

Design and Study of a New Continuously Variable Transmission with Potential Application to Lower Limb Prosthesis

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Abstract

Series elastic actuators (SEA) are very popular in rehabilitation robotics. Among other advantages, elastic elements between the actuator and the load permit to store and release energy during the task completion, such that the energy balance is improved and the motor power peak is decreased. In rhythmic tasks like walking, this reduces to design the spring stiffness such that it works at resonance. To comply with different gaits and cadences, it is therefore necessary to design Variable Stiffness Actuators (VSA).

The first part of the thesis investigates lower limb prostheses concepts. A particular concept of VSA is applied to active transtibial and transfemoral prostheses. The model of an original actuation concept is reported, relying on the combination of a power motor, a compliant element (a spring), a mechanical differential, and Continuously Variable Transmissions (i.e. a mechanical transmission whose ratio can be continuously varied). It allows to manage the mechanical power flows through the device in both directions (i.e. when energy should be produced or absorbed by the knee and/or ankle), so that the power motor does not face the sharp load power fluctuations. A preliminary approach to synthesize a closed-loop controller for this device, and simulation results of this closed-loop behavior are reported. These results illustrate the capacity of this actuation principle to filter the load power profile, and further highlight the necessity to maximize the mechanical efficiency of each part of this actuation scheme.

The second part investigates a new kind of Continuously Variable Transmissions (CVT) principle which can achieve ratio variations at rest, i.e. with zero velocity of the shaft. This feature is mandatory for the concept developed in the first part. The presented design is a modified planetary gear, whose planets are conical and mounted on inclined shafts, and ring is made of contiguous diabolo-shaped rollers. This configuration allows moving the contact point radius on the cones and then modifying the transmission ratio. Importantly, this movement relies on rolling and not on sliding, such that it requires virtually no actuation energy. Two prototypes have been

designed and experiments were conducted and showed that the CVT transmission ratio can indeed continuously vary by keeping an efficiency between 70 % and 90 %. This proved the possibility to embed the design in a transfemoral prosthesis.

In sum, this thesis deals about a new design of Continuously Variable Transmission allowing ratio variations at rest. This design aimed to be integrated in a new actuation system for transfemoral prosthesis.

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Introduction

1

1.1 Context

Human locomotion is a complex task that is influenced by different factors. Bones, muscles, and tendons provide the mechanical structure and actuation, the central nervous system and the spinal reflex mechanisms provide inputs to create motion patterns like walking and running. Human locomotion is therefore achieved by complex interactions between biomechanical tissues and neural controls.

In 2012, more than 1 million lower limb amputations were performed worldwide. This approximately corresponds to one every 30 seconds [1]. These amputations were due to peripheral vascular diseases (66 %, a majority being due to complications of diabetes), traumatic events (26 %), tumors (5 %) and congenital diseases (3 %) [117].

There is an expected increase of 47 % in lower limb amputations from 1995 to 2020 [1]. Indeed, half of lower limb amputations are performed on people who have diabetes, while the number of patients suffering diabetes is expected to increase from 285 million (in 2009) to 435 million by 2030 [53].

On top of that, amputation has been recognized as one of worldwide concern involving social, economic, industrial and psychological problems and affecting many more people than those actually suffering the amputation [111]. All nations are involved - some more than others - and all seek information, guidance, and advice on how to treat an amputation. It is therefore critical to find solutions that will give back the amputee his/her missing limb(s) functionalities.

For centuries, scientists have tried to develop prosthetic solutions closer and closer to the ideal one. Until the beginning of the 80s, the majority of prostheses used to replace the amputee's lower limb(s) were stiff structures. Since then, major changes in materials and structure improved the functioning of passive lower limb prostheses. A critical breakthrough was the development of elastic carbon feet for

ankle-foot replacement. These carbon blades behave like a leaf spring. They can store and then release energy during push-off, i.e. the walking phase where the ankle propels the body forward.

However, many researches [25, 45, 85, 102] showed that transtibial amputees using passive or quasi-passive ankle prosthesis still experience gait disorders such as higher metabolic demands, larger kinematic and kinetic asymmetries and reduced self-selected walking speeds [42, 47]. This is partly explained by the fact that these prostheses are unable to produce energy. Indeed, the sound ankle muscles produce on average 540 % more work than they store during walking [108]. To compensate for the absence of these muscles, external positive energy is hence required. To achieve that, an actuator is needed. Unlike the ankle, the knee absorbs energy (i.e. negative net work) during level-ground walking [108]. To compensate for the absence of these muscles, external brake or dissipative systems are required. However, for other tasks, like stair climbing, the knee produces energy [78].

Today researchers are developing prostheses that are capable of better mimicking the healthy joints behaviour thanks to the addition of motors and external power. These motors inject or dissipate the needed amount of energy. Such devices are called active prostheses. Developing such technology brings various challenges. Indeed, in addition to challenges related to mechanical design, scientists have to find control algorithms that emulate a non-pathological walking gait.

1.2 Objectives

This thesis aimed to analyse and improve the energy efficiency of such prostheses. A particular objective is to increase the autonomy of the prostheses by decreasing their need in energy. To achieve this goal, different mechanical solutions with different morphologies were investigated. The central elements of each of the designs are springs. Indeed springs allow the energy storage. During negative power phases, the absorbed energy could be injected into a spring instead of being dissipated in a motor. This stored energy can then be reinjected in the joint during positive power phases. Two approaches are possible: either energy can be stored for each joint independently (i.e. each joint has its own spring and the energy transfer occurs between different phases of the gait cycle) or a single spring can store the energy for and from multiple joints. In the second case, energy transfers occur between joints and the global efficiency of the device can possibly increase.

Most of existing lower limb active prostheses use Series Elastics Actuators (SEA), i.e. the serial connection of a standard actuator with an elastic element (e.g. a spring). By reducing the interface stiffness, this concept offers a lot of advantages, including greater shock tolerance, lower reflected inertia, more accurate and stable force control, less inadvertent damage to the environment, and - last but not least - the capacity

to store and release energy [76]. Several lower limb prostheses exploit this principle like e.g. the devices of Lefeber [18, 19, 21, 34, 37], Herr [3, 4, 27, 80], Sugar [8, 48, 50] and Goldfarb [88, 90].

Moreover, it is possible to go even further by actively adapting (i.e. with a motor) the spring stiffness and use Variable Stiffness Actuators (VSA). Indeed, an intrinsic limitation of the SEA is that the mechanical stiffness of the compliant element can be optimized only for a single task and/or user, since it is fixed. Consequently, people developed in the last few years the concept of VSA. Relevant contributions in this field include the work of Lefeber et al. with *MACCEPA* [56, 99], Bicchi et al. with *VSA*, *VSA-II* and *VSA-HD* [10, 15, 16, 81], Tsagarakis et al. with *AWAS* [54, 55] and Stramigioli et al. with the bond graph study of VSA [87, 103–105]. Applying the concept of VSA to prosthesis allow therefore to adapt the actuator stiffness not only to the user, but also to the detected gait and cadence using an appropriate adaptive controller. In this case, the spring stiffness is adapted in order to maximise the energy flows (a spring efficiency is almost 100 %) and thus minimise energy flows from or to the motors. Indeed, typical electrical motors have limited efficiency, e.g. due to Joule losses. A typical high performance motor (e.g. EC-max from Maxon) has an efficiency between 80 % and 90 %.

1.3 Joint torque-angle relationships

A good knowledge of the human walking gait is essential before starting to develop active lower limb prostheses. The biomechanical data (i.e. position, velocity, torque and power) of the ankle and knee are analysed. Further details about the human walking gait are provided in Appendix A.

The torque-angle relationships are important to be analysed in the context of lower limb prosthesis design. Indeed, when torque-angle curves are drawn, different patterns can be identified in these trajectories, e.g. spring-like behavior, and positive or negative work loops. Fig. 1.1 shows these curves for the ankle and knee, i.e. the joints we focused on during this thesis. A clockwise cycle generates energy and a counter clockwise cycle absorbs energy.

During stance, the ankle displays a behavior similar to a non-linear spring (Fig. 1.1), i.e. following a banana shape. During loading and mid-stance, the ankle is absorbing energy. After maximal dorsiflexion, i.e. during terminal stance, plantarflexion occurs and the ankle is producing energy. However both curves do not overlap and make a clockwise loop. It means that energy is produced at the ankle during stance. During swing, the curve stays flat, since almost no torque is applied to the ankle. An ideal powered prosthesis should mimic this human ankle's quasistatic stiffness or the slope of the measured torque-angle curve during stance. An SEA implemented with a series stiffness close to the slope of this ankle torque-angle relationship would pro-

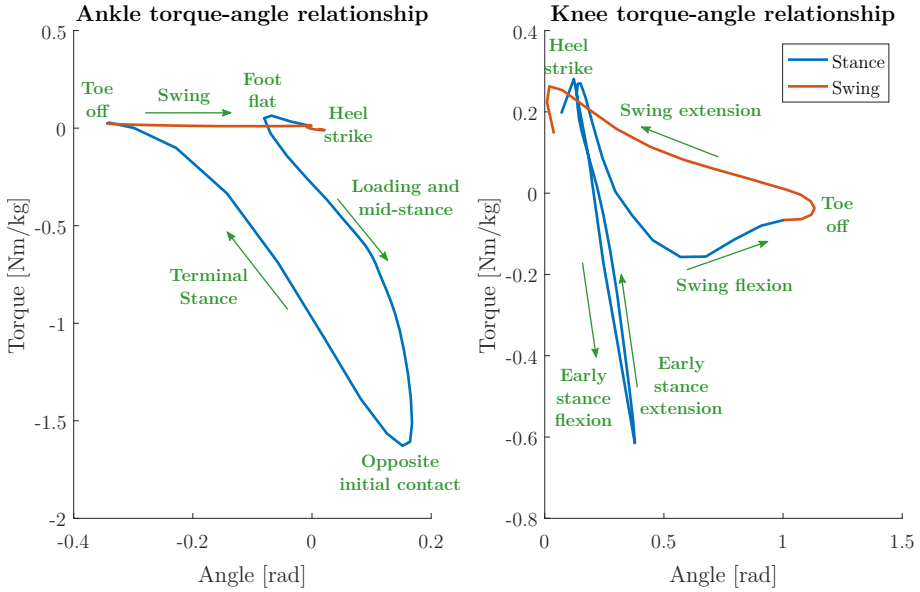


Fig. 1.1: Left: Ankle (left) and knee (right) torques are plotted versus their corresponding angle. The blue line shows the torque-angle behavior during stance, whereas the red line shows the swing phase. The gait events of heel strike, foot flat, opposite initial contact, and toe off are denoted. The area enclosed by the curve is the net work done at the joint during stance: positive for the ankle, negative for the knee. Data from [108].

vide a reduction in the mechanical work required by the motor within the SEA. In other words, the torque-angle relationship would be rendered by the physical spring almost alone requiring reactionary torque to be generated by the motor, at lower speed. The motor would just need to add the needed energy during terminal stance.

During the early stance knee flexion and extension phase, the torque-angle relationship (Fig. 1.1) is predominantly linear, or spring-like, and the dynamics can be replicated by an appropriate spring. An SEA implemented with a series stiffness would approximately reproduce the slope of this knee torque-angle relationship and would thus provide a reduction in the mechanical work required by the motor within the SEA. In other words, the complete torque-angle relationship would be rendered by the physical spring alone requiring only reactionary torque to be generated by the motor, at negligible motor speed. During swing flexion and extension, the knee is absorbing energy. Indeed, it makes a counterclockwise loop. The area enclosed by the loop on the right is the net negative work absorbed at the joint during these phases.

Moreover, since the knee is mostly absorbing energy (-0.15 J/kg during one cycle), it should be valuable to transfer this energy to the ankle which needs energy

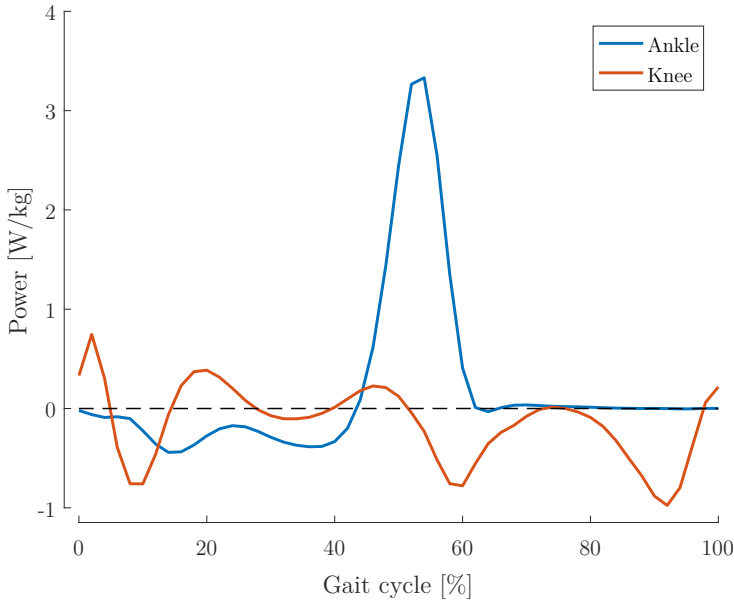


Fig. 1.2: Ankle and knee power during one gait cycle. Data from [108].

(0.21 J/kg during one cycle). However, the ankle energy need is not synchronized with the knee energy absorption during the gait cycle (see Fig. 1.2). Clever mechanisms are thus necessary to transfer energy between the joints with a good timing and reproduce healthy walking as close as possible.

This thesis focuses on the energy transfer from the knee to the ankle thanks to different mechanisms, different lower limb prosthesis concepts will thus be studied.

1.4 Research questions

This thesis targets analyse and improvement of energy efficiency in lower limb prostheses. By improving efficiency, smaller motors and smaller batteries can be used. This will reduce the cost and the weight of the prosthesis.

The main questions that we want to investigate in this thesis are the following:

- *Is it possible to design active lower limb prostheses with motor providing power flow in only one direction during walking?*

Human locomotion involves both high and low hip, knee, and ankle torques in both directions of movement (flexion/extension) within a single gait cycle. Similarly, these three joints display large variations in their angular velocity. This induces large fluctuations in the lower limb joint power profiles [74,108]. Since a power flow in a motor induces losses, there is an interest to minimise

the energy going through the motor to the minimum needed, i.e. the net work at the joint during one gait cycle. If the motor power flow changes sign, it means that the motor dissipates energy that will have to be produced again afterwards. This is therefore something that should be avoided.

- *Is it possible to decrease power and torque peaks in lower limb prosthesis motors?*

As mentioned in the previous question, during walking, large fluctuations occur in the lower limb joint power profiles [74, 108]. However, losses in an electric motor are proportional to the square of the torque. To minimise the electric energy consumption, motor power and torque profiles should be as flat as possible.

- *Is it possible to transfer energy between joints in a lower limb prosthesis?*

During level-ground walking, the ankle consumes energy while the knee absorbs energy. If efficient energy transfer between the knee and the ankle is possible, the global energy consumption of the prosthesis should decrease.

- *Is it possible to design a new mechanism improving the global energy efficiency of a lower limb prosthesis? Is it possible to make it small and lightweight enough to embed it in a lower limb prosthesis?*

Energy flows and torque peaks occurs at different phases of the human walking cycle. The knee is not absorbing energy when the ankle needs energy. New mechanisms managing and storing this energy can be designed. In this thesis, we focused on a Continuously Variable Transmission (CVT), i.e. a transmission whose ratio can be continuously varied. This CVT adapts transmission ratio between joints and springs. The challenge is to miniaturise the such a mechanism in order to embed it in a lower limb prosthesis.

1.5 Executive summary

This thesis is structured according to the publications during its proceedings. More specifically, most of the chapters of the core part of the thesis (i.e. Chapters 4, 5, 7, and 8) corresponds to one paper. Therefore, the text of these chapters is adapted from the related papers, but with the purpose to stay as close as possible to the published versions.

This thesis is divided into two parts. The first part focuses on lower limb prosthesis concepts and the second part on Continuously Variable Transmission (CVT) design. The following key thread is followed by the chapter ordering. We start with a brief introduction about human walking and a literature review on ankle and knee prostheses. After that, an ankle-knee prosthesis model with SEA and parallel springs is studied in terms of energy efficiency. The next chapters focus on a VSA based prosthesis. The design parameters of an ankle prosthesis were optimised to increase the

global energy efficiency. Next these concepts were extended to an ankle-knee prosthesis - i.e. a device for a so-called trans-femoral amputee - allowing energy transfer between joints. In particular, the VSA that we investigated is based on a CVT. In the first part of the manuscript, this CVT was considered as a black box. However, we figured out that this ideal CVT does not exist in the state-of-the-art, so that we decided to develop our own concept. Therefore, the second part of this thesis is entirely devoted at developing our own CVT technology. In particular, existing concepts were not miniaturisable enough to embed it in a prosthesis. We thus focused our design to make it embeddable in a prosthesis in terms of size, power capacity and energy efficiency. Two prototypes were developed. The first one aimed at validating the kinematic concepts, and showed the ability of the design to continuously change its transmission ratio, including when the input shaft is not rotating. The second prototype aimed at validating the torque capacity and energy efficiency of the design.

Now each chapter will be outlined. Chapter 2 overviews the existing ankle and knee prostheses. It starts with a classification of ankle prostheses describing the most recent designs. It separates passive, semi-active and active prostheses. Second part of the chapter presents most recent knee prostheses with the same classification.

Chapter 3 shows the usefulness of SEA and springs in an active ankle-knee prosthesis design. A general model of ankle-knee prosthesis is studied and optimised. It focuses on power flows in the design elements (i.e. springs and motors) and shows a motor consumption decrease compared to direct drive actuation.

Chapter 4 discusses the principle of VSA in the design of an active ankle prosthesis, then explores the relevance of changing the actuator stiffness not only across different tasks (i.e. across the gait cycles), but also within the gait cycle, to further improve the global energy balance; and finally elaborates some concepts of controller to maximize energy-efficiency in the proposed design. A conceptual method using CVT is presented to implement a VSA which can change its stiffness without changing the amount of stored energy in the spring; i.e. with high energy efficiency.

Chapter 5 extends the concept presented in Chapter 4 to an ankle-knee prosthesis. It reports a preliminary design of a closed-loop controller of such a prosthesis, in order to achieve proper torque tracking at both the knee and ankle. This concept is studied not only for level-ground walking but also for stair ascending and descending.

After the theoretical analysis on lower limb prosthesis based on CVTs, the second part presents a new CVT technology which is embeddable in a prosthesis in terms of weight and size. Chapter 6 overviews the more popular and advanced concepts of CVTs.

Chapter 7 proposes a novel CVT principle which can achieve transmission ratio

variations at rest. The developed CVT is a modified planetary gear, whose planets are conical and mounted on inclined shafts, and whose ring is made of contiguous diabolo-shaped rollers. This configuration enables the control of the transmission ratio by adjusting the point of contact between the cones and rollers that comprise the ring. A traditional planetary gear system can be added to the CVT to form an Infinitely Variable Transmission (IVT). An IVT is a CVT whose ratio can be continuously varied from positive to negative values. The prototype presented in this chapter aimed at validating the kinematic concepts, and showed the ability of the design to continuously change its transmission ratio. The size of the prototype is twice the size needed for a prosthesis.

Chapter 8 presents a second prototype of the CVT concept. It aimed at validating the torque capacity and energy efficiency of the design with a size small enough to be embedded in a prosthesis.

Finally Chapter 9 concludes the thesis. It starts with a summary of the thesis. Then a discussion on the results overviews the thesis contributions in answering the research questions of Section 1.4. It shows the limits of the proposed prosthesis and CVT design. Then some perspectives for future work are presented. Suggestions are made in order to overcome the identified limits of the different designs.

1.6 Publications produced during the thesis

During the thesis, 11 contributions were produced. We can separate them in four categories:

1.6.1 Conference proceedings without peer-review

1. **Ch. Everarts**, A. Thissen, B. Dehez, and R. Ronsse. Control and Design of an Active Prosthesis Based on Adaptive Oscillators. *10th Belgian Day on Biomedical Engineering*, 2011
2. **Ch. Everarts**, B. Dehez, and R. Ronsse. Variable Stiffness Actuator Based on Infinitely Variable Transmission: Application to an Active Ankle Prosthesis. *Human Friendly Robotics*, 2012.
3. **Ch. Everarts**, H. Vallery, M. Bolliger, and R. Ronsse. Adaptive Position Anticipation in a Support Robot for Overground Gait Training. *12th Belgian Day on Biomedical Engineering*, 2013
4. **Ch. Everarts**, B. Dehez, and R. Ronsse. Continuously Variable Transmission: Application to an Active Ankle Prosthesis. *14th Belgian Day on Biomedical Engineering*, 2015

1.6.2 Conference proceedings with peer-review

1. **Ch. Everarts**, B. Dehez, and R. Ronsse. Variable Stiffness Actuator applied to an active ankle prosthesis: Principle, energy-efficiency, and control. *IEEE International Conference on Intelligent Robots and Systems*, pages 323-328, 2012.
2. **Ch. Everarts**, H. Vallery, M. Bolliger, and R. Ronsse. Adaptive Position Anticipation in a Support Robot for Overground Gait Training Enhances Transparency. *IEEE International Conference on Rehabilitation Robotics*, 2013.
3. H. Vallery, P. Lutz, J. von Zitzewitz, G. Rauter, M. Fritschi, **Ch. Everarts**, R. Ronsse, A. Curt, and M. Bolliger. Multidirectional Transparent Support for Overground Gait Training. *IEEE International Conference on Rehabilitation Robotics*, 2013.
4. **Ch. Everarts**, B. Dehez, and R. Ronsse. Novel infinitely Variable Transmission allowing efficient transmission ratio variations at rest. *IEEE International Conference on Intelligent Robots and Systems*, pages 5844-5849, 2015.
5. R. Ronsse, M. Tummers, and **Ch. Everarts**. Modeling and Control of a Transfemoral Prosthesis Embedding two Infinitely Variable Transmissions. *IEEE International Conference on Rehabilitation Robotics*, 2017.

1.6.3 Journal paper

1. **Ch. Everarts**, L. Mooney, F. Heremans, B. Dehez, H. Herr, and R. Ronsse. A Continuously Variable Transmission Allowing Ratio Variations at Rest. *IEEE Transactions on Mechatronics*, in preparation.

1.6.4 Patent

1. **Ch. Everarts**, B. Dehez, and R. Ronsse. Continuously Variable Planetary Transmission. WO 2015/169695 A1, 2015

Part I

Lower Limb Prosthesis Concepts

Lower Limb Prosthesis Overview

2

This chapter overviews the existing ankle and knee prostheses. It starts with a classification of ankle prostheses describing the most recent designs. It separates passive, semi-active and active prostheses. Second part of the chapter presents most recent knee and ankle-knee prostheses.

The ankle prosthesis overview is based on the Master Thesis of Alice Boone [11] and on the state of the art review papers from Dirk Lefeber's team at VUB [20, 102].

2.1 Ankle Prostheses

Prior to the early 1980s, most prosthetic feet focused on trying to restore basic walking and basic tasks ability. The wooden foot evolved to the conventional foot which was still very basic. Driven by the desire of amputees to walk more naturally, to reduce metabolic cost and even in some cases to practice sports, prosthetic feet were significantly improved over the last decades [20]. Each of these devices is aimed at improving control, comfort and cosmetic of the prosthesis with slightly different characteristics.

Today's prosthetic feet can generally be classified into three main categories: the conventional feet, the energy-storing-and-returning (ESR) feet, and the bionic feet. Fig. 2.1 presents an overview of this categorisation.

