feed-forward control (e.g. based on adaptive frequency oscillators [41]) is also a feasible possibility. Finally, the delays caused by the hardware and software set-up should be further investigated since they can also play an important role in the controller performance.

The first chapter of this thesis has set the challenge, scope, and requirements for the design of a state of the art autonomous powered ankle foot orthosis that supports ankle plantarflexion. The goal is not only to create a design, but also realize the device and assess its performance.

The second chapter gives a brief overview of the functions needed for the PAFO and how these functions are grouped into systems. Next, the sub-solutions that can fulfil these functions and the choices that led to a final design concept is discussed. There will be a clear distinction between systems that will be designed with a pragmatical approach (control, sensors and communication) and systems that will be thoroughly analysed and optimized (actuation, human interface and structure).

The third chapter introduced a dynamic model of a linear actuator capable of generating plantarflexion power, using an electric motor, ball-screw gear and a series elastic element integrated in a lever arm. By adding constraints that model the limitations of the drive components and a cost function that rewards a concentrated power burst at push-off, a standard minimization problem could be formulated. Running this optimization for combinations of 4 potential motors and 4 gears, resulted in a trade-off between performance and weight, from which – given some interpretation – the best option could be picked. The optimization showed that series elastic elements can drastically improve the power output of the device.

The fourth chapter reports all the details of the orthosis design and the resulting ready to produce 3D model. The ultimate goal is to construct a safe and lightweight device that can be comfortably fit around the foot and lower leg. To ensure the users safety several design checks are done on the components maintaining a safety factor 1.5. The rotating parts of the actuator we fully covered by the structure. To keep the added mass of the device to a minimum, the amount of material used for each custom part is kept minimal. For the spring mechanism, composite materials are chosen to reduce the mass further. To get some insight in what assemblies and component groups add most mass to the device, the mass of each part is obtained from the mass properties in SolidWorks. The mass contribution of the assemblies and component show that the shells and reinforcements of the shank and foot are large contributors. Applying composite materials for these components could further reduce the mass in future designs. The total mass adds up to a theoretical 1.45 *kg*.

The fifth chapter introduces a mobile hardware and software platform to perform real-time control for experiments with the powered ankle foot orthosis. The specifications of the chosen motor controllers, I/O module, and computer are presented, and the battery packs including management to power those components are determined. All these

components were integrated in a hardshell backpack. The software needed to establish communication with EtherCAT and Simulink is described. Finally, an implementation of a torque controller including end-stop protection is discussed.

This sixth chapter discusses the process from a detailed design to a fully functional prototype. Next, the some identification experiments are presented that resulted in completing the joint torque controller. Finally, the results of initial performance tests are revealed and discussed.

These final steps result in a full functional lightweight autonomous actuated orthosis. A preliminary test of the controller shows that the controller needs additional work to provide a stable and fast control. Fortunately, lots of options are still available to improve the control.

Revisiting the goals of this thesis we can conclude that: The design of a lightweight actuated orthosis was created, manufacturing and assembling the design was done, designing and implementing a control algorithm was accomplished, and functional tests of the prototype were performed. The resulting ankle-foot orthosis is powered, autonomous and the lightest currently build. We hope that further research with our orthosis will increase the quality of live of patients currently living with a walking impairment.



Figure 7.1: Photo of the actuator

Acknowledgement

This thesis required a very multidisciplinary skill-set to complete, but luckily I had a lot of support from experts in each field. To them I dedicate these closing credits:

Overall guidance

Wietse van Dijk & Herman van der Kooij

Mechanical design

Jan van Frankenhuijzen

Software

Eelko van Breda

Electronics

Guus Liqui Lung

Control

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Andries Oort & Nisse Linskens & Hans Drop

Orthotics

Niek Wondergem & Joost van Leeuwen & Cees Kribbe

Composite materials

Marcel Gilles & Bas Nijpels

Spelling and Grammar

Pim Meijneke

Finally, for Wietse van Dijk, simply mentioning him in the credits above does not justify the involvement he had and labour he put into supervising my thesis. Without your dedication to this project I wouldn't have accomplished so much in this small time frame. Thank you!