From an energetic point of view, ankle prosthetics are also divided into three groups: passive, semi-active and the active groups. Indeed, as experimentally measured, walking requires a certain amount of energy (see Chapter 1). The prosthetics evolved from conventional feet to the ESR feet by improving the materials and structure to better take advantage of internal energy storage. The ESR feet comprise compliant elements (e.g. a spring) to store and release elastic energy throughout each walking stance period. However due to their passive nature, such prostheses are un-

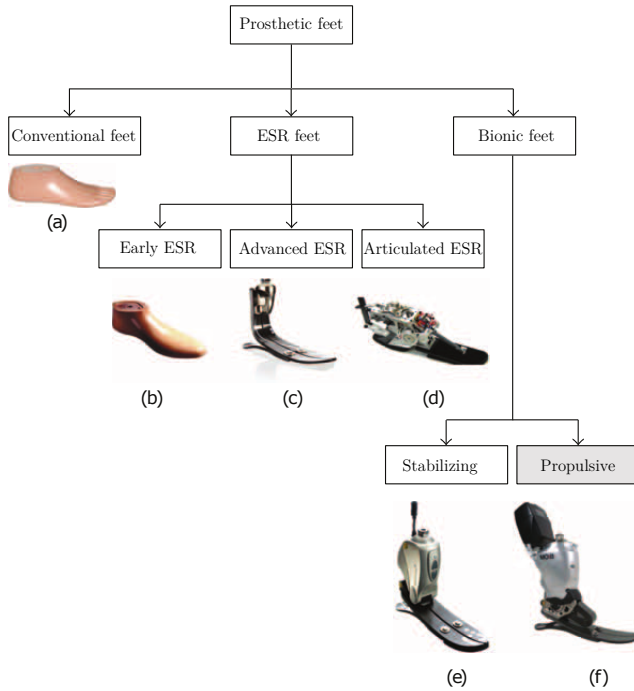


Fig. 2.1: Categorisation of today’s prosthetic feet with as example (a) the SACH-foot, (b) the SAFE foot, (c) the Össur’s Flex-Foot, (d) the CESR foot, (e) the Össur’s Proprio Foot, and (f) the iWalk’s Powerfoot BiOM [20].

able to generate more mechanical energy than what is stored during each walking cycle. To improve the energy release, semi-active prosthetics have been designed. In these prostheses, the energy release time is controlled and occurs exactly when it is needed during the gait cycle. Semi-active prostheses improve the timing of energy storing and release but, like the conventional feet, they are unable to generate more mechanical energy than what is stored during each walking step. To add external power to provide active push-off, active prosthetics have been designed. Thanks to both passive and motorised elements such devices emulate more closely the human ankle foot functions.

2.1.1 Passive and Semi-Active Prostheses

Conventional Feet

The most common conventional prosthetic foot is the solid ankle cushioned heel foot (SACH foot, Fig. 2.2) which for years has been the industry standard. Despite being the simplest non-articulated foot, the SACH foot remains a popular prosthetic device especially in low income countries because it is robust and cheap. Its compressible heel provides pseudo-plantarflexion after heel strike and the rigid keel pro-

vides mid-stance stability. The maximal torque that this foot can provide is however only two-thirds of the one measured during healthy walking [102].



Fig. 2.2: The SACH foot [102]

Energy-Storing and Returning (ESR) Feet

The desire of transtibial amputees to participate in sports and more dynamic activities resulted in the development of energy-storing-and-returning (ESR) feet. This new technology is then capable of storing energy during stance and returning it to the wearer to assist in forward propulsion in late stance. The first class of ESR (early ESR) is a conventional foot filled with the equivalent of a spring. Later, the advanced ESR emerged with a completely new concept that revolutionized the lower limb prosthetic world: prostheses with a flexible carbon fiber shank and heel spring. The last class, articulated-ESR feet, typically make use of electronics to engage or disengage locking mechanisms. It is why they are called semi-active.

Early ESR Feet The first ESR foot was introduced in 1981 as the Seattle foot (Fig. 2.3 A). Thanks to its flexible keel surrounded by polyurethane, it returns a portion of the input work (provided by the weight of the body to load the spring into compression) to the individual later in gait. However, the energy lost in the system as a result of friction remains high and is dissipated as heat and vibrations. Different models available on the market are shown in Fig. 2.3 [102].

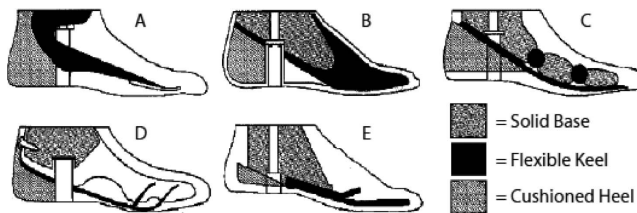


Fig. 2.3: Energy-storing-and-returning feet. (A) Seattle foot. (B) Dynamic foot. (C) STEN foot. (D) SAFE foot. (E) Carbon Copy foot [102]

Advanced ESR Feet In 1987 Flex-foot Inc. commercialized the first advanced ESR foot. This new concept comprised a flexible, 100% carbon fiber shank and heel spring. This device differed from others as it allows the entire prosthesis, rather than solely the foot part, to flex, store and return energy. This idea was further optimized



Fig. 2.4: Advanced ESR feet. (A) Flex-foot Axia (B) LP-Ceterus (C) Talux foot (D) VariFlex (E) Re-Flex VSP (F) Modular III (G) Flex-Sprint (H) Sprinter (I) Springlite foot (J) Pathfinder [102]

and gave birth to many other sophisticated designs once or currently on the market, such those presented in Fig. 2.4.

Articulated ESR Feet The articulated ESR feet typically make use of electronics to engage or disengage locking mechanisms. The energy release time is controlled and occurs when it is needed during the gait cycle. These prostheses improve the timing of energy storing and release. Although these devices are highly elaborated mechatronic systems, they are not active prosthetics since they do not make use of external power to provide their wearer with active push-off properties. It is why they are called semi-active. Like the passive prosthesis, they are unable to generate more mechanical energy than what is stored during each walking step. To name a few, the Brackx et al. AMP-Foot 1.0 and 1.1 [12], the Collins and Kuo's CESR foot [113] and the Mitchell et al. Vanderbilt ankle-foot prosthesis with delayed plantarflexion [82] are all part of the articulated-ESR feet category (Fig. 2.5).

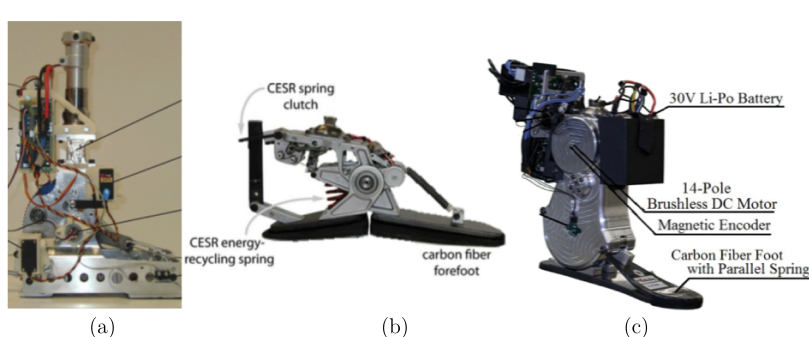


Fig. 2.5: Articulated ESR feet. (a) The AMP-foot 1.0 [12], (b) The CESR foot [113], and (c) The Vanderbilt prototype [82].

Stabilizing Feet

Most of today's commercialized powered transtibial prostheses use actuation to provide stabilization of the ankle-foot complex. This kind of prosthesis uses either hydraulic or electric actuation to provide natural ankle kinematics and intelligent terrain adaptation but does not provide more power to the individual than the one stored during gait. Some examples of popular stabilizing bionic feet (Fig. 2.6) are the Motion and Raize Foot1 (Fillauer, US), the Proprio Foot2 (Össur, Iceland) and the Elan foot3 (Endolite, UK).



Fig. 2.6: Stabilized bionic feet. (a) The Raize foot (Fillauer), (b) the Proprio foot (Össur), and (c) the Elan foot (Endolite).

2.1.2 Active Prostheses

Active prostheses make use of external mechanical power generation to provide active push-off. This actuation can be performed with a stiff or a compliant actuator. But when using the first one, both negative and positive energy flows have to be delivered, the mechanical energy requirements are thus much higher. Unlike stiff actuators, compliant actuators are able to store energy by elastic deformation. Within compliant actuators a distinction can be made based on the energy source, differentiating hereby pneumatically and electrically actuated propulsive devices. Originally, pneumatic actuators were chosen because their design and setup matched best to the musculoskeletal structure and properties of human beings. In fact the first propulsive prosthesis designed in 1998 by Klute et al. [59] was based on pneumatic actuators, also called pneumatic artificial muscles (PAM) [24,58,60]. Although these pneumatic actuators are capable of providing enough force to propel amputees, their control difficulties, large dimensions that result from the high-pressure air pump and the high cost of pressurized air production restrained the development of pneumatic prostheses [116]. Therefore, despite their lower force-to-weight ratio, electric actuators are preferred in more recent prostheses. They present the advantage of reducing the power requirements of the drive resulting in a smaller, less heavy and cheaper actuation setup [20].

The most advanced class of prosthetic feet are the propulsive devices. By combining passive and active elements as well as an appropriate control, these prostheses aim at emulating the human ankle foot functions. Depending on the actuation type the electrically powered propulsive bionic feet can be further categorized into five classes; the series elastic actuation (SEA), the series elastic actuation with parallel spring (SEAPS), the variable stiffness actuation (VSA), the variable stiffness actuation with parallel spring (VSAPS) and the explosive elastic actuation (EEA). Each one of these actuation types has been developed, one way or another, on lower limb prostheses.

This section aims at reviewing the mechanical architectures of the most advanced prostheses that present a promising future in order to outline the strengths and weaknesses of each design.

Series Elastic Actuator (SEA)

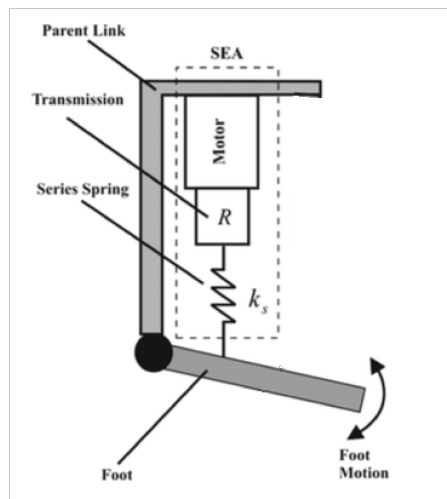


Fig. 2.7: Series Elastic Actuator applied to an ankle prosthesis [6].

As shown in Fig. 2.7, a SEA consists of a motor in series with a compliant element (e.g a spring) [76]. Since they are in series with the load, the required motor torque is still equal to the target ankle torque but the load displacement is equal to the sum of the motor and the elastic element displacements. Consequently, thanks to the elastic energy storage during periods of negative power, the motor speed requirement is lowered when this energy is transferred back into the system during periods of positive power. The required motor power is therefore lowered.

As the ankle rolls over the foot during the stance phase, the spring is compressed

by the user weight. Additional deflection in the spring is achieved by correctly positioning the low power motor so that the desired ankle joint angle and torque are realized [8]. However, thanks to inherent compliance, the ankle is not forced to follow a specific pattern. Environmental irregularities are safely absorbed providing a safer interface between the user and the prosthetic [48].

Many on-development devices such as the SPARKy project, the Marquette project and the PANTOE prosthesis to name a few, are relying on SEA's. To give an overview of this existing technology and their state of progress, the general principles are shortly presented.

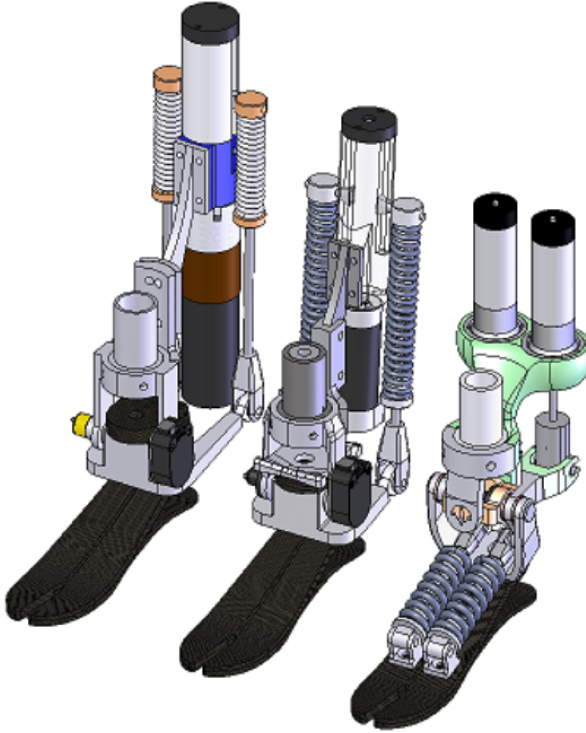


Fig. 2.8: From left to right SPARKy 1, 2 and 3 CAD models [8].

SPARKy The SPARKy project, short for Spring Ankle with Regenerative Kinetics that began at the Arizona State University (ASU, United States) in 2007 led to three different concepts called SPARKy 1, 2 and 3 (Fig. 2.8). SPARKy is based on a robotic tendon to actively stretch springs when the ankle rolls over the foot, thus allowing the springs to thrust or propel the artificial foot forward for the next step [49]. The

follow-up design SPARKy 2 was built from SPARKy 1 to improve its form and functions. It incorporates more efficient linear transmission options using a roller screw, a smaller and more powerful brushless motor without the need for a gearbox and presents a significant overall decrease in size and weight (from 2,7 kg to 2 kg). These two designs have one degree of freedom, their active motion is thus limited to the sagittal plane.

To fulfil the more demanding and athletic movements required for an active duty soldier or an active individual, the more robust and agile robotic prosthesis SPARKy 3 was designed. "We want our finished device to allow soldiers to return to active duty", said Thomas Sugar, the research leader. The main idea of the new device is to use two motors to give the wearer powered running abilities. The two individual motor controls enable the powering of an additional rotational degree-of-freedom. The combined motion has the added benefit of dividing the workload of each motor. As so the motors operate well below their peak capabilities increasing their efficiency. The motors will also produce less heat reducing the risk of failure.

This design is the first to incorporate active control of both plantar and dorsiflexion as well as inversion and eversion. However the second degree-of-freedom and dual actuators add an element of complexity in the challenge of controlling the device.

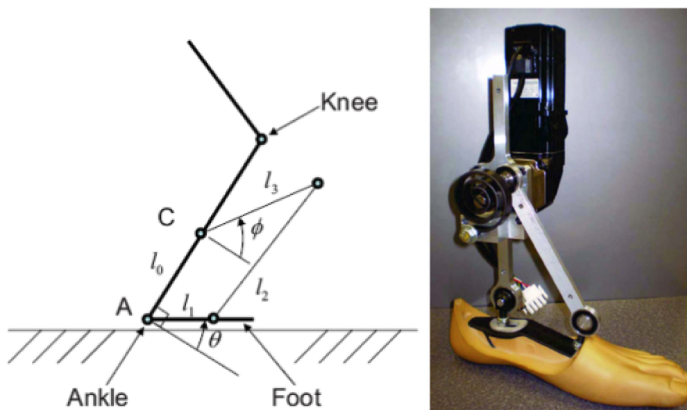


Fig. 2.9: Marquette four-bar prosthesis. (left) Model configuration and (right) prototype [9].

The Four-Bar Mechanism Prosthesis At Marquette University (Milwaukee, United States), Bergelin and his team developed a powered ankle prosthesis which uses a four-bar mechanism in combination with a linear spiral torsion spring and a motor [9]. In this design (Fig. 2.9), the SEA is located at a point on the calf (joint C) and provide a torque at the ankle (joint A). The idea of using a four-bar mechanism came from the issue of mimicking the non-linear stiffness of the ankle behaviour. Indeed,

by appropriate link length selection the non-linear relationship between input and output link angles can be regulated.

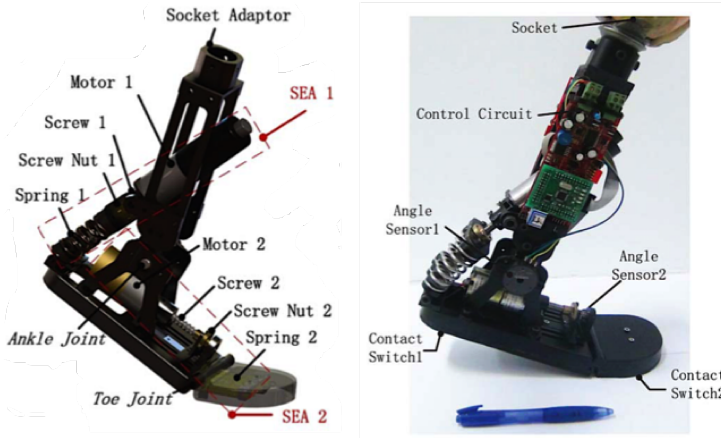


Fig. 2.10: PANTOE prosthesis. CAD and prototype configuration models [115].

PANTOE Zhu et al. at Peking University (China) developed an ankle prosthesis named PANTOE, short for powered transtibial prosthesis with ankle and toe joints [115]. This prototype is the first prosthesis to provide active actuation of the toes. In fact it consists in both an ankle joint and a segmented foot with toe joint, each being driven by a SEA.

To provide the torque to the ankle, 83 W brushed DC motor was used, while a 30 W DC brushed motor actuates the toe joint. When both motors are not functioning, PANTOE can act as a traditional passive prosthesis with spring-like characteristics. The angle range of the ankle joint goes from 20° dorsiflexion to 30° plantarflexion and the angle range of the toe joint is from 0° to 90° dorsiflexion. The PANTOE weighs 1.47 kg (not including the rechargeable battery). The CAD and the prototype of PANTOE are shown in Fig. 2.10.

Experimental results indicate that the PANTOE prosthesis improved the walking performance of amputee compared with a commercial passive one. Joint angles and vertical ground reaction forces of both prosthetic and sound sides have been shown to be symmetric.

Series Elastic Actuator with Parallel Spring (SEAPS)

Another type of actuation is the SEAPS. It comprises a compliant element in series and one in parallel (SEA + parallel spring). This actuation scheme has the potential to lower both the motor speed and torque requirements. While the SEA reduces the

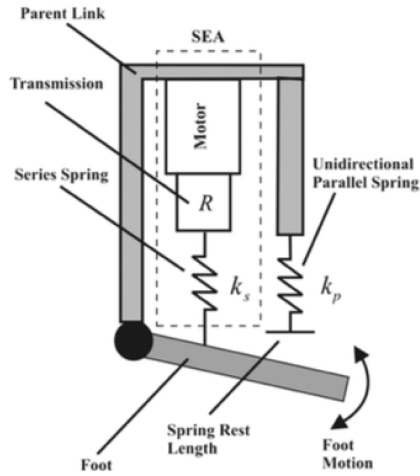


Fig. 2.11: Series Elastic Actuator with Parallel Spring applied to an ankle prosthesis [6].

velocity required by the motor, the parallel spring reduces the load required by the motor.

The MIT Prosthesis A major contribution of the research on powered prosthesis with electric actuators is conducted by Hugh Herr and his team in the Biomechanics Lab at MIT [4–7]. Fig. 2.12 shows the evolution of their work. The two first ankle-foot prototypes led to the first operating powered foot (third prototype), the next two prototypes led to the PowerFoot 1 prototype (not shown) and the last two prosthesis are the BiOM prototype and its commercialized version [20]. Motivated by human ankle biomechanics, these prostheses with electric actuators are the most relevant example of the SEAPS configuration.



Fig. 2.12: Ankle-foot prostheses developed at the MIT [20].

These powered foot prostheses were developed using a combination of a unidirectional parallel spring and a high-power Series Elastic Actuator (SEA). Its working principle consists in loading the SEA during the controlled dorsiflexion phase and in activating the torque source in parallel when peak power is needed [20]. As a result of this, the system has human-like characteristics; energy is added to the system to provide push-off and the joint stiffness/damping is controlled to mimic the sound ankle behaviour.

iWalk, the spin-off company from the Biomechatronics Lab recently commercialized the first active prosthesis on the market, the BiOM prosthesis (device on the right in Fig. 2.12). The BiOM is a completely self-contained device weighting approximately 1.8 kg and, unlike the majority of other prostheses, it has the size of an intact biological ankle-foot complex. The foot includes a passive low profile Flex-Foot 1 minimizing hereby ground contact shock to the wearer. The unidirectional leaf spring acts across the ankle joint when the ankle and the foot are perpendicular to each other. The series spring consist of a Kevlar-composite leaf spring which is connected to the ball nut with a direction dependent moment arm. Thanks to this, the actuator stiffness is different for positive and negative torque outputs. The external power is provided by a 0.22 kg lithium polymer battery. This battery is able to provide two hours of powered walking which corresponds to 5000 steps. The device includes a neuromuscular control exhibiting an inherent adaptation to ground slopes [20].

Variable Stiffness Actuator (VSA)

To comply with different gaits and cadences, it is useful to modify the stiffness of the actuator. Indeed a given stiffness optimizes the power flow only for a given walking cadence (roughly corresponding to the natural frequency of the system). Therefore, optimizing the power flow at another cadence would require another stiffness; to do so the design of VSA is hence indispensable. The advantage of using variable stiffness actuators is that they can be optimized considering different step lengths, walking speeds, and terrain adaptation. The drawbacks of this type of actuator are the added complexity and weight to the system.

An example of VSA is the MACCEPA [98] conducted by Dirk Lefeber's team at the Vrije Universiteit Brussel (VUB, Belgium). In Fig. 2.13, the essential parts of a MACCEPA are drawn. The main motor (M) is attached to the anchored link and drives the moment arm (b) around the ankle joint (a). The moment arm is displaced by an angle α , compressing the MACCEPA spring along the foot shaft (C), creating an ankle torque. The small motor (m) is used to precompress the main spring, and is attached to the foot link. The precompression of the spring changes the seen stiffness by the joint [35].

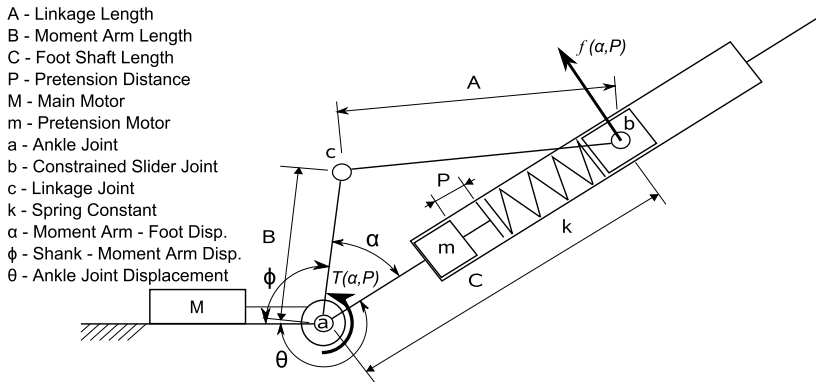


Fig. 2.13: Configuration of a MACCEPA using rigid linkages [35].

An example of VSA prosthesis based on MACCEPA is the transfemoral CYBERLEGS project (Fig. 2.14) designed by the same team. This prosthesis is part of the CYBERLEGS FP7-ICT Project, which combines a prosthesis system to replace a lost limb in parallel with an exoskeleton to assist the sound leg, and sensory array to control both systems [23, 33–35, 37]. In order to reduce as much as possible the extra weight in the ankle module, a very small motor was selected with a very high reduction ratio to vary the stiffness of the joint. This way the stiffness of the ankle joint cannot be changed during one stride but only over a series of steps providing slow adaptation to different gaits.

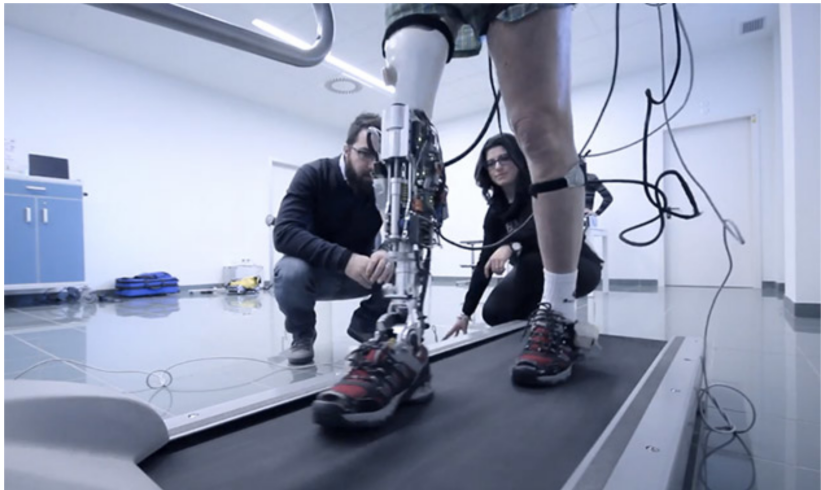


Fig. 2.14: α -Prototype of the CYBERLEGS transfemoral prosthesis.

Fig. 2.15 shows all of the physical components of the ankle joint prosthesis of the Cyberlegs α -prototype. There is a plastic plate that is used to provide toe flexion and a rubber cosmetic cover that fits over the mechanism and allows it to fit snug

within a shoe. The cosmetic cover also provides a smooth surface to interface with the insoles used for center of pressure measurements [35].

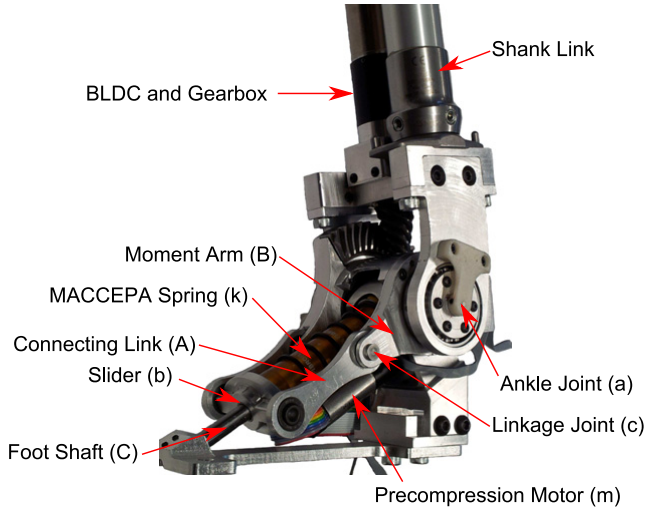


Fig. 2.15: Implementation of the MACCEPA actuator in an ankle prosthesis [35].

Explosive Elastic Actuator (EEA)

In general, the explosive elastic actuator (EEA) consists of a spring behind a locking mechanism placed in series with a SEA. The physics of this catapult-like mechanism is based on the progressive storage of energy which is released when needed, in one shot, to hurl the payload hereby creating an energetic explosion. This kind of explosive motions is also required and widely used in, for example, jumping, kicking, throwing, and hammering robots [20]. The torque requirements of the EEA are identical to the SEA. However thanks to the locking mechanism, the overall power rating of the electric drive is considerably reduced compared to the other bionic devices. Indeed, in this case, the motor works during a time three times longer reducing therefore by the same amount its actuator speed and power. Therefore, instead of using a high power drive during a short lapse of time, the mechanism enables the use of a low power motor during a longer lapse of time.

The AMP-Foot The Robotics & Multibody Mechanics Research group at the Vrije Universiteit Brussel (VUB, Belgium) has been working on this optimal power distribution and created the Ankle Mimicking Prosthetic (AMP)-Foot. The AMP-Foot 2 with a total weight of 2.5 kg consists of a linear actuator (60W brushed DC motor + ball screw) attached to the leg part of the prosthesis. As shown in Fig. 2.16, the output of the linear actuator is connected to a lever arm with two tension springs, called the push-off springs. The lever arm and the leg rotate around the ankle axis. A

similar connection is made between the foot and the lever arm by means of a spring, named the plantar flexion spring.

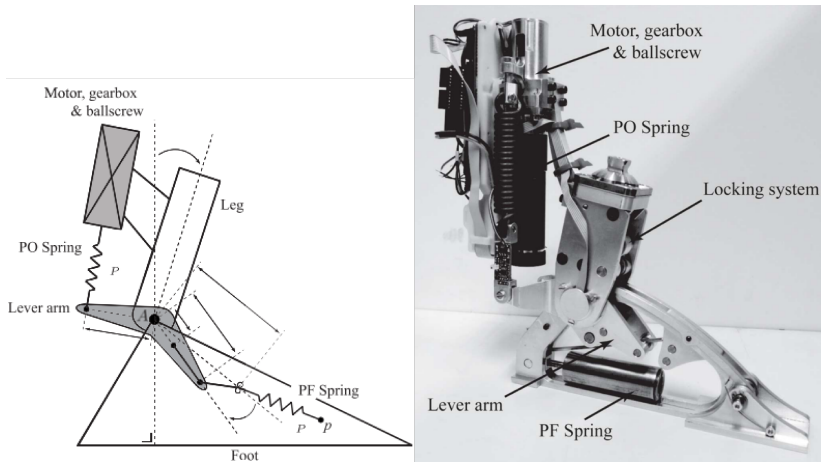


Fig. 2.16: The Ankle Mimicking Prosthetic (AMP-) Foot 2 schematic and prototype models [19,20].

One important element not shown in the schematic but in the prototype model is the locking mechanism. When the locking mechanism is engaged the lever arm is fixed to the leg and the prosthesis behaves like an ESR prosthesis. On the contrary, when it is disengaged, the lever arm, the leg and the foot can move freely but remain constrained under the influence of the springs. It is clear that one of the most critical part in this system is the locking and unlocking mechanism. In this case, the energy storage makes sense if no (or very little) extra energy is used to release it. Thanks to the low force needed to disengage it the VUB team developed a four bar mechanism moving in and out of its singular position. Further details can be found in [19].

As a follow up of the AMP-Foot 2, the AMP-Foot 3 was developed [17]. The novelty in the AMP-Foot 3 prototype (shown in Fig. 2.17) is the design and incorporation of an extra locking mechanism, namely a resettable overrunning clutch. The addition of this locking system to the device has the advantage of improving energy storage during the early stance phase and naturally adapting the prosthesis to different walking speeds and slopes. This prototype uses this extra locking mechanism in order to mimic the change in rest position of the robotic ankle after foot-flat is achieved. Thanks to this extra locking mechanism more gravitational energy can be stored in the plantar flexion spring. As an effect of this the motor actuating the catapult actuator is rated to only 40W (efficiency included). Therefore, a 50W Maxon brushless motor has been selected.

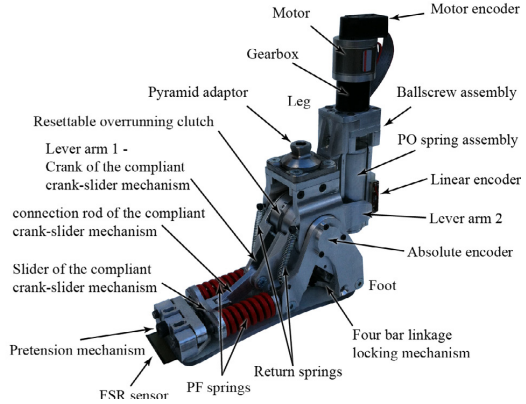


Fig. 2.17: The Ankle Mimicking Prosthetic (AMP-) Foot 3 prototype [17].

2.2 Knee Prostheses

Level ground walking is generally an energetically negative action for the knee [108], therefore most passive damping solutions are generally sufficient for emulating its behavior. In contrast, the sit-to-stand and stair climbing require positive energy injection at the knee [108].

Stance flexion is not often designed into current knee prostheses, instead opting for what is termed "stability" in the prosthetics nomenclature. This means that the knee is locked into position and flexion is avoided to make the prosthesis more predictable in its behavior. This comes at a cost of higher heel collision forces and non-physiological gait kinematics.

As for the ankle, knee prostheses can be sorted into three categories from an energetic point of view: passive, semi-active and active.

Passive knees are tuned with a constant damping and stiffness. However, semi-active knees can adapt damping and springs positioning to fit with the patient motions and gait. As for the ankle, these prostheses can not inject mechanical power in the joint.

2.2.1 Passive Knee Prostheses

Fully passive knees, such as the Balance Knee or the Mauch knee (see Fig. 2.18) can have varying characteristics such as different damping characteristics during stance and swing, but are limited to a preset number of values. They are designed for the less active users and provide a high level of security and stability.



Fig. 2.18: Fully passive prosthesis. (a) the Össur's Balance Knee, and (b) the Össur's Mauch Knee.

2.2.2 Semi-Active Knee Prostheses

The majority of the top knee prostheses on the market today are the microprocessor controlled prostheses. It means that they can continuously adapt damping and spring positioning systems. They use small actuators that allow the characteristics of the device to adapt to the motion of the individual during walking. Examples of this can be found in the the Ottobock Genium and C-Leg, Össur Rheo Knee, Freedom Innovations Plié, among others (see Fig. 2.19).



Fig. 2.19: Semi-active knees. (a) the Ottobock C-Leg, (b) the Ottobock Genium, (c) the Össur Rheo Knee, and (d) the Freedom Innovations Plié.

Some knee prosthesis are designed to work together with an ankle prosthesis by sharing information such as the Össur Symbiotic Leg, or the Blatchford/Endolite Linx (see Fig. 2.20).

However all these prostheses are unable to generate mechanical energy, the only



Fig. 2.20: Semi-active legs. (a) the Össur Symbionic Leg, and (b) the Blatchford/Endolite Linx.

energy used to actuate the joint is energy that has been harvested from the gait cycle. To provide mechanical energy to the joint, an active prosthesis must be used.

2.2.3 Active Knee Prosthesis

There have been a number of recent energetically active actuated knees that are capable of producing net positive knee joint power.

This section aims at reviewing the mechanical architectures of the most advanced knee prostheses that present a promising future to outline the strengths and weaknesses of each design.

Össur Power Knee

The Össur Power Knee, while not widely used at the moment, is beginning to find use in the market (see Fig. 2.21).

Össur's Power knee uses a SEA as motion system. This allow stance flexion of the knee in the period directly after heel strike. The Power Knee uses a high power motor behind its SEA to provide active energy as well as provide necessary SEA bandwidth to track the rest of the gait cycle.

Since this knee has been developed with a commercial goal, nothing has been published about it except a patent [40].



Fig. 2.21: Össur Power Knee.

Vanderbilt Prosthetic Leg

The Vanderbilt prosthetic leg, illustrated in Fig. 2.22, is made of completely decoupled knee and ankle [61]. The knee module is a fully motor driven device, using a backdrivable motor and no spring.

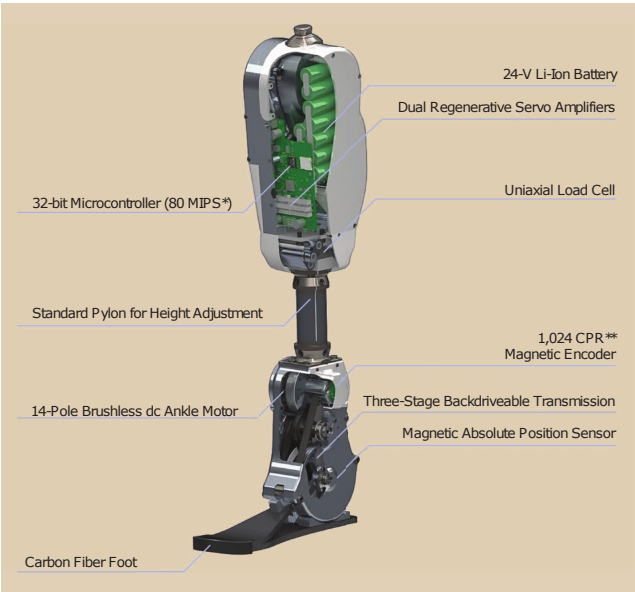


Fig. 2.22: CAD model of the Vanderbilt powered leg prosthesis [61].

The prosthesis uses impedance control to imitate healthy behavior. The control approach requires the prosthesis to be capable of using feedback control to provide

software emulation of a range of stiffness and damping behaviors. This requires a high level of control to reproduce this healthy gait.

RIC Hybrid Knee Prosthesis

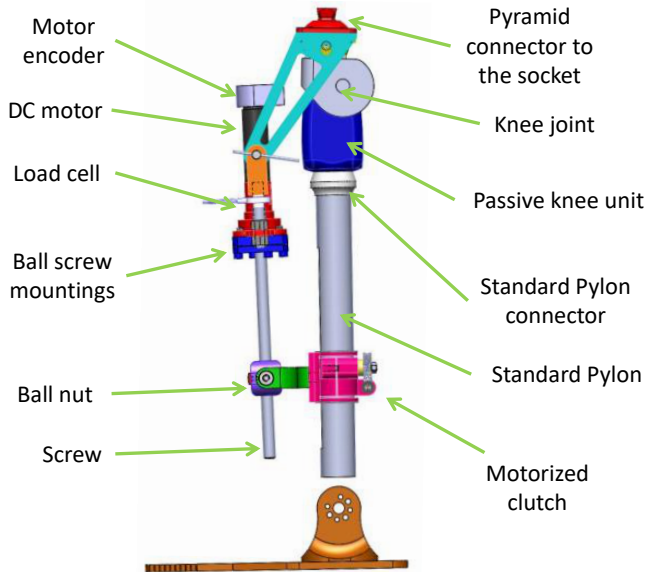


Fig. 2.23: CAD model of the RIC Hybrid Knee Prosthesis [62].

The RIC Hybrid Knee Prosthesis [62] is a completely passive knee that is used for most normal walking conditions. However, it also has a motor that can be clutched in and out when actuation is required (e.g. sit-to-stand and stair climbing operations). A CAD model of the prosthesis is represented on Fig. 2.23.

ANGELAA

The device named ANGELAA (ANGLE-dependent ELastic Actuator) was developed at ETH and Delft [75]. It spent fine attention to the arrangement of parallel and series elastic elements to passively match the actuator stiffness to the desired actual joint stiffness based on simulation and gait studies. This prosthesis is pictured in Fig. 2.24.

The prosthesis is actuated by a Series (Visco-) Elastic Actuator (SVA), where a nonlinear transmission changes the actuator stiffness as a function of the joint angle. Instead of metal springs, which are commonly used as series-elastic elements, the concept relies on rubber cords. In addition, the concept also incorporates parallel springs that were positioned such that, in combination with the actuator, the asymmetric moment-angle profile of a physiological joint is approximated. This al-

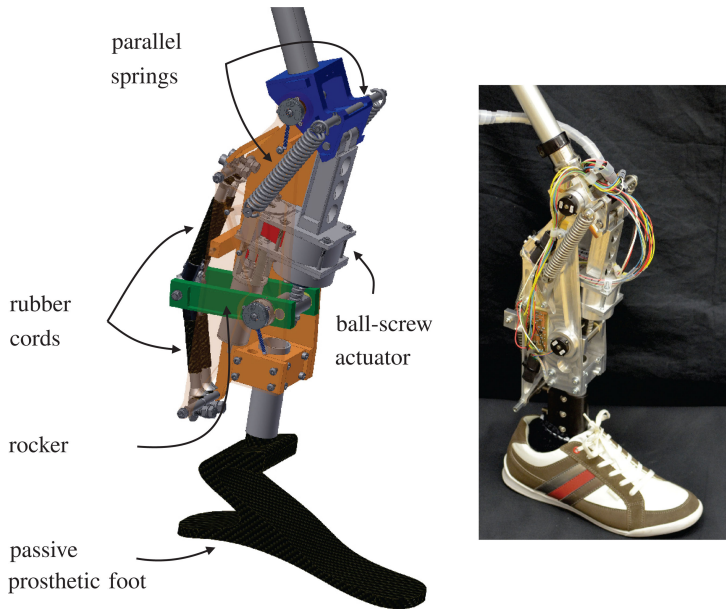


Fig. 2.24: CAD Drawing of the ANGELAA-leg (left) and picture of finished prototype (right). [75].

lows a passive minimization of actuator work necessary to provide correct output impedance.

MIT CSEA knee

The CSEA Knee, developed in the Biomechanics Group at the MIT Media Lab [80], combines a SEA with a clutch. The mechanism is able to provide the benefits of a SEA while simultaneously providing a low-energy, elastic state when a mechanical clutch is engaged. Fig. 2.25 shows a CAD model and the actual prototype of the prosthesis.

The series stiffness can be designed to provide a specific torque-angle relationship optimized for the dynamics of a particular task. In the CSEA, the series spring stiffness has been chosen to imitate measured biological knee stiffness during stance flexion (i.e. the early-stance phase of walking). During this period, the clutch is engaged to provide the reaction torque on the motor shaft, thereby bypassing the electric motor. The energetic economy stems from the difference in electrical energy required to provide the reaction torque.

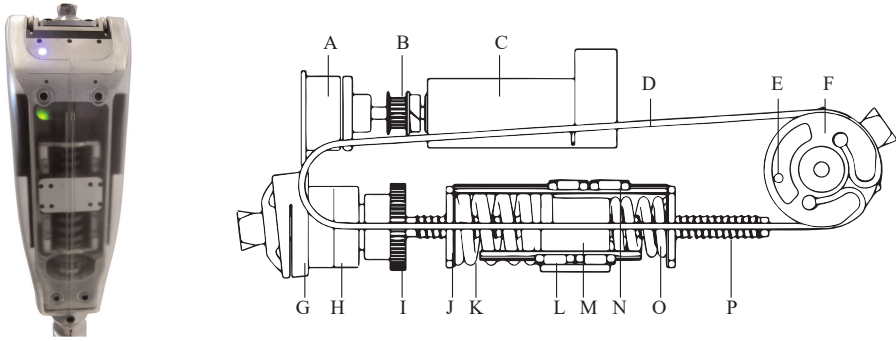


Fig. 2.25: The major components of the clutchable series-elastic actuator knee are depicted: (A) electromagnetic clutch; (B) motor timing pulley; (C) Maxon EC-40 4-Pole 200 W brushless motor; (D) steel drive cable; (E) internal hard stop; (F) knee pulley and joint; (G) distal pyramid with embedded load sensor; (H) paired 35° angular bearings; (I) ballscrew timing pulley; (J) series-elastic housing; (K) extension spring; (L) ball nut linear bearings; (M) ball nut; (N) series-elastic housing linear bearings; (O) flexion spring; (P) ballscrew [80].

2.3 Coupled Ankle-Knee Prostheses

Some transfemoral prostheses are designed to allow energy transfer between joints. Ankle and knee modules can not be separated, they form one single device. Energy transfer can be realised through springs or cables.

2.3.1 WalkMECH

Koopman and his team developed a fully passive energy efficient transfemoral prosthesis exchanging energy between ankle and knee [92–94].

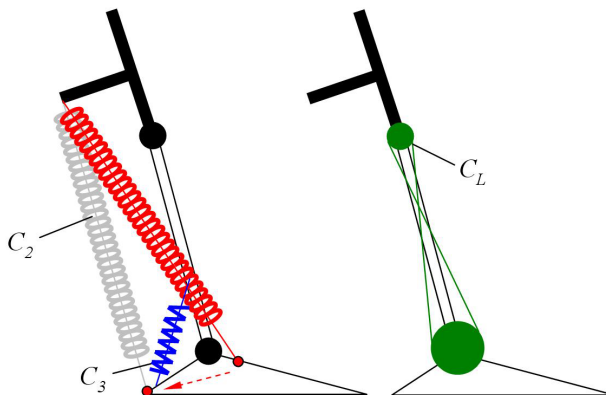


Fig. 2.26: Conceptual principle of WalkMECH [94].

The proposed concept relies on the energy transfer between both joints. The

transfemoral prosthesis absorbs energy during stance, swing phase and at heel strike, and it releases energy during ankle push-off. This has been realized by using three storage elements as depicted in Fig. 2.26 and explained hereafter, i.e.:

- The linkage mechanism C_L couples the knee and ankle joints kinematically and is responsible for the transfer of a part of the energy absorbed at the knee to the ankle push-off, as a consequence of the closed-loop kinematic chain.
- The elastic element C_2 couples the upper and lower leg, and its attachment point can move on the foot. C_2 stores the kinetic energy of the lower leg during swing motion. At the beginning of the swing phase, the attachment point of C_2 is changed from the heel to the upper part of the foot. At the end of the swing, the spring is loaded and its position changes back to the heel for releasing the stored energy for ankle push-off.
- The ankle elastic element C_3 connects the foot and the lower leg and is responsible for the main part of the energy absorption during dorsiflexion of the ankle. During the stance phase, a braking torque is applied to the ankle in order to bear the weight of the body. Instead of dissipating the energy by using a brake system, this elastic element provides the brake torque and stores the energy.

Fig. 2.27 represents the prototype and CAD of the prosthesis.

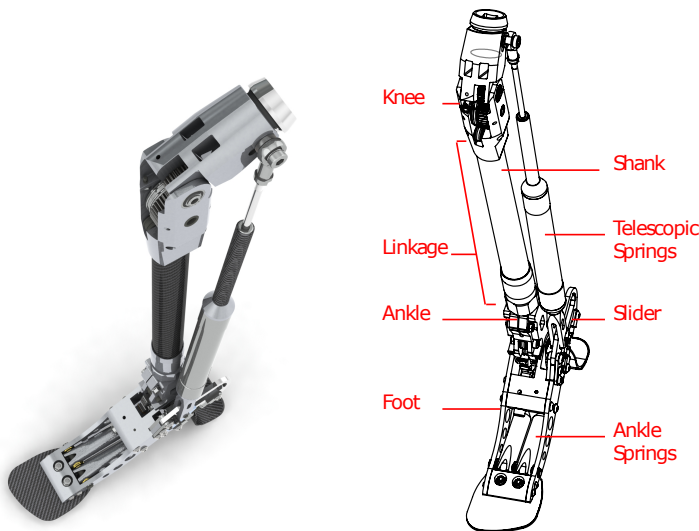


Fig. 2.27: Mechanical realization of WalkMECH [94].

2.3.2 CYBERLEGs Beta-Prosthesis

The CYBERLEGs β -Prosthesis was created by Dirk Lefeber and his team at VUB. This prosthesis is part of the CYBERLEGs FP7-ICT Project, which combines a prosthesis system to replace a lost limb in parallel with an exoskeleton to assist the sound leg, and sensory array to control both systems [23, 33–35, 37]. CYBERLEGs Beta-Prosthesis is presented in [33] (submitted but not published yet).

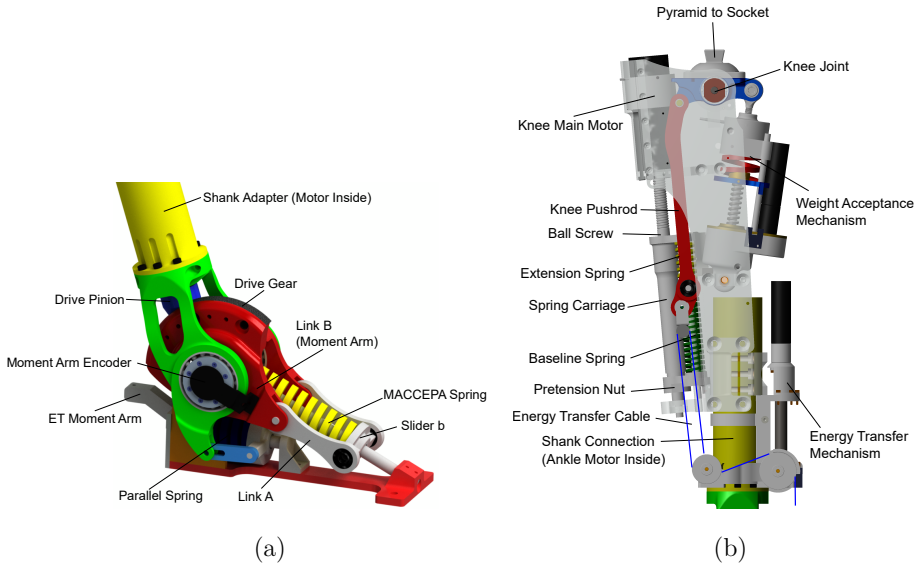


Fig. 2.28: CYBERLEGs Beta-Prosthesis. (a) CAD drawing of the ankle joint with corresponding parts labeled. (b) CAD model of the knee mechanism showing the three major systems of the knee. The front of the knee shows the components of the knee drive including the carriage and series elastic springs. The Weight Acceptance and Energy Transfer mechanisms can be found at the back of the knee. [33].