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η_g	Ball-screw gear pitch	$[-]$
ω_{max}	Motor maximal permissible speed	$[W]$
ψ	Initial lever-arm angle	$[rad]$
γ	Angle between lever-arm and angle	$[rad]$
ρ	Material density	$[kg/m^3]$
σ	Material bending strength	$[N/m^2]$
b_{eq}	Equivalent damping coefficient	$[Ns/m]$
c_s	Rotational spring stiffness w.r.t. the ankle joint	$[Nm/rad]$
c_f	Filter coefficient	$[-]$
E	Modules of elasticity	$[N/m^2]$
e	Controller error	$[m]$
E_{cu}	Dissipated energy from copper losses	$[J]$
E_{mech}	Injected mechanical energy	$[J]$
$\tilde{F}_s$	Experimentally determined spring force	$[N]$
f	Normalized residual torque or power exerted by the user	$[-]$
F_a	Force in the actuator	$[N]$
f_c	Crossover frequency	$[-]$
F_d	Force from friction	$[N]$
F_m	Force from the motor magnets	$[N]$
F_{int}	Gear maximal permissible intermittent force	$[N]$
g	Vector with motor constraints	$\dots$
GM	Gain margin	$[dB]$

h	Thickness of the spring	$[m]$
I_m	Motor current through windings	$[A]$
I_{nom}	Motor nominal current	$[A]$
J_g	Ball-screw spindle inertia	$[kgm^2]$
J_m	Rotor inertia	$[kgm^2]$
K_m	Motor constant	$[Nm/\sqrt{Nm/s}]$
K_t	Motor torque constant	$[Nm/A]$
K_d	Derivative gain	$[-]$
K_i	Integrator gain	$[-]$
K_p	Proportional gain	$[-]$
k_s	Linear stiffness of the spring	$[N/m]$
L_0	Length of the actuator at $q_j = 0$ and $q_s = 0$	$[m]$
L_a	Length of the actuator	$[m]$
L_n	Length of the actuator for $q_s = 0$	$[m]$
L_{max}	Actuator maximal length	$[m]$
$\tilde{m}_{g2}$	Specific ball-screw spindle mass	$[kg/m]$
m_m	Motor mass	$[kg]$
M_{eq}	Equivalent mass	$[kg]$
m_{g1}	Ball-screw nut mass	$[kg]$
m_{g2}	Ball-screw spindle mass	$[kg]$
m_{tot}	Total mass of motor/gear combination	$[kg]$
p_g	Ball-screw gear pitch	$[m]$
P_{cu}	Power from copper losses	$[W]$
P_{el}	Motor electrical power	$[W]$
P_{max}	Motor maximal permissible power	$[W]$
P_{mech}	Motor mechanical power	$[W]$
PM	Phase margin	$[deg]$
q_a	Angle of the deformed lever-arm	$[rad]$
q_j	Recorded ankle joint rotation	$[rad]$
q_m	Motor rotor position	$[rad]$

q_n	Angle of the undeformed lever-arm (neutral line)	$[rad]$
q_s	Deflection angle of the spring	$[rad]$
q_s	Deflection angle of the spring	$[rad]$
r_1	Proximal lever-arm length	$[m]$
r_2	Distal lever-arm length	$[m]$
R_m	Motor winding resistance	$[V]$
R_t	Ball-screw gear transmission ratio (Force)	$[m]$
R_x	Ball-screw gear transmission ratio (Travel)	$[m]$
r_{eff}	Effective lever-arm length	$[m]$
$\bar{S}$	Specific energy density	...
T_j	Recorded ankle joint torque	$[Nm]$
T_m	Motor torque	$[Nm]$
T_s	Torque of the deformed spring w.r.t. the ankle joint	$[Nm]$
t_{cycle}	Total cycle time	$[s]$
t_{on}	Time above nominal	$[s]$
T_s	Sample time	$[s]$
U_m	Motor voltage over the windings	$[V]$
U_{max}	Motor maximal permissible winding voltage	$[V]$
v_{max}	Gear maximal permissible velocity	$[m/s]$
w	Width of the spring	$[m]$
$\bar{x}_m$	Actuator stroke vector	$[m]$
Δx_n	Stroke deviation	$[m]$
$\tilde{x}_n$	Experimentally determined neutral stroke	$[m]$
x_a	Actuator stroke	$[m]$
x_n	Neutral stroke	$[m]$
z	Set of optimization parameters	$[-]$

A

Multimedia CD

A.1. Literature Survey	81
A.2. Parts list	81
A.3. Technical Drawings	81
A.4. Photos and videos	81

This appendix explains the content of the multimedia CD included with this document (next page). First of all a digital version of the report can be found in the root.

A.1 Literature Survey

The folder A1_LiteratureReview contains the literature report made before the start of the thesis.

A.2 Parts list

The folder A2_PartsList contains a list with all the parts and a 3D pdf file with the assembly.

A.3 Technical Drawings

The folder A3_TechnicalDrawings contains a few pdf binders (one per supplier) with technical drawings of all custom made parts.

A.4 Photos and videos

The folder A4_PhotosAndVideos contains videos of the various tests that are performed. Additionally, some photo of the author wearing the device are present in this folder.