Both joints of the CYBERLEGs Beta-Prosthesis are actuated. The prosthesis consists of four major systems which are combined to produce the total torque required for normal walking with a good energy efficiency [33].

1. A powered ankle based on a MACCEPA architecture (see Section 2.1.2) which gives the ankle the ability to change stiffness. The ankle has an added parallel spring to assist the motor and reduce the torque peak needed.
2. A simple spring that is inserted at the knee during early stance phase to provide the natural characteristics of stance flexion without need of powered actuation. This mechanism is called the Weight Acceptance system. This system is capable of producing large reaction torques when external torques are applied, the main actuator does not need to operate during early stance and this system decrease the energy consumption.

3. The Knee Drive Baseline Actuator. This actuator is the main positive energy source of the knee joint, but under nominal use is primarily used to hold the Baseline Spring in place. During the gait cycle it is possible to fully drive the knee using this system, and it provides all of the power for sit-to-stand and stair climbing.
4. The Energy Transfer system. This system provides the late stance extension torque of the knee as the knee flexes, delivering this negative work from the knee joint as positive work at the ankle to reduce the ankle torque, known as the energy transfer period. This system does not provide net output energy, but rather uses a binary locked/unlocked condition to physically connect the knee and ankle.

The combination of these four systems provides energy efficient and natural gait kinematics during level ground walking cycle. The prosthesis uses minimal actuation but provides the possibility to modify its behavior delivering or removing external energy during the gait cycle and provide different characteristics while sit-to-stand and stair climbing.

2.4 Conclusion

In conclusion, many mechanical architectures were developed and implemented. Depending on the energy optimization, some are more complex than others. There is a trade-off between functions and complexity. To validate their research the authors compared the results with Winter's data or with matched control subjects. The range-of-motion, torques, powers and sometimes the energy consumption were analysed.

In this thesis, we will focus on a new concept of prosthesis trying to flatten as much as possible the motor power consumption and torque being needed.

Ankle-Knee Prosthesis: Series Elastic Actuators with Parallels Springs

3

This chapter shows the usefulness of SEA and springs in an active ankle-knee prosthesis design. Chapter 1 showed angle torque relationships for knee and ankle. These relationships show spring-like behavior during some phases of the cycle. This chapter aims at showing usefulness of springs to reproduce this behavior and decrease the energy flows in motors. In other words, if the torque-angle relationship could be rendered by a physical spring, the motor power is decreased. Only reactionary torque of SEA has to be generated by the motor, at lower motor speed.

For this aim, we used a general model of an ankle-knee prosthesis. Each joint is actuated by a SEA and connected to a parallel spring. The parallel spring aims at decreasing the torque that the SEA has to provide to actuate the joint. Stiffnesses of the elastic elements are non-linear and captured by 4th order polynomes. In addition to that, a coupling spring is placed between both joints in order to allow energy transfer between them. This spring is connected through a gear ratio in order to transfer energy but not especially with the same torque on both joints. This chapter aims at finding the best stiffness profiles for this model. To achieve that, a genetic algorithm has been implemented. This algorithm tends to minimise the energy flows in both motors.

3.1 SEA model

Fig. 3.1 shows the kinematic diagram of the studied prosthesis model. Both joints are connected to 3 springs:

1. A parallel spring with a stiffness directly connected between the tibia and the

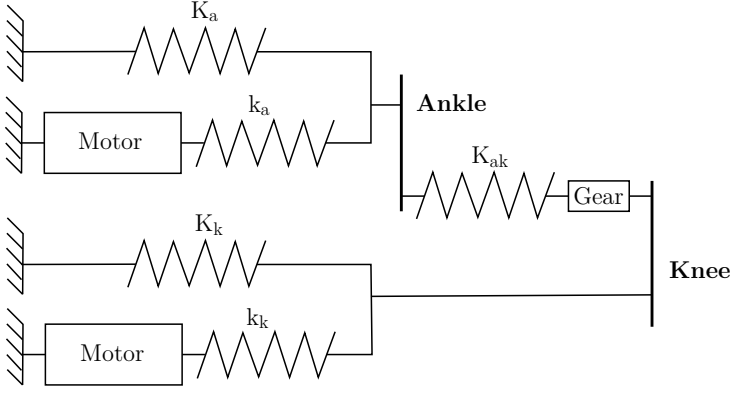


Fig. 3.1: Ankle/Knee prosthesis kinematic diagram

joint.

2. A SEA which is the connexion in series of a motor and a spring. The motor is connected to the tibia and the spring to the joint.
3. A coupling spring which is directly connected between both joints.

The model variables are listed as following:

$\theta_{m_i}, \omega_{m_i}, T_{m_i}$	Motor i position, velocity and torque.
$\Delta\theta_{k_i}, \theta_{0,k_i}$	SEA spring deformation and free-length.
$\Delta\theta_{K_i}, \theta_{0,K_i}$	Parallel spring deformation and free-length.
$\Delta\theta_{K_{ak}}, \theta_{0,K_{ak}}$	Coupling spring deformation and free-length.
n	Transmission ratio between the coupling spring and the knee
θ_i, ω_i, T_i	Joint position, velocity and torque (before the IVT). As such, $\theta_t = \theta_m - \Delta\theta_s$
k_i, K_i, K_{ak}	Intrinsic stiffness of the SEA, parallel and coupling springs respectively
$T_{k_i}, T_{K_i}, T_{K_{ak}}$	Torque of the SEA, parallel and coupling springs respectively

where i can be either a or k for ankle and knee respectively.

Each spring has a non-linear stiffness captured by two different sets of parameters for positive and negative deformations.

The relationship between torque and deformation was thus captured as a polynome of the 4th order. We choose this complexity to allow the characteristic to have peaks (i.e. allow the presence of unstable equilibrium position). For a spring with a deformation $\Delta\theta$, the torque T can be obtained with the following relationship:

$$\begin{cases} T = a \Delta\theta^4 + b \Delta\theta^3 + c \Delta\theta^2 + d \Delta\theta & \text{if } \Delta\theta > 0 \\ T = e \Delta\theta^4 + f \Delta\theta^3 + g \Delta\theta^2 + h \Delta\theta & \text{if } \Delta\theta \leq 0 \end{cases} \quad (3.1)$$

Each joint has three power inputs. Two are passive (i.e. the parallel spring and coupling spring) and the third, the SEA, is active.

1. The parallel spring deformation is defined as the difference between the joint position and the spring free length.

$$\Delta\theta_{K_i} = \theta_i - \theta_{0,K_i} \quad (3.2)$$

2. The coupling spring deformation is defined as the difference between the knee position multiplied by the transmission ratio n , the spring free length and the ankle position.

$$\Delta\theta_{K_{ak}} = n \theta_k - \theta_a - \theta_{0,K_{ak}} \quad (3.3)$$

3. The SEA spring deformation is defined as the difference between the joint position, the motor position and the spring free length.

$$\Delta\theta_{k_i} = \theta_i - \theta_{m_i} - \theta_{0,k_i} \quad (3.4)$$

The SEA has to provide the remaining torque needed to the joint. Indeed, parallel spring and coupling spring torques are entirely determined by the joint kinematics and the SEA motor adjusts its position to apply the desired torque to the joint. Winter position data are used to force the kinematic and the SEA tracks the Winter torque [108].

$$\begin{cases} T_{m_a} = T_{k_a} = T_a - T_{K_a} - T_{K_{ak}} \\ T_{m_k} = T_{k_k} = T_k - T_{K_k} - n T_{K_{ak}} \end{cases} \quad (3.5)$$

By introducing T_{k_i} and $\Delta\theta_{k_i}$ in Eq. (3.1), SEA spring deformation can be obtained. θ_{m_i} can then be calculated with Eq. (3.4).

3.2 Optimisation Method: Genetic Algorithm

Eqs. (3.1) to (3.5) give the model of a generic ankle-knee prosthesis design. This model has a lot of parameters. There are 5 springs and a gear ratio to dimension. Each spring has 9 parameters: its stiffness parameters (a, b, c, d, e, f, g and h) and its free length. The model has thus 46 parameters (vector X) which have to be optimised in order to find the best configuration. In order to achieve that, a genetic algorithm was implemented. The parameters X have been optimised to reduce the absolute value of the energy supplied by the motors with the following objectives functions:

$$F_1(X) = \int_S |P_{m_a}| dt = \int_S |T_{m_a}| \cdot |\omega_{m_a}| dt \quad (3.6)$$

$$F_2(X) = \int_S |P_{m_k}| dt = \int_S |T_{m_k}| \cdot |\omega_{m_k}| dt \quad (3.7)$$

where P_{m_i} is the power supplied by motor i , S is the period of one step and t is the time. These objective function minimised the power flows in the motors and then Joules losses.

The algorithm used was developed by Jonathan Denies during his PhD thesis [26]. The program is written in Matlab. The algorithm ran during 50 000 generations with a population of 100 individuals. Between each generation, 2% of mutation and 90% of crossover occurred. Winter data [108] were used as kinematic and dynamic data to run the algorithm.

Boundaries on the parameters were the following:

- Polynome coefficients (a, b, c, d, e, f, g, h) : $[-100; 100]$
- Free-length (θ_0) : $[-\frac{\pi}{2}; \frac{\pi}{2}]$
- Transmission ratio (n) : $[0; 10]$

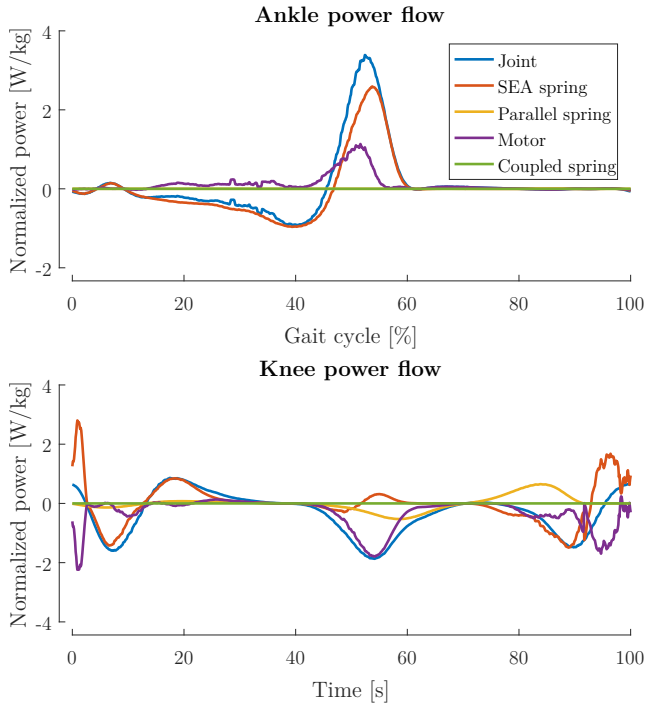
3.3 Results

The outcome of this optimisation corresponds to the parameters given in Table 3.1.

It shows that the parallel spring on the ankle joint and the coupling spring are not desirable. Indeed, optimisation converged to zero coefficients for these springs. This means that with the presented model, the coupling spring is not necessary and the problem can be separated for both joints. Fig. 3.2 shows the power flows in both joints with the optimised parameters.

Table 3.1: SEA Ankle/Knee prosthesis results

X	a	b	c	d	e	f	g	h	θ_0
K_a	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.05
k_a	-88.06	22.52	-1.73	0.49	0.00	0.02	0.16	0.40	0.00
K_k	0.05	-0.25	0.00	0.26	-14.67	-64.29	85.17	56.71	0.03
k_k	-0.14	0.22	-0.8	0.18	0.77	-10.38	-9.34	0.55	0.00
K_{ak}	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
n	0								

**Fig. 3.2:** Power flows in ankle and knee joints in different elements of the prosthesis.

Results for the ankle show that the motor is only producing energy. During the early stance phase, the SEA spring is loaded. During the late stance phase, the spring releases its energy (giving rise to a power peak below the one of the healthy ankle), while the motor is providing the missing power which is smaller. During the swing phase, almost no torque is applied, the motor is just following the ankle desired trajectory.

Results for the knee show that the motor is only absorbing energy. During the early stance phase, the SEA spring is catching the power fluctuations (positive and negative). During the late stance phase, the motor absorbs a big part of the negative energy provided by the knee, the rest being stored in the parallel spring. During the

swing phase, the parallel spring releases the stored energy, the SEA spring oscillates (storing and then releasing energy) while the motor is absorbing the released energy.

Fig. 3.3 shows the torque-angle relationship for knee and ankle elements. For the knee, it shows that for positive torque, the SEA spring is really stiff. It follows the knee characteristic during heel strike (early stance). During this phase, motor provides some torque since it is connected in series with the spring but it does not move and does not produce power. For negative torques, the SEA stiffness follows the end of the swing phase profile. The parallel spring helps during the late stance and early swing by catching power during the absorption peak and restoring it in the SEA during swing. For the ankle, the SEA spring follows the shape of the profile. The motor still has to produce the total ankle torque but with a smaller displacement. Power flows are then decreased in the motor.

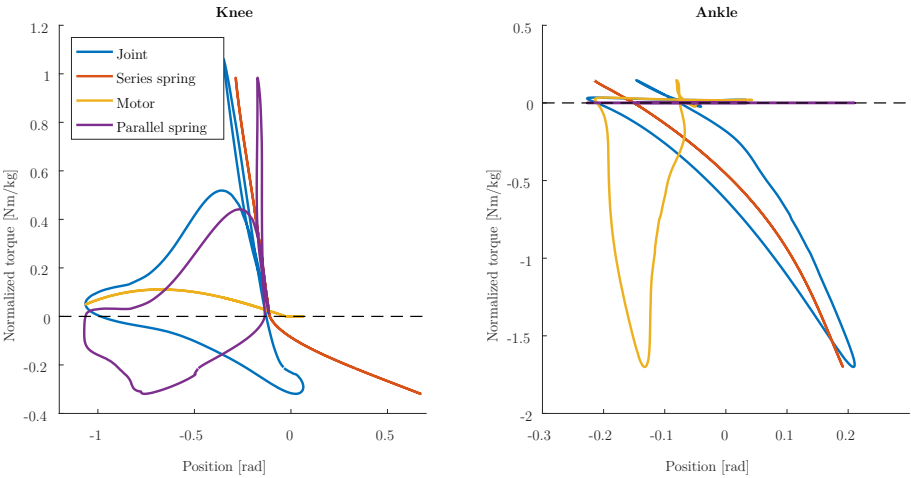


Fig. 3.3: Torque-angle relationship for knee and ankle elements.

Fig. 3.4 illustrates power flows in both motors of the prosthesis. It shows that the optimisation minimised the energy flowing in the motors. Indeed the power is always positive for the ankle and almost only negative for the knee. Since power flows keep the same sign, no energy is lost due to energy circulating back and forth in the motors. Indeed, the energy flowing through the motor correspond to the net energy needed by the joint.

3.4 Conclusion

This chapter showed that SEA and parallel springs can reduce energy consumption of the prosthesis by tuning springs stiffnesses. The outcome of the optimisation algorithm showed that these stiffnesses tend to follow the angle-torque characteristic of a sound limb. However, results are different from the prosthesis seen in the literature.

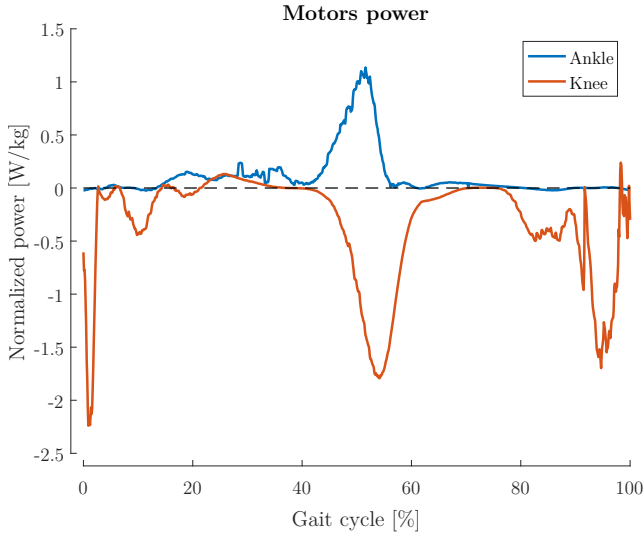


Fig. 3.4: Ankle and Knee motors power flows.

This can be explained by the fact that results of an optimisation always depend on the objective function. Another possible objective function would have been directly Joule losses in motors.

However, in the literature SEA prosthesis and the optimisation of this chapter, motors still have to produce power peaks. This is mainly due to the fact that with passive spring, energy transfers can not be controlled. Indeed, these transfers depend on the joint kinematic and dynamic. It is why locking mechanism (e.g. a clutch) are used in recent active prosthesis like the AMP-Foot [17, 18] or the CSEA knee [80]. The locking mechanism in the AMP-foot allows the motor to load a spring during swing with a lower power and release this energy rapidly during swing. This mechanism flattens the power curve of the motor which does not need to provide a large power peak during push-off. The clutch in the CSEA knee blocks the motor during early stance (i.e. during spring behavior of the knee). During this phase the motor has not to provide the reactionary torque since the clutch does.

The next chapters will focus on a different approach to control energy flows. It consists on using Continuously Variable Transmissions adjusting the power flows in the prosthesis. This approach is presented in Chapters 4 and 5.

Ankle Prosthesis: Continuously Variable Transmission

4

Publication

The material presented in this chapter is adapted from:

Ch. Everarts, B. Dehez, and R. Ronsse. Variable Stiffness Actuator applied to an active ankle prosthesis: Principle, energy-efficiency, and control. *IEEE International Conference on Intelligent Robots and Systems*, pages 323-328, 2012.

This chapter aims at presenting a new ankle prosthesis topology and a way to control it. We suggest a new approach to increase energy efficiency of an ankle prosthesis by controlling energy flows thanks to a Continuously Variable Transmission (CVT). This method, allows to decrease energy consumption of the motor compared to SEAs presented in chapter 3.

First, we apply a particular concept of VSA to an active ankle prosthesis. This concept, based on CVT, consists in storing energy in a spring with a constant power during the whole cycle and use the CVT to control energy flows (positive or negative) between the spring and the joint.

Then, we elaborate some control strategies for this concept. Our guideline is to track a mechanical design and a controller maximizing energy efficiency. We establish that a promising approach is simply to control the amount of energy stored in the elastic element.

4.1 Introduction

Walking is a complex task, involving many energy exchanges across the body joints and segments, and with the environment. Human walking splits up into two phases: the stance phase when the foot is in contact with the ground to propel the body forward. And the swing phase when the foot does not touch the ground, corresponding to a forward leg movement to take another step.

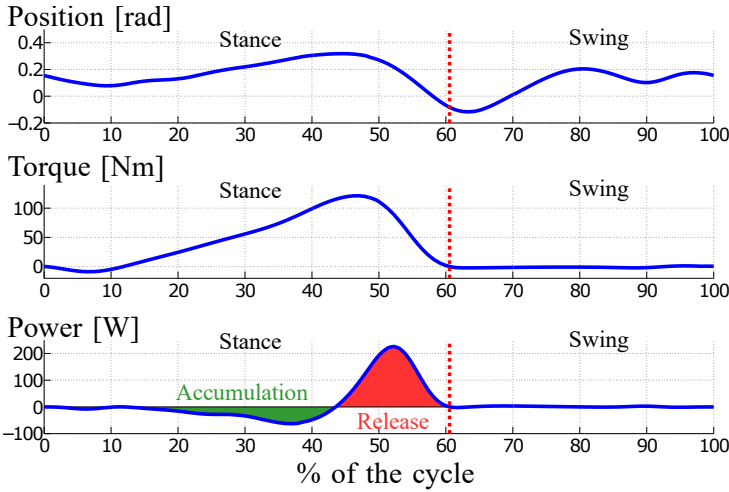


Fig. 4.1: Ankle position, produced torque and power during a representative cycle of walking for a healthy subject [97].

Figure 4.1 shows the ankle kinematics and dynamics (in the sagittal plane) of a healthy subject during self-paced walking [97]. In the lower panel, it is visible that a non-trivial energy flow occurs during the stance phase. During the load acceptance phase (about 200 ms after heel strike), the ankle absorbs energy (negative power, green area). Later, some energy is produced (positive power, red area) during the second part of the stance phase, corresponding to the body propulsion phase. The red area being larger than the green one, the net work of the ankle during self-paced walking is positive.

A purely passive ankle prosthesis, as most of the transtibial amputees are using nowadays, fails in reproducing this complex pattern. Even if the most sophisticated of them comprise an elastic element to store and release energy (like e.g. the flex-foot by Össur [72]), an actuator is necessary to retrieve the positive energy balance. Consequently, transtibial amputees walking with a passive prosthesis have to readapt the whole walking pattern, giving rise to an increase in the overall metabolic consumption of about 60% and compensatory movements by the intact joints (both ipsi- and contra-lateral) [88].

To optimally exploit the necessity to first absorb then produce energy, the most

advanced active ankle prostheses rely on the concept of Series Elastic Actuators (SEA), i.e. the serial connection of a standard actuator with an elastic element (e.g. a spring). By reducing the interface stiffness, this concept offers a lot of advantages, including greater shock tolerance, lower reflected inertia, more accurate and stable force control, less inadvertent damage to the environment, and - last but not least - the capacity to store energy [76]. Several ankle prostheses exploit this principle like e.g. the devices of Lefeber [18, 19, 21, 34, 37], Herr [3, 4, 27], Sugar [8, 48, 50] and Goldfarb [88, 90].

Putting a spring in series between the actuator and the effector in an ankle prosthesis decreases the power peak that has to be provided by the actuator. Indeed the spring stores energy at heel strike (the one that would have to be absorbed by the actuator otherwise) and releases it when the highest power peak is needed. Consequently a smaller motor can be used (the peak power is one of the key factors influencing the size of an electric motor), reducing the device cost, encumbrance and weight. For a given gait and cadence, it can be shown that it exists a particular stiffness that will bring the motor power peak to a minimum value [100], corresponding to a situation where the spring works at resonance.

An intrinsic limitation of the SEA is that the mechanical stiffness of the compliant element can be optimized only for a single task and/or user, since it is fixed. Consequently, people developed in the last few years the concept of Variable Stiffness Actuators (VSA). Relevant contributions in this field include the work of Bicchi et al. with *VSA*, *VSA-II* and *VSA-HD* [10, 15, 16, 81], Lefeber et al. with *MAC-CEPA* [56, 99], Tsagarakis et al. with *AWAS* [54, 55] and Stramigioli et al. with the bond graph study of VSA [87, 103–105]. Applying the concept of VSA to an ankle prosthesis would therefore allow to adapt the actuator stiffness not only to the user, but also to the detected gait and cadence using an appropriate adaptive controller.

This chapter's objectives are threefold: (i) to discuss the principle of VSA in the design of an active ankle prosthesis; (ii) to explore the relevance of changing the actuator stiffness not only across different tasks (i.e. across the gait cycles), but also within the gait cycle, to further improve the global energy balance; and (iii) to elaborate some concepts of controller to maximize energy-efficiency in the proposed design. A conceptual method using Infinitely Variable Transmission is presented to implement a VSA which can change its stiffness without changing the amount of stored energy in the spring; i.e. achieving high energy efficiency.

4.2 Variable Stiffness Actuators with IVT

As already mentioned, the main drawback of the SEA design is that its stiffness is optimized only for a particular gait and cadence. A natural extension of the previous approach would be to use a VSA, i.e. an elastic actuator whose intrinsic stiffness is

not fixed but can be changed by means of a second motor. Such a principle has been explored in [114]. Taking this idea a little further, the actuator's stiffness could also be changed within the cycle. As such, the main motor would just have to provide the mean positive power required at the ankle - i.e. the net positive energy provided during a walking cycle, divided by the cycle duration - while the stiffness would be constantly adapted to render the desired torque. Using data from Figure 4.1, this mean power corresponds to approximately 10 W. This approach is analogous to a decoupling between a booster motor injecting energy into a spring with constant power, and a valve releasing just the energy necessary to the ankle for walking.

To make this approach viable in terms of energy-efficiency, it is mandatory to use an efficient mechanism to change the actuator stiffness. Said differently, this would require that changing the actuator stiffness requires little to no energy, a feature that is not encountered with most of the VSA to date. In theory, changing the actuator stiffness would require no energy if its equilibrium position is changed in parallel, such that the energy stored in the elastic element is kept constant. Recently, Stramigioli et al. proposed a concept to implement this idea [87]. It consists of a motor connected in series with a torsional spring. This spring is connected to the effector via an Infinitely Variable Transmission (IVT), i.e. a transmission that can continuously change its ratio from positive to negative values, consuming virtually no energy. An IVT allows to produce the desired output torque with an arbitrary level of energy in the spring. This mechanism is therefore not a VSA *stricto sensu*, since the spring stiffness is kept constant, but renders a variable stiffness by means of the IVT. Importantly, modifying the rendered stiffness does not modify the amount of energy stored in the spring. This concept is illustrated in Figure 4.2.

The model variables are listed as following:

θ_m, ω_m	Motor position and velocity.
$\Delta\theta_s, \Delta\dot{\theta}_s$	Spring relative position (i.e. overall deformation) and velocity.
θ_t, ω_t	Spring position and velocity at the load side (before the IVT). As such, $\theta_t = \theta_m - \Delta\theta_s$
$\theta_{ank}, \omega_{ank}$	Load (i.e. in this case the ankle) position and velocity
T_{in}	Torque in the motor, spring, and at the input of the IVT
T_{ank}	Ankle torque (i.e. at the output of the IVT)
n	Instantaneous transmission ratio
k	Intrinsic stiffness of the spring

Torque and velocity equations based on the power conservation in the transmis-

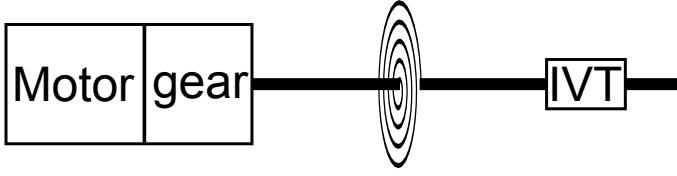


Fig. 4.2: VSA model with IVT

sion are given as follows:

$$T_{in} = -k \Delta\theta_s = \frac{1}{n} T_{ank} \quad (4.1)$$

$$\omega_t = \omega_m + \Delta\dot{\theta}_s = n \omega_{ank} \quad (4.2)$$

Spring torque $-k\Delta\theta_s$ appears negative because when the motor applies a torque in one direction, the spring deformation goes in the opposite direction: when the spring stores energy, its velocity is negative.

These equations further reduce to :

$$\omega_m + \Delta\dot{\theta}_s = -\frac{\omega_{ank} T_{ank}}{k \Delta\theta_s} \quad (4.3)$$

$$n = -\frac{T_{ank}}{k \Delta\theta_s} \quad (4.4)$$

Equation (4.4) captures the evolution of the transmission ratio that would have to be provided to exactly render the desired torque, i.e. T_{ank} . Assuming that this can be perfectly achieved, the only remaining differential equation is

$$\Delta\dot{\theta}_s = -\frac{\omega_{ank} T_{ank}}{k \Delta\theta_s} - \omega_m \quad (4.5)$$

4.3 Control Strategies

The goal of this Section is to propose different control laws for the variable ω_m , and to discuss their respective energy-efficiency, taking realistic actuator dynamics into account. Two control laws are developed: one with no feedback and another with feedback on the spring deformation.

4.3.1 Actuator Model

The classical differential equations governing the dynamics of a DC-motor system are given as following:

$$U = R I + L \frac{dI}{dt} + K \omega \quad (4.6)$$

$$T = K I - J \dot{\omega} - T_f \quad (4.7)$$

where R represents the electric resistance, L the inductance, K the torque constant of the motor, J the inertia of the rotor and T_f the friction torque. The variables U , I , T and ω represent respectively voltage, current, torque and velocity. Assuming the electrical time constant to be negligible with respect to the mechanical time constant, and eliminating I from (4.6) and (4.7), gives:

$$\dot{\omega} = \frac{K(U - K\omega)}{J R} - \frac{T + T_f}{J} \quad (4.8)$$

By combining Equations (4.5) and (4.8) and defining the motor gear ratio to be equal to χ and the gear efficiency η_{gear} , i.e. $\omega = \chi \omega_m$ and $T = \frac{-1}{\chi \eta_{gear}} k \Delta \theta_s$, the following non-linear state-space system is obtained:

$$\begin{pmatrix} \Delta \dot{\theta}_s \\ \dot{\omega}_m \end{pmatrix} = \begin{pmatrix} 0 & -1 \\ \frac{k}{J \chi^2 \eta_{gear}} & -\frac{K^2}{\chi R J} \end{pmatrix} \begin{pmatrix} \Delta \theta_s \\ \omega_m \end{pmatrix} + \begin{pmatrix} -\frac{1}{k \Delta \theta_s} & 0 & 0 \\ 0 & -\frac{1}{\chi J} & \frac{K}{\chi J R} \end{pmatrix} \begin{pmatrix} \omega_{ank} T_{ank} \\ T_f \\ U \end{pmatrix} \quad (4.9)$$

This system isolates the three inputs, namely the desired power $\omega_{ank} T_{ank}$, the friction torque T_f and the motor voltage U ; and the corresponding state variables $\Delta \theta_s$ and ω_m . Before discussing two different control laws to set the voltage U , we set the parameters to the following numerical values to be used in simulations:

- Spring stiffness : $k = 3 \text{ Nm/rad}$
- Motor features (copying those of a Maxon EC 45 flat motor of 30 W):
 - $K = 70.6 \text{ mNm/A}$
 - $R = 6.70 \Omega$
 - $T_{max} = 380 \text{ mNm}$
 - $T_n = 70.6 \text{ mNm}$
 - $\omega_n = 3210 \text{ rpm}$
- and a compatible gear box:

- $\chi = 186$ with efficiency being equal to $\eta_{gear} = 64\%$

Simulations were carried out using Matlab (the Mathworks, Natick, MA), with above numerical values and the ankle data presented in Figure 4.1.

4.3.2 Constant voltage

The first approach is to supply the motor with a constant voltage U . In a certain range, this open loop control¹ is stable.

Figure 4.3 shows results of this simulation with a voltage of 21 V. The corresponding desired transmission ratio (given by (4.4)) varies between -0.7 and 7.6 and the spring deformation oscillates between -4 and -5.5 radians. The motor power stays close to the mean power which is 10 W, it oscillates between 9.5 and 10.2 W, representing therefore a peak power reduction by a factor 16 with respect to the healthy ankle.

This approach therefore provides an elegant way to decouple the power feeding (provided by an open-loop 10 W motor) and the control (provided by the motor controlling the transmission ratio n , theoretically spending no energy).

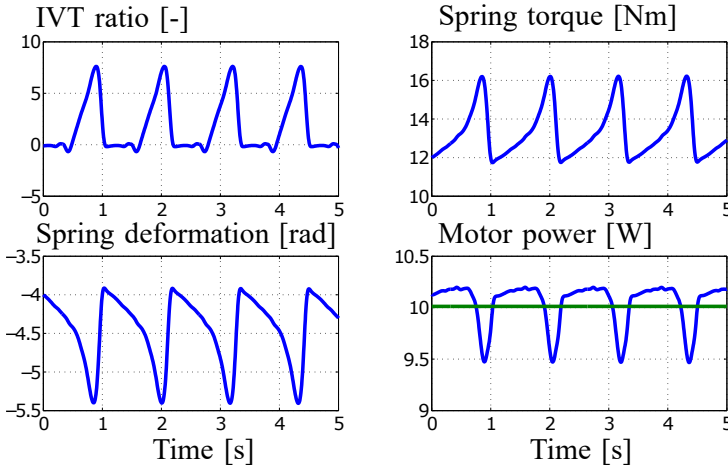


Fig. 4.3: Simulation results for constant voltage control

Figure 4.4 shows the results of the simulation for different input voltages. It shows that the system is stable up to 21 V and that the spring deformation and the motor speed are adapted to provide the needed power.

This is due to the fact that the power-torque curve of the DC motor (neglecting the dynamic effects and the gear efficiency, i.e. assuming $J = 0$) follows an

¹open loop means here that there is no feedback on the voltage U . There is of course always feedback to implement the transmission ratio, through Equation (4.4)

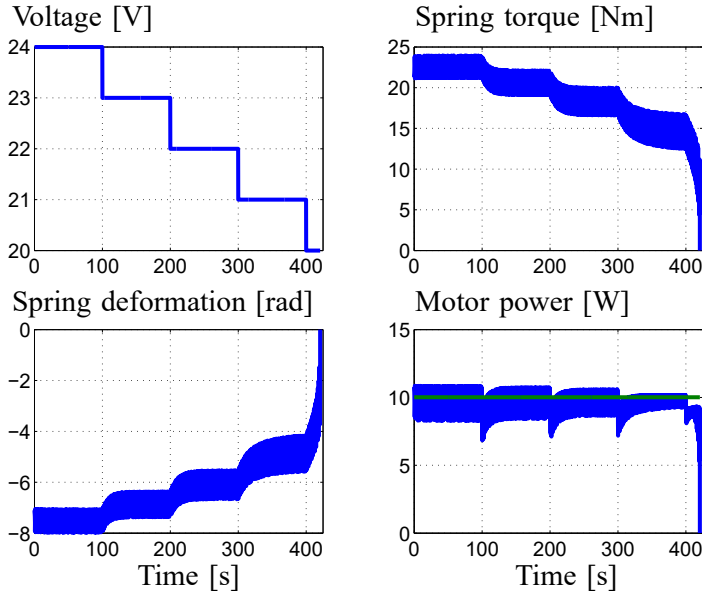


Fig. 4.4: Simulation results for different constant voltage control

inverted U-shape (see an example in Figure 4.5). The above simulations work in the descending part of the curve, therefore achieving stable behaviour: when power is required at the ankle, the spring elongation decreases, so the motor torque; due to the negative slope, this increases the motor power, tending to stabilize the spring elongation. However, this region corresponds to a small efficiency of the DC motor (below 40 %).

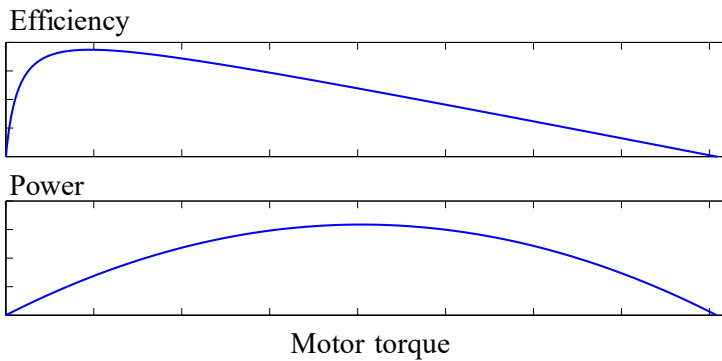


Fig. 4.5: Power/torque and efficiency/torque relationship of a DC-motor

Figure 4.6 shows the same relationships for the data from Figure 4.4 (i.e. with the dynamic effect taken into account). Importantly, the figure shows that the motor efficiency is indeed very bad, with this control, mainly with high input voltage.

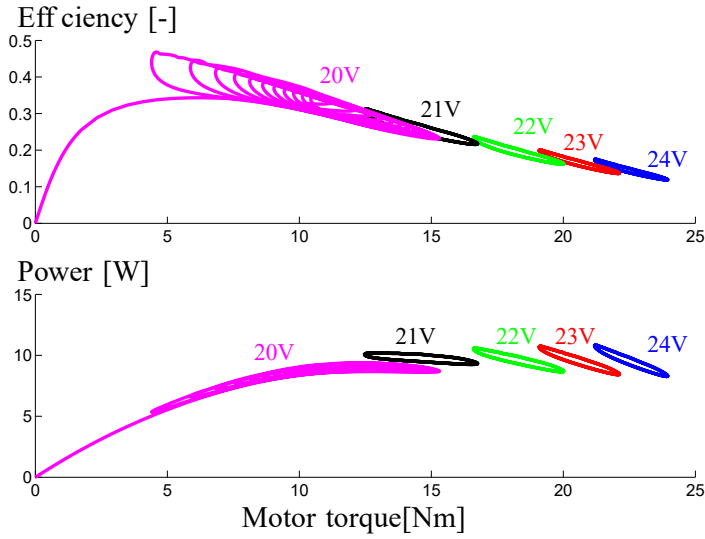


Fig. 4.6: Power and efficiency of the motor with different constant voltage

Figure 4.7 shows the electrical power, UI , and the mechanical power after the gear, $T\omega$. The mechanical power stays around 10 W but the electrical power is about 4 times higher with an input voltage of 21 V. This tendency get further worse as the input voltage increases.

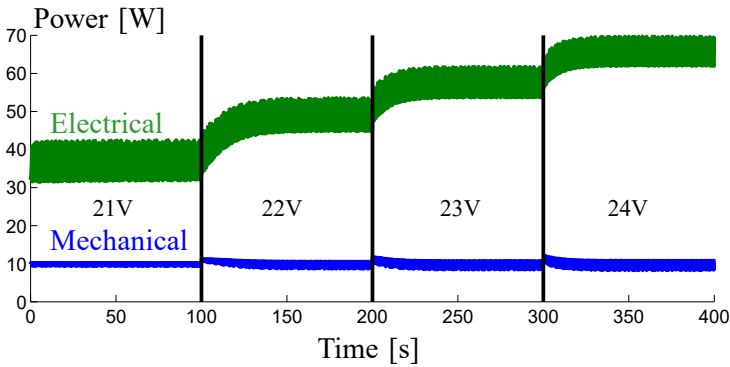


Fig. 4.7: Input (electrical) and output (mechanical) power of the motor with different constant voltages.

When a constant voltage is applied, the motor is not operating in its optimal area. To solve this issue, a second method is proposed, in which some feedback on the spring deformation is provided to control the motor.

4.3.3 Voltage versus spring deformation control

The second approach implements a feedback on the spring deformation to control the motor supply voltage.

The proportional law is written as follows :

$$U = U_c + k_p(\Delta\theta_s - \overline{\Delta\theta_s}) \quad (4.10)$$

This law combines a constant voltage similar to the feed-forward term that we used in the previous section and a proportional term stabilizing the spring deformation around $\overline{\Delta\theta_s}$. Here, we use $U_c = 23$ V, $k_p = 4 \frac{\text{V}}{\text{rad}}$, and $\overline{\Delta\theta_s} = -3$ rad.

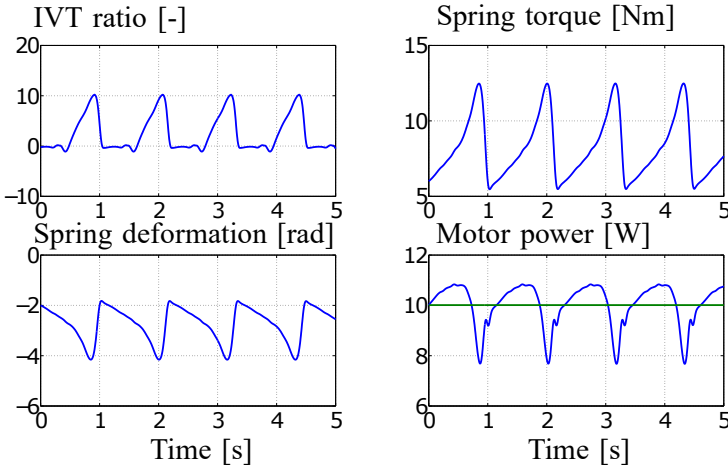


Fig. 4.8: Simulation results with varying voltage.

Using this law in Equation (4.9), the following behaviour is obtained (Figure 4.9). Spring deformation oscillates around -3 rad, this decrease the torque applied by the motor compared to the previous section (between 6 and 13 Nm instead of 12 and 16 Nm).

Figure 4.10 shows the efficiency of the motor versus the torque. With this second method, the motor operates in a better area: the efficiency oscillates between 0.25 and 0.55.

Figure 4.11 shows the electrical power, and the mechanical power. The mechanical power stays around 10 W (as before) and this time the electrical power varies between 20 and 32 W. This fluctuation is due to the fact that the motor efficiency depends on the torque which varies within the cycle.

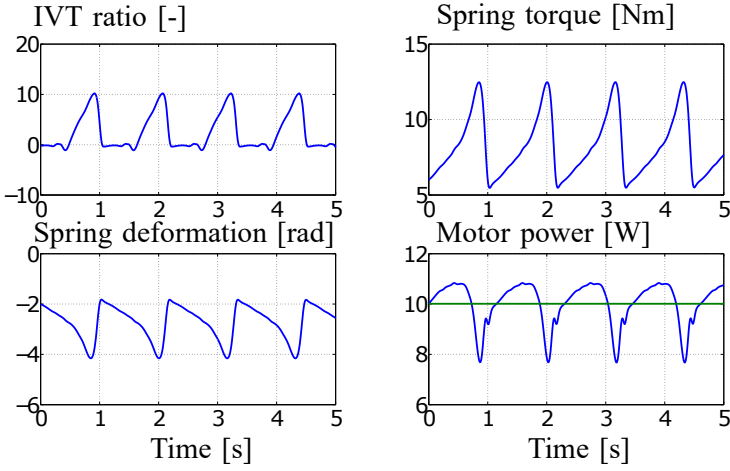


Fig. 4.9: Simulation results with varying voltage.

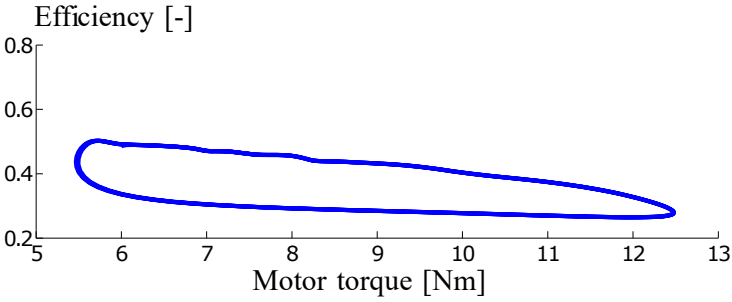


Fig. 4.10: Efficiency of the motor with controlled voltage

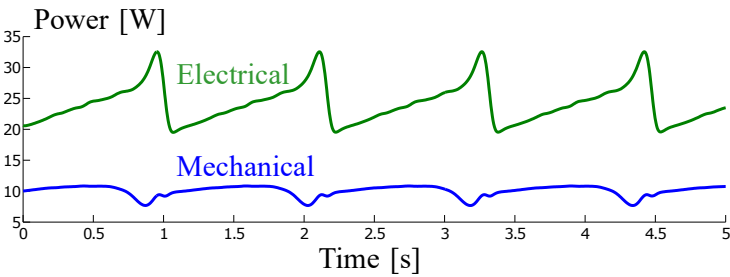


Fig. 4.11: Input and output power of the motor with controlled voltage

4.4 Conclusion

This chapter investigated the use of Variable Stiffness Actuators for the design of an active ankle prosthesis, with two main objectives.

First, we introduced the concept of VSA, i.e. an elastic actuator whose intrinsic

stiffness can be changed by means of a second motor, to circumvent the limitation of SEA. We explored potential benefits of changing the actuator stiffness not only across the cycles (i.e. to achieve resonance tuning for a given gait and cadence), but also within the cycle. This permitted to decrease the motor power by a factor 17 relative to the peak power, nearly corresponding to the minimum theoretical value, i.e. the total net energy produced by the joint during a walking cycle, divided by the cycle duration. This was achieved by means of a VSA rendering a continuously variable stiffness through a continuously variable transmission. This concept is analogous to using a flywheel in applications requiring high power peaks with a constant input power. A critical difference is that the flywheel stores kinetic energy, while the VSA stores potential energy.

Second, we proposed two control strategies for the motor feeding the spring energy. The first one was very simple, since it required the motor to be fed with a constant voltage. We showed that this strategy is stable over a wide range of input voltages, owing to an intrinsically stable torque-to-power relationship typical in DC motors. A major drawback of this approach is that it stabilized in a region where the motor efficiency is very poor, giving rise to an inefficient electromechanical conversion. This was improved by our second controller, which included a dynamical term stabilizing the spring elongation around a desired value. This permitted to improve the electromechanical conversion efficiency by about 25%.

Our results tend to validate the use of a VSA based on a Continuously Variable Transmission for the design of an active ankle prosthesis. So far, we assumed this IVT to consume no energy for changing its transmission ratio.

Ankle-Knee Prosthesis: Continuously Variable Transmission

5

Publication

The material presented in this chapter is adapted from:

R.Ronsse, M. Tummers, and **Ch. Everarts**. Modeling and Control of a Transfemoral Prosthesis Embedding two Infinitely Variable Transmissions. *IEEE International Conference on Rehabilitation Robotics*, 2017.

My contribution to this paper is the idea of the concept of a biarticular prosthesis with IVTs. I brought the idea of coupling two IVT's with a differential to a single spring and a single motor for both joints. I obtained first results for level-ground walking. I also contributed in developing the efficiency model of the differential.

This chapter aims at applying to concept developed in chapter 4 to an ankle-knee prosthesis with energy exchange between both joints.

A model of an actuation concept is reported for a transfemoral prosthesis, relying on the combination of a single power motor, a compliant element (a spring), a mechanical differential, and two Infinitely Variable Transmissions (IVTs). It allows to manage the mechanical power flows through the device in both directions (i.e. when energy should be produced or absorbed by the knee and ankle), so that the power motor does not face the sharp load power fluctuations and energy flows between both joints are possible.

The chapter further reports a preliminary approach to synthesize a closed-loop controller for this device, and simulation results of this closed-loop behavior for three locomotion tasks: level-ground walking and stair ascent/descent.

These results illustrate the capacity of this actuation principle to filter the load

Table 5.1: Average cycle period and energy being produced (if positive) or absorbed (if negative) by the the ankle and knee joints, for the three considered tasks. This energy is normalized by the subject’s weight (in kg). Data adapted from [36,78].

	Level-ground walking	Stair ascending	Stair descending
Period [s]	1.17	1.45	1.26
Ankle energy [J/kg]	0.164	0.404	-0.325
Knee energy [J/kg]	-0.223	0.671	-1.146

power profile, and further highlight the necessity to maximize the mechanical efficiency of each part of this actuation scheme.

5.1 Introduction

Ideally, active leg prostheses should be able to produce the same locomotion profiles as a sound leg. This is however a critical challenge for the prosthesis designer, because the kinematic and torque profiles of the leg joints are highly fluctuating within a gait cycle and across locomotion tasks. Fig. 5.1 shows the knee and ankle profiles for level ground walking and stair ascent/descent (data taken from [36,78]). It illustrates the above statement: over the cycle, both the velocity and torque profiles oscillate between large extremes and cross the zero line several times. So does the power profile, since it is the product of both. The prosthesis actuators and mechanical parts should thus be dimensioned in order to potentially fulfill similar specs.

Integrating the power profiles over time reveals the energy balance of the joint during the corresponding task: if this energy is positive, the corresponding joint produces energy during the cycle, e.g. to propel the body forward; if negative, the joint absorbs energy during the cycle, e.g. to absorb potential energy during stair descending (Table 5.1). Intuitively, during stair ascending (resp., descending), both joints produce (resp., absorb) energy, mainly in order to vary the level of potential energy of the body. Interestingly, the level-ground walking cycle is unbalanced: while the ankle produces energy – mainly in late stance in order to propel the body forward –, the knee absorbs. Furthermore, data reveal that the knee absorbs more than what the ankle produces, so that – in theory – globally passive transfemoral prostheses could reproduce the healthy knee and ankle profiles [68,94].

Nowadays transfemoral amputees do not use integrated prostheses, but rather combine appropriate knee and ankle modules. The simplest and cheapest ankle prostheses are passive, i.e. have no embedded actuators. They are capable of storing energy in elastic elements and returning a majority of this energy to assist in forward propulsion [43]. More advanced elastic prostheses also embed an actuated clutch, to

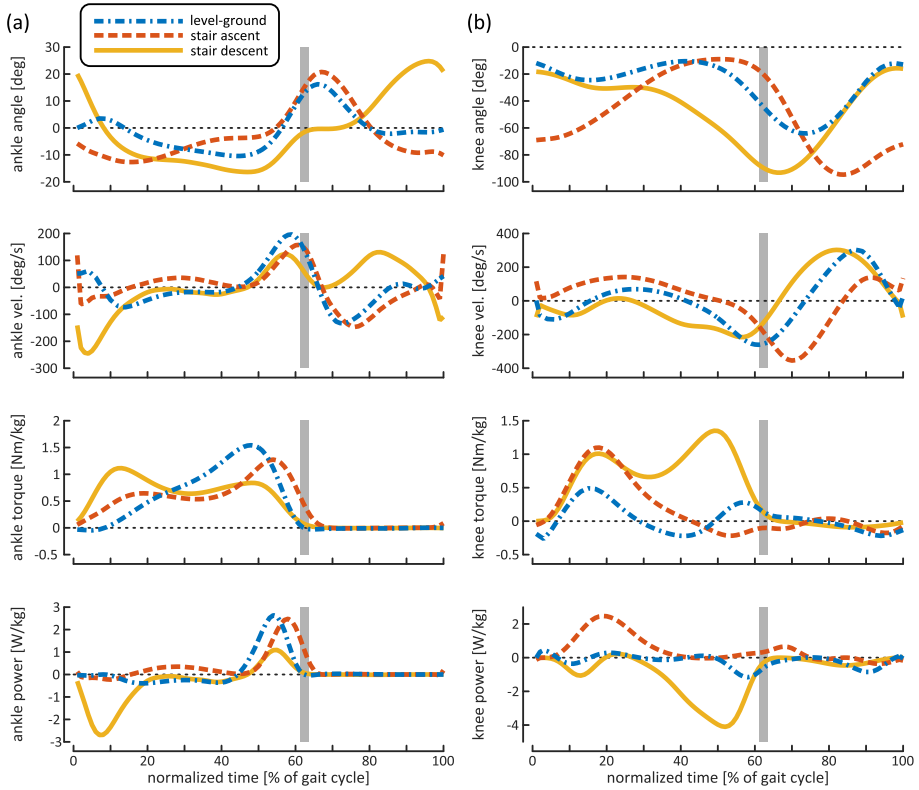


Fig. 5.1: From top to bottom: ankle (a) and knee (b) angular position and velocity, torque, and power profiles, as function of the normalized gait cycle phase (in %) and for the three different tasks. 0% and 100% capture two successive foot strikes, while the shaded gray rectangles capture the transition from stance to swing. The torque and power profiles are normalized by the subject's weight (in kg). Positive/negative values capture plantarflexion/dorsiflexion for the ankle (a), and extension/flexion, for the knee (b), respectively. Data taken from [36,78].

release the elastic energy at the most appropriate time [22]. The main drawback of these passive solutions is that they cannot reproduce the positive mechanical work of a normal ankle, even during level-ground walking [109], since they can only provide the energy that was stored during the earlier phase. This gap is bridged by the next generation of ankle prostheses, being equipped with power actuator [102]. Interestingly, these prototypes often rely on the concept of Series Elastic Actuator [76], since they embed a compliant element between their actuator and the ankle joint. This allows storing-and-releasing energy during the walking phases, such that the motor undergoes smaller displacements and thus faces smaller power peaks. Such a device recently proved to normalize the gait pattern of trans-tibial amputees, both regarding kinematics and metabolic cost [27, 46].

In contrast to the ankle, a power actuator is not necessary in a knee module to reproduce natural kinematics and dynamics, at least for level-ground walking. The simplest knee prostheses are hinge joints being locked during walking and unlocked only for sitting, thus producing unaesthetical gaits. More recent evolutions rely on servo-controlled springs and dampers. However, none of these solutions are capable of emulating a sound knee behavior during stair climbing, since they do not produce energy.

A limited amount of research prototypes addressed an integrated design for the knee and ankle replacement. Integration offers more flexibility to optimize the mechanical design and to share resources like battery or electronics, see e.g. [89]. Others exploited the unbalance in energy production and absorption of the knee and ankle joints through entirely passive transfemoral prostheses with mechanical energy transfer [68,94]. Recently, such an energy transfer mechanism was further combined with an active integrated prosthesis [37].

In this chapter, we also aim at achieving energy transfers between both joints, although we adopt a radically different framework. Moreover, our approach targets not only level-ground walking, but also stair ascent or descent, i.e. tasks with a net positive or negative energy balance [78,89]. Our approach also relies on a compliant element storing energy, but is so that this energy can be flexibly released or harvested by using Infinitely Variable Transmissions (IVTs). IVTs are special sort of Continuously Variable Transmissions (CVTs), so that they can produce both positive and negative transmission ratios. CVTs and IVTs were initially designed for the automobile industry in order to enable the car engine to run at its most efficient rotational velocity for a large range of vehicle speeds. Recently, it was extended to robotic applications requiring compliance and energy efficiency (the V2E2 concept of [87]). We [28] and others [69] studied this principle for single joint lower-limb prostheses. In particular, we figured out that such a prosthesis requires to change the IVT transmission ratio at low-to-zero rotational velocity, a feature which is not achieved by existing IVTs. Consequently, we developed our own IVT concept having the capacity to change the transmission ratio whatever the rotational velocity [29] (see Chapters 7 and 8).

In sum, an active transfemoral prosthesis which can reproduce healthy patterns in different locomotion tasks should follow highly varying kinematic and dynamic profiles, while producing or absorbing mechanical energy with one or both of its joints. The goal of this chapter is to provide a new actuation concept for such a prosthesis, and to report the benefits it could bring regarding efficiency and reduction of the power peaks. Section 5.2 builds up the model of this new actuation concept. Section 5.3 reports a preliminary design of a closed-loop controller of such a prosthesis, in order to achieve proper torque tracking at both the knee and ankle. Section 5.4 reports simulated results of this feedback controlled prosthesis, for the three tasks

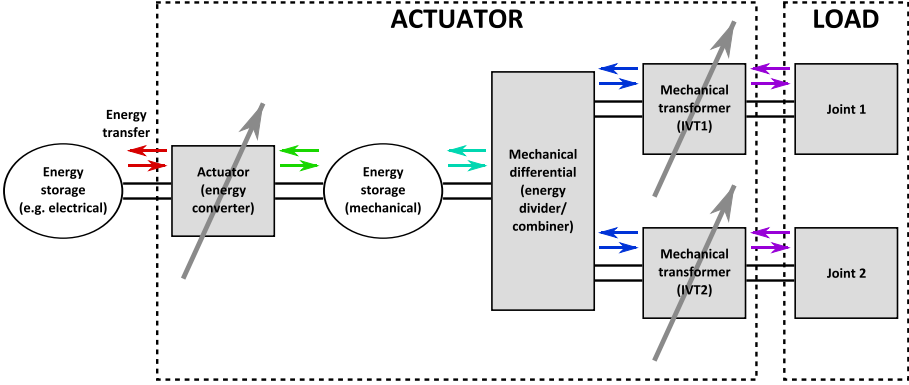


Fig. 5.2: Actuator concept for a load having two degrees of freedom, i.e. two joints. The gray arrows capture the fact that the amount of energy being converted or transferred can be varied. The color code of the power flows is consistent with the profiles of Figs. 5.4a and 5.5.

mentioned above. The chapter ends with a conclusion.

5.2 Model of a Transfemoral Prosthesis with two IVTs

This section overviews the conceptualization of an active transfemoral prosthesis achieving flexible power flows management, by embedding two IVTs. Extending the approach of [38, 87], we developed an integrated actuator for such a prosthesis by appropriately combining a power actuator, a compliant element storing mechanical energy, and *two* IVTs which modulate the output torques as a function of the available energy. More precisely, we extended the V2E2 concept of [87], as pictured in Fig. 5.2.

The V2E2 concept is a single joint actuator, having thus no mechanical differential as in Fig. 5.2. In this concept, the IVT is used to modulate its output torque $\tau_{l,out}$, i.e. the torque actuating the load joint, across positive and negative values, and whatever its input torque $\tau_{l,in}$ (pending it is $\neq 0$). Indeed, the input and output torques of an IVT are – by definition – related to each other through the instantaneous transmission ratio $\rho_l(t)$, i.e.:

$$\tau_{l,out}(t) = \rho_l(t)\tau_{l,in}(t). \quad (5.1)$$

Note that (5.1) assumes a perfect IVT, i.e. with efficiency equal to 1. This assumption will be relaxed later on.

The V2E2 concept reveals its main interest when considering that a compliant element – providing mechanical energy storage – is used as an intermediate step in

order to produce $\tau_{l,in}$. Assuming this compliant element is a spring, it produces a torque τ_s equal to the product of its stiffness k and deformation θ_s , i.e.:

$$\tau_s(t) = k\theta_s(t). \quad (5.2)$$

Combining (5.1) and (5.2) (with $\tau_{l,in} = \tau_s$) reveals that the V2E2 can provide *any* desired positive or negative torque $\tau_{l,out}(t)$, whatever $\theta_s(t)$ (pending it is $\neq 0$), by properly controlling the transmission ratio, i.e.:

$$\rho_l(t) = \frac{\tau_{l,out}(t)}{k\theta_s(t)}. \quad (5.3)$$

Consequently, if negative work is done on the load, i.e. if the load produces energy that needs to be "absorbed" by the actuator, this work would not be lost but instead stored in the compliant element. In sum, this actuator could exactly render any physical impedance by storing or releasing energy in/from the spring. A consequence of this statement – critical for applications with varying power profiles – is that the compliant element can be used as a buffer of mechanical energy, so that the power actuator providing (and absorbing) mechanical energy to/from the spring faces a much smoother power profile than the one of the load [28].

In the present chapter, we extend this concept for actuating two degrees of freedom, i.e. the knee and ankle joints of a lower-limb prosthesis. This is achieved by augmenting the V2E2 concept as pictured in Fig. 5.2, i.e. with a mechanical differential distributing the mechanical power flow through both joints. Interestingly, this architecture offers not only to store negative work produced by the load and to smooth the power profile, but also to achieve this *across* both joints. Said differently: (i) if one joint produces energy while the other absorbs (like in level-ground walking), the power flows through the differential naturally adapt; and (ii) the main actuator faces a power profile being a smoothed version of the *sum* of both joints' profiles. These statements will be detailed in the following sections, providing a model of the successive elements of Fig. 5.2. These models follow the concept of power ports like in [87]. They consist in every domain of a coupled pair of variables called effort and flow. In the electric domain, these are the voltage (effort) and current (flow); in the mechanical domain, these are the force or torque (effort) and (rotational) velocity (flow). Importantly, the product of the effort and the corresponding flow is equal to the power being instantaneously transferred through that port, whatever the domain. Flows are considered to be positive if entering into the corresponding port, both from the "input" and "output" sides. This explains the sign convention adopted in Eqs. (5.4), (5.7), (5.10), (5.11), and Table 5.2.

5.2.1 Model of the loads

Ideally, the loads should be modeled as admittance operators: each load joint rotates with a velocity ω_l depending on the torque it receives from the IVT $\tau_l = \tau_{l,out}$,

internal state variables and dynamical parameters. However, when the load joints are the knee and ankle of an active prosthesis, these operators should account not only for the prosthesis dynamics, but also for the dynamics of the rest of the body and – more critically – for the adaptive motor control strategy of the amputee walking with the prosthesis. In order to avoid this huge modeling effort, in the present work, we assume the load joints to always follow the kinematic profiles from Fig. 5.1, whatever the torque they receive. We assume that this provides a good approximation of the whole device dynamic behavior when the steady-state torques and kinematic profiles are close to those of Fig. 5.1. Note finally that the loads are not necessarily passive: Table 5.1 revealed that – in some cases – the leg joints produce energy through the gait cycle.

5.2.2 Model of the IVTs

An IVT is a mechanical transmission whose output velocity $\omega_{l,out}$ equals the input one $\omega_{l,in}$, divided by the time-varying transmission ratio ρ_l :

$$\omega_{l,out}(t) = -\omega_{l,in}(t)/\rho_l(t). \quad (5.4)$$

Ideally, power is conserved through an IVT, so that the input/output torques obey the opposite relationship (see (5.1)). However, in the real world, no mechanical transmission displays a perfect efficiency. This fact must be accounted for in a contribution like the present one, where the global energetic performance of the device is a central point. Therefore, (5.1) is replaced by:

$$\tau_{l,out}(t) = \eta_l(t)\rho_l(t)\tau_{l,in}(t) \quad (5.5)$$

where $\eta_l(t)$ is the (time-varying) IVT efficiency. For the sake of simplicity, we will neglect situations where the IVT gears are stuck (see e.g. [73] for more details). Consequently, the IVT efficiency will be captured by a coefficient smaller than one if the IVT brings power to the joint and the inverse of this coefficient otherwise. In mathematical terms:

$$\eta_l(t) = \begin{cases} \tilde{\eta}_l & \text{if } \tau_{l,in}(t)\omega_{l,in}(t) > 0 \\ 1 & \text{if } \tau_{l,in}(t)\omega_{l,in}(t) = 0 \\ 1/\tilde{\eta}_l & \text{if } \tau_{l,in}(t)\omega_{l,in}(t) < 0 \end{cases} \quad (5.6)$$

where $0 \leq \tilde{\eta}_l < 1$.

5.2.3 Model of the differential

A mechanical differential has three mechanical shafts, thus its model has three ports. Since it is a passive device which cannot store energy, at every moment, at least one

of them must face a positive power while at least another faces negative power. Following the layout of Fig. 5.2, the port on the spring side is called the “input”, and the ports on the IVTs side the “outputs”. The velocity relationship is thus:

$$\chi_d \omega_{d,out1}(t) + (1 - \chi_d) \omega_{d,out2}(t) = -\omega_{d,in}(t) \quad (5.7)$$

where $\omega_{d,in}$, $\omega_{d,out1}$, and $\omega_{d,out2}$ denote the input and outputs velocities, respectively, and $0 < \chi_d < 1$ is a constant parameter capturing possibly different transmission ratios between the input and both outputs. Similarly, the torque relationships are:

$$\tau_{d,out1}(t) = \eta_{d1}(t) \chi_d \tau_{d,in}(t), \quad (5.8)$$

$$\tau_{d,out2}(t) = \eta_{d2}(t) (1 - \chi_d) \tau_{d,in}(t). \quad (5.9)$$

Similarly to η_ℓ in (5.5), η_{d1} and η_{d2} capture the time-varying efficiencies of the differential, varying as a function of the direction of the power flows:

$$\eta_{d1}(t) = \begin{cases} \tilde{\eta}_d & \text{if } \tau_{d,in}(t) \omega_{d,in}(t) > 0 \text{ and } \tau_{d,out1}(t) \omega_{d,out1}(t) \leq 0 \\ 1 & \text{if } \tau_{d,in}(t) \omega_{d,in}(t) > 0 \text{ and } \tau_{d,out1}(t) \omega_{d,out1}(t) > 0 \\ 1 & \text{if } \tau_{d,in}(t) \omega_{d,in}(t) \leq 0 \text{ and } \tau_{d,out1}(t) \omega_{d,out1}(t) \leq 0 \\ 1/\tilde{\eta}_d & \text{if } \tau_{d,in}(t) \omega_{d,in}(t) \leq 0 \text{ and } \tau_{d,out1}(t) \omega_{d,out1}(t) > 0 \end{cases} \quad (5.10)$$

where $0 \leq \tilde{\eta}_d < 1$ is the coefficient governing efficiency as a function of the power flows direction. A similar equation exists for $\eta_{d2}(t)$.

5.2.4 Model of the compliant element (spring)

The compliant element is a spring whose torque-to-deformation relationship has been given in (5.2). Importantly, a spring is a dynamical element. It is thus governed by a state equation capturing the rate of change of its deformation as a function of its input and output velocities:

$$\dot{\theta}_s(t) = \omega_{s,out}(t) + \omega_{s,in}(t). \quad (5.11)$$

5.2.5 Model of the power actuator

The actuator in Fig. 5.2 is in charge of the conversion between the mechanical energy being absorbed or produced by the joints, and another type of energy being stored. We assume this actuator to be a DC motor, so that this external energy is electrical, stored e.g. in a battery. Equations capturing the dynamics of such a motor are typically:

$$u_m(t) = K_m \omega_{m'}(t) + R_m i_m(t); \quad (5.12)$$

$$J_{m'} \dot{\omega}_{m'}(t) = K_m i_m(t) - \tau_f(t) - \tau_{m'}(t). \quad (5.13)$$

Table 5.2: Effort and flow relationships between the blocks in Fig. 5.2.

	Effort	Flow
Actuator to spring	$\tau_m = k\theta_s$	$\omega_{s,in} = \omega_m$
Spring to differential	$\tau_{d,in} = k\theta_s$	$\omega_{d,in} = -\omega_{s,out}$
Differential to IVTs	$\tau_{i1,in} = \tau_{d,out1}$ and $\tau_{i2,in} = \tau_{d,out2}$	$\omega_{i1,in} = -\omega_{d,out1}$ and $\omega_{i2,in} = -\omega_{d,out2}$
IVTs to loads	$\tau_{l1} = \tau_{i1,out}$ and $\tau_{l2} = \tau_{i2,out}$	$\omega_{l1} = -\omega_{i1,out}$ and $\omega_{l2} = -\omega_{i2,out}$

In (5.12), u_m denotes the input voltage (effort) and i_m the motor current (flow); R_m and K_m are the motor parameters, i.e. the armature resistance and back electro-motive constant; and ω_m is the state variable, i.e. the motor shaft velocity. Note that this so-called electrical equation neglects the armature inductance, which is a frequent assumption [84]. Eq. (5.13) captures the relationship between the motor current and produced torque $\tau_{m'}(t)$, $J_{m'}$ being the motor inertia, and $\tau_f(t)$ capturing all friction torques in the motor. Such a motor would often be connected to a gearbox of ratio χ_g , whose output velocity and torque are thus given by:

$$\omega_m(t) = \omega_{m'}(t)/\chi_g, \quad (5.14)$$

$$\tau_m(t) = \eta_g(t) (\chi_g \tau_{m'}(t) + J_g \dot{\omega}_m(t)). \quad (5.15)$$

where $\eta_g(t)$ captures the gearbox efficiency and obey a similar equation as (5.6), and J_g captures the gearbox inertia. Rearranging (5.12-5.15) gives the motor+gearbox state equation:

$$J_m \dot{\omega}_m(t) = \frac{-K_m^2}{R_m} \omega_m(t) + \frac{K_m}{\chi_g R_m} u_m(t) - \frac{1}{\chi_g} \tau_f(t) - \frac{1}{\eta_g(t) \chi_g^2} \tau_m(t), \quad (5.16)$$

where $J_m = J_{m'} + J_g/\chi_g^2$ is the equivalent motor+gearbox inertia. By convention, we consider ω_m to be positive when going out of the motor port. It is thus equal to the opposite of the motor's output flow.

5.2.6 Block connections

The various blocks modeled in the sections above are actually connected to each other, according to Fig. 5.2. Therefore, trivial relationships exist between the effort and flow variables of the different blocks, as summarized in Table 5.2. In this table, variables associated to the first and second loads and IVTs are denoted with a subscript 1 and 2, respectively.

5.2.7 Model of the IVT actuation units

Last but not least, this section provides a model of the IVT actuators, i.e. the mechanisms actually varying the transmission ratios ρ_{l1} and ρ_{l2} . This model relies on the IVT design reported in Chapters 7 and 8. In short, the transmission ratio of this IVT is changed by controlling the position x_l of an external ring, rolling on the tilted conical planets of a modified planetary gear. The relationship between x_l and the actual transmission ratio is (see Chapters 7 and 8 for details):

$$\rho_l(t) = 1 - \frac{R}{\sin \alpha x_l(t)}, \quad (5.17)$$

where R is a gain depending on the size of the various IVT gears, and α is the inclination angle of the tilted planets. Controlling x_l is achieved through a lead screw and a similar DC motor as the one described in Section 5.2.5:

$$J_{ml}\xi\ddot{x}_l(t) = \frac{-K_{ml}^2}{R_{ml}}\xi\dot{x}_l(t) + \frac{K_{ml}}{R_{ml}}u_{ml}(t), \quad (5.18)$$

where ξ is the lead screw pitch, and J_{ml} the equivalent inertia of the motor, lead screw and IVT ring. We assume that this mechanism faces no friction.

5.3 Closed-Loop Control

The model outlined in Section 5.2 is a redundant system: it has 3 inputs, represented by the arrows in Fig. 5.2 (the voltage of the main power actuator u_m and the voltages of the IVT actuators u_{ml1} and u_{ml2}), but only 2 reference signals to track, namely the reference torques to be transmitted to the joints ($\tau_{ref,l1}$ and $\tau_{ref,l2}$). In this section, we report synthesis methods for designing closed-loop controllers achieving the tracking of these 2 references by commanding these 3 inputs.

5.3.1 Closed-loop control of the IVT actuators

Ideally, the desired transmission ratio for each of the IVTs is obtained from a relationship similar to (5.3), yet generalized to our 2 degrees-of-freedom configuration: $\rho_{ref,l}(t) = \tau_{ref,l}(t)/\tau_{l,in}(t)$, assuming that $\tau_{l,in}$ can be properly measured. However, because the IVT is known to have non-ideal efficiency (see (5.6)), the actual torque transmitted to the load τ_l will not be equal to $\tau_{ref,l}$. Therefore, the relationship between the reference output torque and the corresponding transmission ratio is augmented with error corrective terms:

$$\begin{aligned} \rho_{ref,l}(t) = & \frac{\tau_{ref,l}(t)}{\tau_{l,in}(t)} + k_{p,l}(\tau_{ref,l}(t) - \tau_l(t)) \\ & + k_{i,l} \int (\tau_{ref,l}(t) - \tau_l(t)) dt, \end{aligned} \quad (5.19)$$

where $k_{p,l}$ and $k_{i,l}$ are, respectively, the proportional and integral gains of this error correction mechanism. There are little clues to achieve model-based synthesis of these gains, since the corresponding dynamics depends on the efficiency model and is thus highly non-linear. Next, $\rho_{ref,l}$ is saturated between the physical limits of the IVT, namely $\rho_{min,l}$ and $\rho_{max,l}$, and an anti-windup mechanism must be implemented to switch off the integral action when this saturation is reached.

Finally, this reference transmission ratio $\rho_{ref,l}$ is converted into a reference position of the IVT actuator $x_{ref,l}$ via (5.17). This reference position can be controlled via a classical proportional-derivative controller, whose transfer function is given by:

$$\frac{U_{ml}(s)}{\Delta X_l(s)} = k_{p,ml} + k_{d,ml} \frac{k_{n,ml}}{1 + \frac{k_{n,ml}}{s}}, \quad (5.20)$$

where $\Delta X_l(s)$ is the Laplace transform of the position error, i.e. $\Delta x_l(t) = x_{ref,l}(t) - x_l(t)$, and $k_{p,ml}$, $k_{d,ml}$, and $k_{n,ml}$ are, respectively, the controller proportional and derivative gains, and the filter coefficient for the derivative action. Note that it is theoretically possible to achieve a model-based synthesis of these gains, since the corresponding open-loop dynamics are given by (5.18). In sum, an overview of the closed-loop controller of one IVT actuator is provided in Fig. 5.3.

5.3.2 Closed-loop control of the main power actuator

Since the IVT actuators already take care of transmitting the exact desired torques $\tau_{ref,l}$ to the joints, the purpose of the main power actuator controller is only to guarantee that the spring deformation never drops to 0 (see Section 5.2). In order to achieve this, we implemented a closed-loop proportional-integral controller maintaining a desired deformation in the spring $\theta_{ref,s}$ (and thus a desired level of energy), i.e.:

$$u_m(t) = k_{p,m}(\theta_{ref,s} - \theta_s(t)) + k_{i,m} \int (\theta_{ref,s} - \theta_s(t)) dt, \quad (5.21)$$

Table 5.3: Numerical values given to the model parameters. Number in parentheses refer to the corresponding equations.

Power motor and gearbox: (5.16)		Ankle IVT only: (5.17),(5.19)	
equivalent inertia	$J_m = 1.47 \cdot 10^{-7}$ [kgm ²]	min ratio	$\rho_{min,\iota 1} = -1.5$ [-]
armature resistance	$R_m = 0.569$ [Ω]	max ratio	$\rho_{max,\iota 1} = 0.0335$ [-]
back emf constant	$K_m = 5.25 \cdot 10^{-3}$ [Nm/A]	gear gain	$R_1 = 0.0145$ [m]
		Knee IVT only: (5.17),(5.19)	
friction torque	$\tau_f = 0$ [Nm]	min ratio	$\rho_{min,\iota 2} = -0.932$ [-]
ratio	$\chi_g = 208.8$ [-]	max ratio	$\rho_{max,\iota 2} = 0.253$ [-]
efficiency	$\tilde{\eta}_g = 0.7$ [-]		
Motor of both IVTs: (5.18)		gear gain	$R_2 = 0.0112$ [m]
equivalent inertia	$J_{m\iota} = 1.06 \cdot 10^{-7}$ [kgm ²]	Controller of the power motor: (5.21)	
armature resistance	$R_{m\iota} = 1.53$ [Ω]	P gain	$k_{p,m} = 1.29$ [V/rad]
back emf constant	$K_{m\iota} = 5.29 \cdot 10^{-3}$ [Nm/A]	I gain	$k_{i,m} = 0.395$ [V/(rad s)]
Controller of both IVTs: (5.19-5.20)		Compliant element (spring): (5.2)	
torque corrective term - P gain	$k_{p,\iota} = 0.4$ [1/Nm] [1/Nm]	stiffness	$k = 0.04$ [Nm/rad]
torque corrective term - I gain	$k_{i,\iota} = 0.016$ [1/(Nm s)]	Differential: (5.7-5.10)	
		efficiency	$\tilde{\eta}_d = 0.95$ [-]
actuator controller - P gain	$k_{p,m\iota} = 173$ [V/m]	symmetry	$\chi_d = 0.546$ [-]
actuator controller - D gain	$k_{d,m\iota} = -1.82$ [Vs/m]	Ankle and knee IVTs: (5.6),(5.17),(5.18)	
		efficiency	$\tilde{\eta}_\iota = 0.6$ [-]
actuator controller - D filter	$k_{n,m\iota} = 548$ [1/s]	planet angle	$\alpha = \pi/4$ [-]
		lead screw pitch	$\xi = 1.43 \cdot 10^3$ [rad/m]

where $k_{p,m}$ and $k_{i,m}$ are, respectively, the proportional and integral gains of this controller. A similar anti-windup mechanism as the one pictured in Fig. 5.3 is moreover implemented to cope with voltage limits. Again, note that the closed-loop dynamics of this system can be – in theory – achieved through model-based gain synthesis, since the corresponding open-loop dynamics are given by (5.16). This requires however to treat both the friction torques and the load torque induced by the spring as unmodeled disturbances.

5.4 Results and Discussion

The numerical values that were given to the many parameters of this model are given in Table 5.3. The rationales for selecting these values are reported in the following paragraphs.

The power motor parameters were taken from the datasheet of the 60W 'EC16-395558' brushless motor from Maxon Motor AG (Sachseln, Switzerland). Its gearbox parameters were taken from the datasheet of the 'GP22A-134162' from the same company. Note that we assumed the motor to have no friction ($\tau_f = 0$), so that energy losses in the power motor+gearbox are only captured by the gearbox non-ideal efficiency. Similarly, the IVT motors parameters were taken from the datasheet of the 30W 'EC16-400161' brushless motor from the same company.

The spring stiffness was selected to be quite compliant. If too stiff, large energy variations would correspond to large displacements on the motor side, thus not flattening the power profile. In contrast, a smaller stiffness would require to take a target spring deformation $\theta_{ref,s}$ that is not reachable with a physical spring, in order to store the appropriate amount of energy.

The geometrical parameters of the differential and IVTs were selected to be compliant with the IVT design of Chapters 7 and 8 and as a function of the torque peaks that have to be transmitted to the ankle and knee (see Fig. 5.1). The IVT efficiency was also taken from the experiments reported in Chapters 7 and 8 and other unpublished data.

The power motor controller gains were selected to achieve a closed-loop bandwidth of 0.6Hz with critical damping. This relatively small bandwidth (actually below the one of the tasks themselves, see Table 5.1) was selected so that the spring could actually undergo large deformations, thus carrying out its duty of flattening the power profile.

In contrast, the power IVT controllers gains were selected to achieve a much higher closed-loop bandwidth, i.e. 60Hz, also with critical damping. In this case, the position closed-loop dynamics must be as high as possible, to make x_l following $x_{ref,l}$ with the lowest possible tracking error.

We simulated 20s of the closed-loop behavior of this transfemoral prosthesis in Matlab/Simulink (R2015a; the MathWorks, Natick, MA), for the three different tasks, and for a subject of 75kg. All simulations were generated by using the variable-step solver 'ode45'. The desired torque profiles $\tau_{ref,l}$ were constructed by concatenating several times the profiles of Fig. 5.1, and dividing this by a factor -100 , capturing the effect of a final transmission stage (e.g. a lead screw) that was not accounted for in our model. Similarly, the resulting load velocities ω_l were constructed by concatenating the corresponding velocity profiles, this time multiplied by the same

factor -100 . Note that the laws governing the real-time efficiencies (Eqs. (5.6) and (5.10)) were actually replaced by smoother sigmoid functions, for numerical issues. These functions were such that they reach 95% of their plateaus for about $\pm 7\text{W}$ of transmitted output power. In the following, we report only graphs showing the steady-state regime, i.e. the last two of the simulated cycles.

For level-ground walking, the target spring deformation was set to $\theta_{ref,s} = 54.8\text{rad}$. This corresponds to an energy stored in the spring equal to $k\theta_{ref,s}^2/2 = 60\text{J}$, i.e. about twice the energy produced by the power motor during a level ground walking cycle. Fig. 5.4a reports the profiles of the power flows going through the different ports of the device. It highlights two important facts: (i) as power progresses through the device (i.e. from the electrical power supplied to the actuator to the power actually measured across the loads), electrical and mechanical losses cause power drops; and (ii) the power profiles before the IVTs are much smoother than the ones after. Fact (i) is confirmed by Table 5.4, reporting the energy balance (i.e. the time-integral of the power curves) between the different parts. Note that the small discrepancy of energy balance at both sides of the spring (while the spring model was loss-less) is likely because the simulation did not reach a pure steady-state after 20 cycles. As expected, the largest power/energy drop is happening in the IVTs themselves, since their efficiency was equal to 60% only. Indeed, while the load energy balance is dissipative (19.9J are produced by the knee while only 11.1J are provided to the ankle), this balance is reversed at the other side of the IVTs: only 7.4J can be harvested from the knee IVT, while 28.7J must be supplied to the ankle IVT. At the end, the electrical actuator consumes 38.9J of electrical energy, so that the overall device efficiency is actually of -23% . This negative result should however be taken with care, since the current modeling of the IVT efficiency (see (5.6)) is only a crude image of the actual one. Indeed, a more realistic modeling of our IVT efficiency should make it dependent on the transmission ratio, so that higher efficiencies would be precisely faced when the largest amount of mechanical power has to be transmitted (see e.g. [67] for more details).

Fact (ii) is much more favorable for our design. Indeed the mechanical power supplied by the main actuator oscillates between 11.3W and 56.4W (with an electrical power supply varying from 16.2W to 62.0W), while the mechanical power transmitted to the loads fluctuates between -131.3W (knee) and 179W (ankle). The use of two IVTs, a differential, and a spring thus reduced the power peak by a factor 3. The spring plays a significant role in this mechanical buffering, storing an energy which varies between about 44.6J and 71.2J (see Fig. 5.4b). These fluctuations are caused by the large angular velocities of the spring output shaft, while the input shaft rotates with a much more stationary profile (see Fig. 5.4c), both being images of the corresponding power flows.

The torques being transmitted to both joints are reported in Fig. 5.4d,f. They

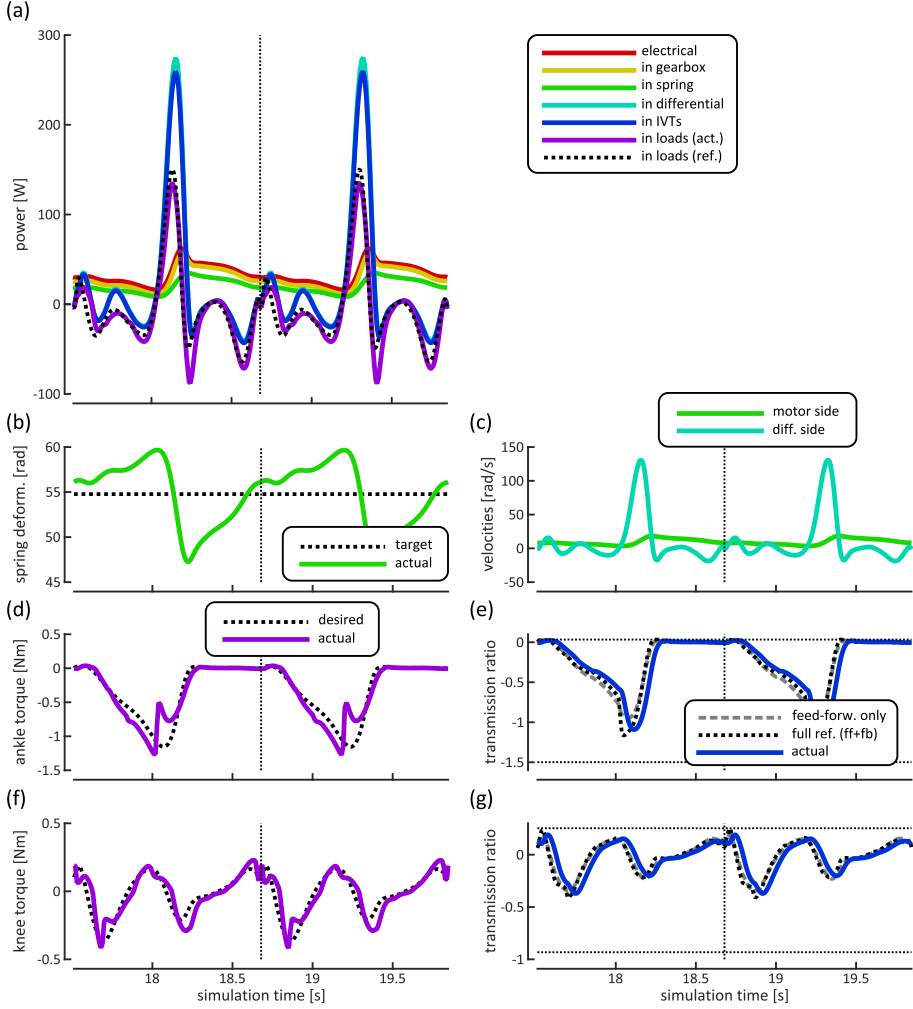


Fig. 5.4: Simulated level-ground walking. (a): profiles (as a function of time) of the power flows circulating between the different parts of the actuator, as represented in Fig. 5.2. (b): actual spring deformation $\theta_s(t)$, compared to the reference $\theta_{ref,s}$. (c): input ($\omega_m(t)$, motor side) and output ($\omega_{d,in}(t)$, differential side) shaft velocities, at both sides of the spring. (d): desired $\tau_{ref,l1}(t)$ and actual $\tau_{l1}(t)$ ankle torque. (e): transmission ratio of the ankle IVT: feed-forward only (i.e. the first term of (5.19)), full feedback-corrected reference $\rho_{ref,l1}(t)$ (i.e. all terms of (5.19)), and actual ratio $\rho_{l1}(t)$. (f,g): same as (d,e), for the knee joint. In (e,g), the horizontal lines represent the physical limits of the transmission ratios, i.e. $\rho_{min,l}$ and $\rho_{max,l}$. The vertical lines capture the separation between the two represented gait cycles.

are controlled via the IVTs transmission ratios ρ_{l1} and ρ_{l2} , reported in Fig. 5.4e,g. This highlights both the role of the feedback-driven terms of (5.19) correcting for the mechanical losses in the IVTs, and the closed-loop dynamics of the IVTs' actuators

Table 5.4: Energy balance during the last cycle of the power profiles reported in Figs. 5.4a and 5.5, in Joules. If Positive, energy is produced; if negative, it is absorbed.

	Level-ground walking	Stair descending	Stair ascending
Electrical	38.9 J	-50.4 J	218.4 J
in Gearbox	33.6 J	-52.4 J	173.8 J
in Spring	23.8 J	-74.9 J	121.8 J
in Differential	24.1 J	-74.7 J	121.6 J
in Ankle IVT	28.7 J	-7.8 J	46.3 J
in Knee IVT	-7.4 J	-71.7 J	68.7 J
TOTAL	21.3 J	-79.5 J	115.1 J

in Ankle load (ref)	11.1 J (12.3 J)	-27.1 J (-24.5 J)	25.4 J (30.3 J)
in Knee load (ref)	-19.9 J (-16.7 J)	-120.6 J (-86.0 J)	38.4 J (50.3 J)
TOTAL (ref)	-8.8 J (-4.4 J)	-147.7 J (-110.4 J)	63.7 J (80.6 J)

((5.18) and (5.20)). Altogether, these dynamics explain the differences between the desired and actual joint torques (Fig. 5.4d,f), also giving rise to differences between the reference and actual power profiles (see Fig. 5.4a) and joint energy balance (see Table 5.4).

Fig. 5.4d,f reveals that the torques transmitted to the loads face significant drops at some phases of the cycle. This is particularly visible in the middle of the ankle negative torque peak. These drops happen when the load shaft velocity (and consequently, the power) crosses the zero line or, said differently, when there is a shift between energy production and absorption and thus, according to (5.6), when the IVT efficiency abruptly switches to a different value. Again, it is likely that a real mechatronic device would behave differently: the torque drop would induce a drop in the shaft acceleration, and thus a smoother change in the load velocity and power. In sum, the loads model (see Section 5.2.1) should be revisited to account for these complex dynamics.

Fig. 5.5 shows the same power profiles as Fig. 5.4a, for both other tasks (stair descent/ascent). The corresponding energy balances are also reported in Table 5.4. The same observations as for level-ground walking can be reported: a dissipation of energy across the device, and a filtering of the power profile through the spring. Stair descent is purely dissipative: while 147.7J could be harvested from both joints during each of the cycles, only 50.4J can be harvested and potentially stored in the electrical battery. The device efficiency is thus equal to 34%. On the other hand, the

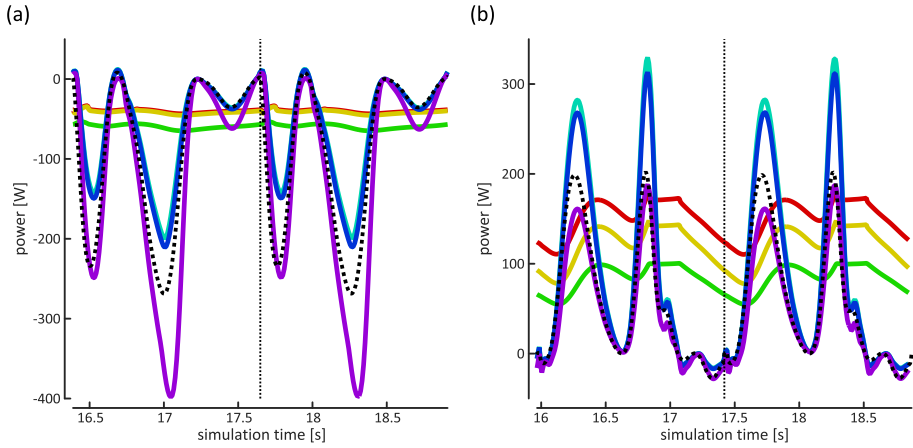


Fig. 5.5: Power profiles of simulated stair descent (a) and ascent (b). See Fig. 5.4a for the color code.

power peak reduction is this time of about a factor 9.

Stair ascent requires – in contrast – to produce energy through both of the joints. More precisely, 63.7J are transmitted to the joints during a cycle, requiring to produce 218.4J of electrical energy. The global efficiency is thus, in this case, equal to 29%. This time, the power peak reduction is only of about a factor 1.3. Note that this last simulation required to increase the target spring deformation to $\theta_{ref,s} = 141.4\text{rad}$, corresponding to a stored energy of 400J, i.e. again about twice larger than the energy spent by the electrical motor over a cycle. This increase was necessary in order to produce a larger spring torque, and thus to keep the transmission ratios within their respective boundaries. As a consequence, the motor current is higher, and thus the electrical losses within the motor are larger. Note finally that the peak power of the main actuator (180W as electrical and 148W as mechanical power) is much too high for the 60W motor that we selected in the simulated design. Improving this would require to improve the IVTs efficiency (again), to upgrade the device’s main actuator, or to perform the stair ascending task at a slower pace than the one reported in Table 5.1.

5.5 Conclusions

Motivated by the challenge to design light-weight and energy efficient lower-limb prostheses, this chapter presented a novel actuation concept for a transfemoral prosthesis. It relied on two IVTs modulating the torque transmitted to both the ankle and knee joints, a mechanical differential conveying the energy flows in both directions (production/absorption), and a single compliant element (spring) filtering the power profile between the load and the motor sides.

The chapter reported a full modeling of this actuation principle, a preliminary approach for closed-loop control (tracking torque references at both joints), and simulation results of three representative locomotion tasks: level-ground walking, and stair ascending/descending. These tasks featured proper functioning of our prosthesis in different configurations: stair ascending (resp. descending) required to produce (resp. absorb) energy at both joints, while level-ground walking was in-between. Although the simulated power profiles revealed a significant filtering across the spring, energy balances also revealed that the overall device efficiency requires to be improved.

Acknowledgment

We are grateful to the authors of [78] who accepted to share a digital version of the data reported in their paper.

Part II

Continuously Variable Transmission Design

Continuously Variable Transmissions: Existing Technologies

6

Chapters 4 and 5 presented CVT based prosthesis morphologies. However, no CVT technologies are specified in these chapters.

The simplicity and stepless nature of CVTs make them an ideal transmission for a variety of machines and devices. CVTs have been used for years in power tools and drill presses. They have also been used in a variety of vehicles, including tractors, snowmobiles and motor scooters. But the main application of CVTs is cars. This chapter aims at giving an overview on existing CVT technologies and identify issues to embed it in a prosthesis.

Unlike traditional transmissions, continuously variable transmissions do not have a gearbox with a fixed number of gears, which means they do not have interlocking toothed wheels. The most common type of CVT operates on an ingenious pulley system that allows an infinite variability between highest and lowest gears with no discrete steps or shifts. However other technologies emerged over the last decades. This chapter will present the most common technologies of CVT.

6.1 Pulley-based CVTs

A traditional transmission is a complex combination of gears, brakes, clutches and governing devices. By comparison, a continuously variable transmission is simple. Pulley-based CVTs only have three basic components which are the key elements enabling the technology to work:

- A high-power metal or rubber belt

- A variable-input "driving" pulley
- An output "driven" pulley

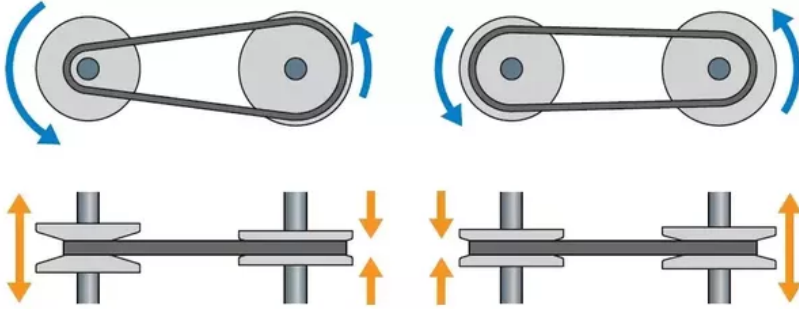


Fig. 6.1: Pulley-based CVT principle [44].

The variable-diameter pulleys are the heart of this CVT. Each pulley is made of two cones facing each other. A belt rides in the groove between the two cones. V-belts are preferred if the belt is made of rubber. V-belts get their name from the fact that the belts bear a V-shaped cross section, which increases its frictional grip.

When both cones of the pulley are far apart, the belt rides lower in the groove, and the radius of the belt loop going around the pulley gets smaller. When the cones are close together, the belt rides higher in the groove, and the radius of the belt loop going around the pulley gets larger (see Fig. 6.1). CVTs may use hydraulic pressure, centrifugal force or spring tension to create the force necessary to adjust the pulley halves.

Variable-diameter pulleys must always come in pairs. One of the pulleys is also called the input pulley because it is where the energy from the engine enters the transmission. The second pulley is called the driven pulley because the first pulley is actuating it. As an output pulley, the driven pulley transfers energy to the driveshaft.

When one pulley increases its radius, the other decreases its radius to keep the belt tight. As both pulleys change their radii relative to one another, they create an infinite number of gear ratios (i.e. from low to high and everything in between). For example, when the pitch radius is small on the driving pulley and large on the driven pulley, then the rotational speed of the driven pulley decreases, resulting in a lower gear ratio. When the pitch radius is large on the driving pulley and small on the driven pulley, then the rotational speed of the driven pulley increases, resulting in a higher gear ratio. Thus, in theory, a CVT has an infinite number of ratios that it can run through at any time, at any input speed. Fig. 6.2 shows high and low gear ratios.

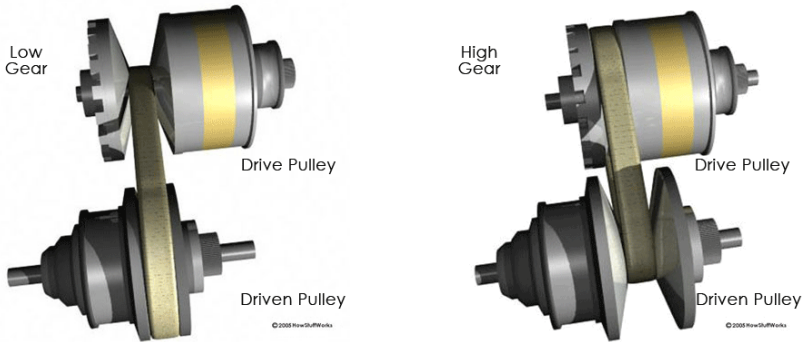


Fig. 6.2: Pulley-based CVT, with high and low gear ratio. The distance between the center of the pulleys to where the belt makes contact in the groove is called the pitch radius. When the pulleys are far apart, the belt rides lower and the pitch radius decreases. When the pulleys are close together, the belt rides higher and the pitch radius increases. The ratio of the pitch radius on the driving pulley to the pitch radius on the driven pulley determines the gear ratio [44].

Initially, these CVT transmissions relied on high-density rubber belts, which can slip and stretch, thereby reducing their efficiency. The introduction of new materials makes pulley-based CVTs even more reliable and efficient. One of the most important advances has been the design and development of metal belts to connect the pulleys. These flexible belts are composed of several thin bands of steel that hold together high-strength, bow-tie-shaped pieces of metal (see Fig. 6.3).

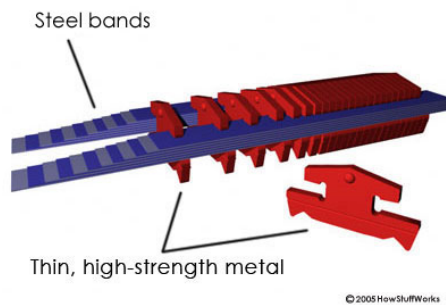


Fig. 6.3: Metal belt design [44].

Metal belts do not slip and are highly durable, enabling CVTs to handle more torque. Also, they produce less noise than rubber-belt-driven CVTs.

Fig. 6.4 shows an implementation designed by Jatco of a pulley-based CVT.

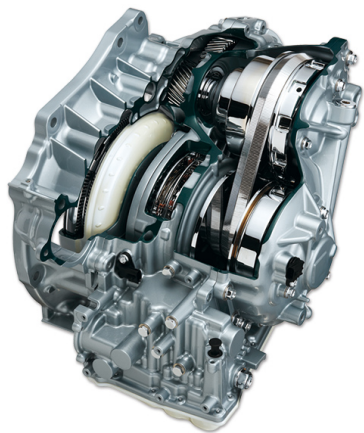


Fig. 6.4: Jatco CVT8 [63].

6.2 Toroidal CVTs

Another version of CVT is the toroidal CVT system. This mechanism replaces the belts and pulleys with discs and power rollers.

Although such a system seems drastically different, all of the components are analogous to a belt-and-pulley system and lead to the same results. Here is how it works:

- One disc connects to the input. This is equivalent to the driving pulley.
- Another disc connects to the output shaft. This is equivalent to the driven pulley.
- Rollers, or wheels, located between the discs act like the belt, transmitting power from one disc to the other.

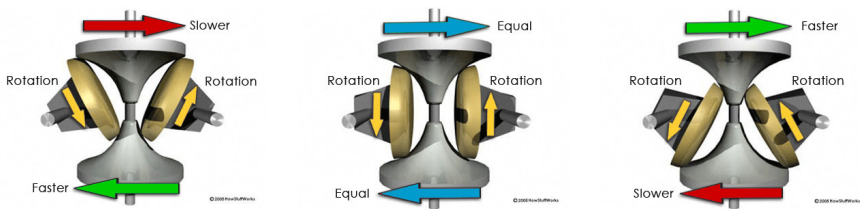


Fig. 6.5: Toroidal CVT principle [44].

The wheels can rotate along two axes. They spin around their axis of revolution and tilt in or out around the vertical axis, which allows the wheels to touch the discs

(see Fig. 6.5). When the wheels are in contact with the driving disc near the center, they must contact the driven disc near the rim, resulting in a reduction in speed and an increase in torque (i.e. low gear ratio). When the wheels touch the driving disc near the rim, they must contact the driven disc near the center, resulting in an increase in speed and a decrease in torque (i.e. high gear ratio). A simple tilt of the wheels incrementally changes the gear ratio providing smooth ratio changes.

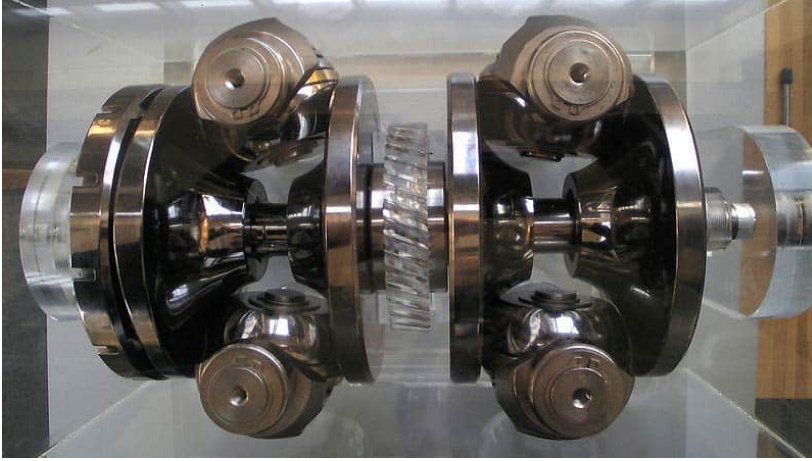


Fig. 6.6: Nissan's toroidal CVT [71].

Fig. 6.6 shows an implementation designed by Nissan of a toroidal CVT.

6.3 NuVinci

The NuVinci CVT has been developed by Fallbrook Technologies Inc. for bikes.

NuVinci technology is a CVT that is based on a set of rotating, tilting balls fitted between two rings. Fig. 6.7 illustrates the basic components of the NuVinci technology. The NuVinci technology has:

- An input disc, or ring, driven by the power source.
- An output disc, or ring, connected to the CVT output.
- A set of balls each rotating on its own axle and fitted between the input disc, output disc and a central sun to help maintaining the balls in position.

Torque from an engine, motor or other input source is transferred through the input disc to the balls using a thin layer of traction fluid.

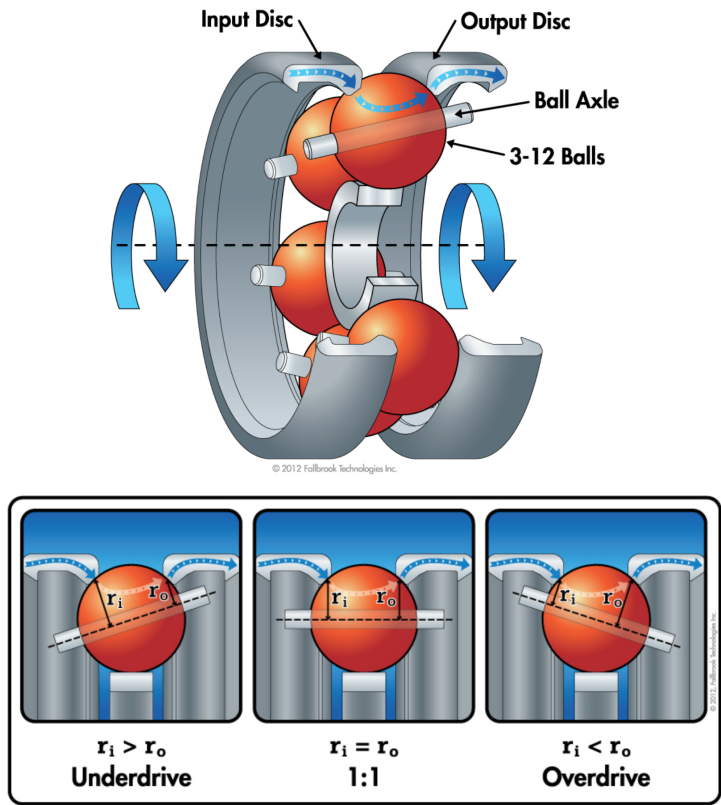


Fig. 6.7: NuVinci CVT principle [14].

The torque is then transmitted through the balls to the output disc via another thin layer of traction fluid. The input disc and output disc are clamped onto the balls tightly so that the required amount of clamping force is provided for the amount of torque being transmitted. However, as indicated above, torque is transmitted through the traction fluid, which prevents destructive metal to metal contact between the balls and discs while providing traction for the balls and rings and lubrication for bearings and other components.

The speed of the output disc compared to the speed of the input disc (gear ratio) is controlled by the angle of the ball axles relative to the axis of the transmission. Fig. 6.7 illustrates that tilting the ball axles shifts the transmission from low to high (as shown left to right), or from high to low, or to any ratio in between. The number of balls used depends on several factors including torque and speed requirements, operational requirements and space considerations, among others [14].

Fig. 6.8 shows the actual transmission which has been opened.



Fig. 6.8: NuVinci CVT [14].

6.4 Cones-Ring-Transmission (KRG)

The Cone-Ring-Transmission (KRG) is an automatic transmission for use in smaller vehicles developed by GIF [39].

Power transfer in the KRG is via a metal ring, which rotates between the two opposing cones. To vary the transmission ratio, the angle of the ring is adjusted causing it to move along the longitudinal axes of the cones.



Fig. 6.9: The Cone-Ring-Transmission (KRG) [39].

Unlike other current CVT's, the KRG is a purely mechanical system designed without the use of a hydraulic system for adjusting the contact pressure. This means the CVT has a comparatively low weight and a compact design.

6.5 Conclusion

The common point between all CVT technologies is the friction-based power transmission. Indeed, to obtain an infinite number of possible ratios, tooth cannot be used to transmit mechanical power.

The presented technologies can provide different ratios by moving the contact point radius between two elements. One of the drawback of these existing CVTs is that changing the ratio requires for the mechanism to overcome friction (which, on the other hand, is needed to transmit power) and make elements slips at their contact points. This is possible only if the transmission is moving fast enough. Consequently, it is impossible to change the ratio at rest or at low speed of the input shaft.

However an efficient use of CVT in walking applications requires changing the ratio with almost no energy cost, even at rest, i.e. when the input shaft is not rotating. We decided then to focus on developing a new CVT technology which can fulfil these requirements. Chapters 7 and 8 present the prototypes we developed during this thesis.

Continuously Variable Transmission Allowing Ratio Variations at Rest:

α -Prototype

7

Publication

The material presented in this chapter is adapted from:

Ch. Everarts, B. Dehez, and R. Ronsse. Novel infinitely Variable Transmission allowing efficient transmission ratio variations at rest. *IEEE International Conference on Intelligent Robots and Systems*, pages 5844-5849, 2015.

This chapter proposes a novel CVT and IVT principle which can achieve transmission ratio change at a minimal energy cost, even at rest, i.e. when the input shaft is not rotating. As explained in chapter 6, this ability is missing in existing CVTs.

The presented design is a modified planetary gear, whose planets are conical and mounted on inclined shafts, and ring is made of contiguous diablo-shaped rollers. This configuration allows moving the contact point radius on the cones and then modifying the transmission ratio. Importantly, this movement relies on rolling and not on sliding, such that it requires virtually no actuation energy. The IVT is an extension of the CVT combining the CVT with a traditional planetary gear in order to shift the transmission ratio range across zero.

The mechanical principle is presented and preliminary experiments were conducted on a first prototype.

7.1 Introduction

Legged locomotion involves both high and low torques in both directions within a single gait cycle, together with large variations in the joints velocity. This results in large fluctuations of the lower limb joint power profiles [74, 108]. Moreover, the power profile of all leg joints switches several times between positive and negative values as illustrated in Fig. 7.1 for the hip, knee and ankle of a 75 kg healthy walker during one step [108].

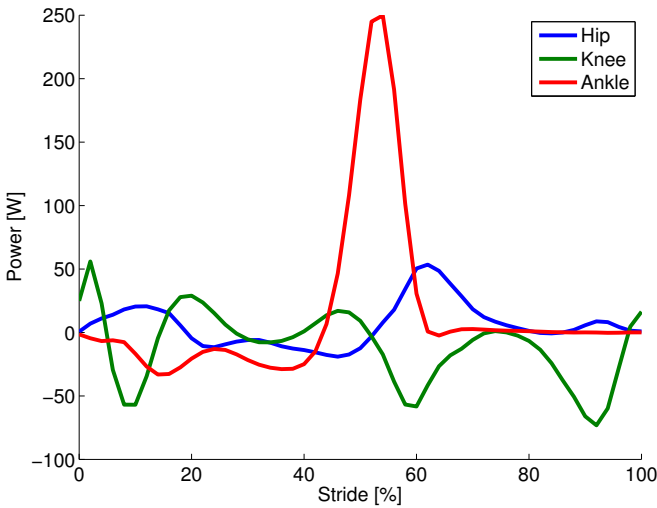


Fig. 7.1: Lower limb joints power. Data adapted from [108] for a 75 kg walker.

Humanoid robots aim to reproduce this dynamical behavior of human joints, and therefore must provide similar power profiles. This is even more critical in the development of lower-limb prostheses, whose ideal kinematics and dynamics should be close to the one of their human user. One architecture for actuating artificial leg joints leverages Series Elastic Actuators, e.g. the ankle-foot prosthesis developed by [2]. It includes a serial compliant element between the transmission output and the load. The compliant element has the ability to store and release energy and then provides a fraction of the fluctuating power. With this mechanism, the power profile provided by the motor is flattened with respect to the one transmitted to the load. However, the selected actuator stiffness typically optimizes this "power filtering" for only a single cadence [100].

Recent studies showed that the use of Continuously Variable Transmissions (CVT) or Infinitely Variable Transmissions (IVT) can drastically improve the motor efficiency, for a large range of walking cadences [28, 69, 86]. A CVT is a transmission whose transmission ratio can be continuously varied. An IVT is a CVT with a ratio

changing range crossing zero, i.e. from negative to positive values. In most cases, an IVT is implemented by adding a planetary gear at the output of a CVT [66, 67]. Indeed, a planetary gear can shift the ratio range of a CVT around zero.

In the literature, two different strategies of using CVTs or IVTs are proposed to improve efficiency of lower limb joints. The first is based on the use of a CVT between a motor and a series elastic element [69]. This concept allows the motor to operate at lower torque and at speed regimes of higher efficiency. The second is based on the use of an IVT between the elastic element and the load, in order to produce the desired output torque from an arbitrary level of energy in the elastic element, typically a spring [28, 86]. The idea is to operate the motor within a constant torque-velocity regime corresponding to its maximum efficiency. The motor feeds the spring with almost constant power and the desired torque is applied to the load by adjusting the IVT ratio. This concept is analogous to using a flywheel in applications requiring high power peaks with a constant input power. A critical difference is that the flywheel are heavy and stores kinetic energy, while the spring are lightweight and stores potential energy.

In order to bridge the gap between simulations and reality, these concepts require a CVT or IVT design being small, lightweight, and efficient at both power transfer and ratio control. Various implementations of CVTs and IVTs are proposed in the literature [30–32, 41, 51, 52, 64, 65, 91, 112]. This kind of CVT is remarkably efficient for automobile applications but they are complex, heavy, difficult to miniaturize and the transmission ratio cannot be efficiently changed at rest. CVTs and IVTs for smaller applications can be found, but they cannot operate continuous rotations, their range of motion being limited [30–32, 41, 52, 91, 112].

The present chapter proposes a new mechanism of CVT that was developed for robotic legged locomotion. This novel CVT can further be turned into an IVT by adding a planetary gear at the output. This chapter is organized as follows. In Section 7.2, the transmission principle is presented. In Section 7.3, an implementation of this concept, and a prototype of the IVT are described. In Section 7.4, preliminary experiments on the prototype are reported. The main results show that the output velocity can vary from negative to positive values while maintaining a constant input velocity. This experiment validates the model by showing good agreement between the expected and actual transmission ratios. Finally, section 7.5 concludes the chapter with a discussion on the experimental results, and opens perspectives for improvements.

7.2 Working Principle

This section develops the kinematic model of a new concept of IVT based on the combination of a common planetary gear (PG) and an original CVT whose principle

is directly inspired from a PG.

7.2.1 Planetary Gear Transmission

The kinematic diagram of a PG is given in Fig. 7.2 and the kinematic equation characterizing this well-known transmission is:

$$\frac{\omega_{pg,1} - \omega_{pg,4}}{\omega_{pg,3} - \omega_{pg,4}} = -\frac{R_{pg,3}R_{pg,2}}{R'_{pg,2}R_{pg,1}} \quad (7.1)$$

where $\omega_{pg,1}$, $\omega_{pg,3}$ and $\omega_{pg,4}$ are the angular velocities of the sun, the ring and the planet carrier, respectively, and $R_{pg,i}$ is the pitch radius of gear i .

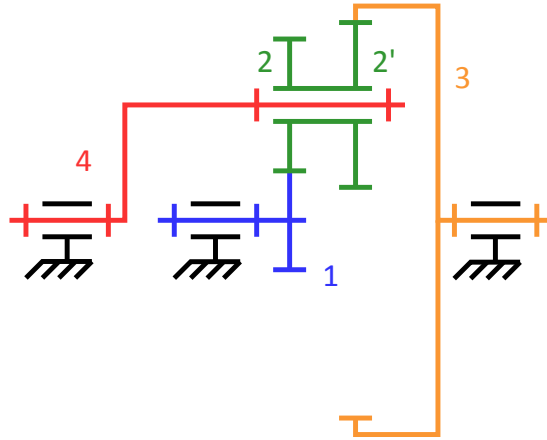


Fig. 7.2: Planetary Gear kinematic diagram. The different part of the planetary gear are called *sun* (1), *planets* (2, 2'), *ring* (3) and *planet carrier* (4).

7.2.2 Continuously Variable Transmission

The proposed CVT can be seen as a PG whose planets are inclined and cone-shaped to induce a ratio variation by moving the contact point on the cone surface. A kinematic diagram of this principle is illustrated in Fig. 7.3.

The key element of this new concept is the transmission between the ring and the planets. The planets on the ring side (6') are conical and their shafts are inclined so that the outer surface of the cones are tangent with a cylinder being concentric with the sun (5) and ring (7) axes. In this way, the ring can translate along its axis, while keeping contact with the cones, in order to vary the planet pitch radius $R'_{cvt,6}$ and therefore the transmission ratio.

Since the planet radius varies continuously, the motion cannot be transmitted with gears. Indeed, an integer number of teeth for each radius cannot be found.

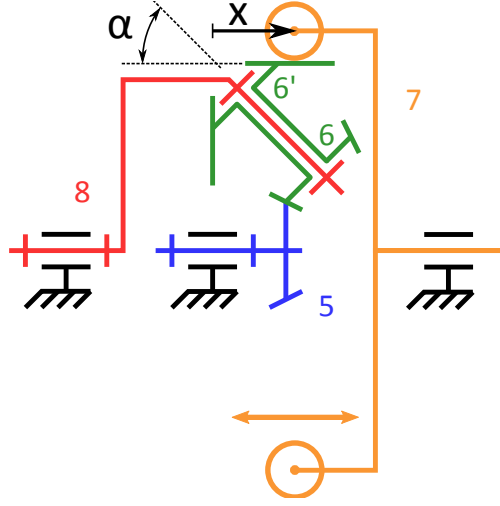


Fig. 7.3: Continuously Variable Transmission kinematic diagram. The sun is depicted in blue, the tilted planets in green, the carrier in red, and the ring in orange. This color code will be preserved throughout all the figures, including those presenting the prototype CAD

An alternative is thus to transmit the motion between the planets and the ring by friction.

It is moreover critical to minimize the energy required to change the ratio, and thus friction should be avoided for this movement. To achieve ratio changes by rolling instead of friction, we propose to build up the ring with several rollers whose axes are perpendicular to the ring axis and whose shape is such that they form a continuous circle of radius $R_{cvt,7}$ in contact with the planets, as illustrated in Fig. 7.4. In a way similar to the popular Swedish wheels being used in mobile robotics [13], the ring is thus blocked in the direction of motion transmission and free to rotate in the ring translation direction, i.e. the ratio changing direction.

Since the planets and sun axes are not parallel, the torque transmission between the planets (6) and sun (5) is achieved with bevel gears.

The kinematic equation linking the angular speed of the sun, the ring and the planet carrier is the same as for the PG:

$$\frac{\omega_{cvt,5} - \omega_{cvt,8}}{\omega_{cvt,7} - \omega_{cvt,8}} = -\frac{R_{cvt,7}R_{cvt,6}}{R'_{cvt,6}R_{cvt,5}} \quad (7.2)$$

but with $R'_{cvt,6}$ depending on the ring varying axial position x and the cone inclination angle α through :

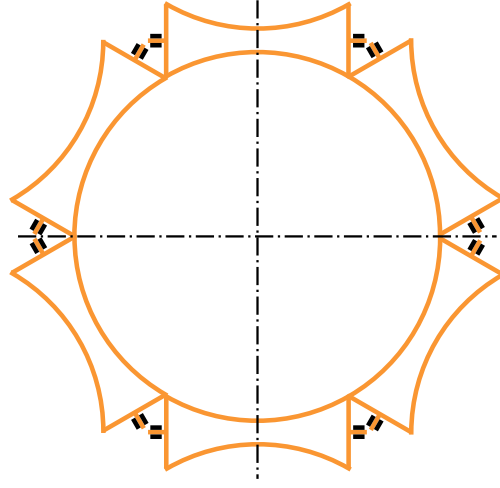


Fig. 7.4: Ring composed of several rollers whose axes are perpendicular to the ring axis

$$R'_{cvt,6} = x \sin \alpha \quad (7.3)$$

where $x = 0$ when the contact point is at the top of the cone.

7.2.3 Infinitely Variable Transmission

The CVT described in the previous section can be turned into an IVT by combining it with an inverted planetary gear, i.e. a PG whose ring is inside instead of being outside the planets, as illustrated in Fig. 7.5. The kinematic equation is exactly the same as for the classical PG, but opposite in sign:

$$\frac{\omega_{pg,1} - \omega_{pg,4}}{\omega_{pg,3} - \omega_{pg,4}} = \frac{R_{pg,3}R_{pg,2}}{R'_{pg,2}R_{pg,1}} \quad (7.4)$$

In order to obtain an IVT from the CVT and the inverted PG, the planet carrier of the CVT is connected to the ring of the PG (3) and the sun of the CVT to the sun of the PG, which is equivalent to imposing the following kinematic constraints:

$$\omega_{pg,1} = \omega_{cvt,5} \quad (7.5)$$

$$\omega_{pg,3} = \omega_{cvt,8} \quad (7.6)$$

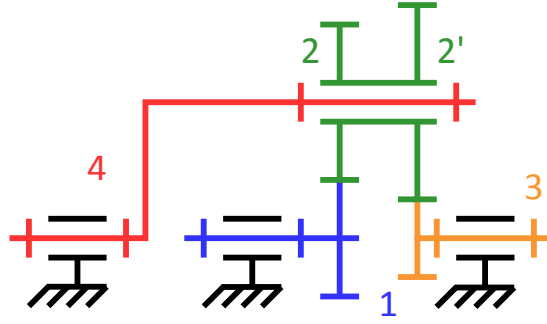


Fig. 7.5: Inverted Planetary Gear kinematic diagram

In addition, the ring of the CVT is blocked in rotation, which is equivalent to imposing the kinematic constraint:

$$\omega_{cvt,7} = 0 \quad (7.7)$$

According to this configuration, the input and the output of the IVT correspond to the planet carriers of the CVT and of the PG, respectively. The kinematic diagram of this IVT is given in Fig. 7.6.

From Eqs. 7.2 to 7.4, characterizing the CVT and the inverted PG, and Eqs. 7.5 to 7.7, corresponding to the kinematic constraints imposed to the CVT and the PG, the IVT variable transmission ratio τ_{ivt} can be derived as:

$$\tau_{ivt} = \frac{\omega_{pg,4}}{\omega_{cvt,8}} \quad (7.8)$$

$$= 1 + \frac{R_{cvt,7}R_{cvt,6}}{R_{cvt,5} \left(1 - \frac{R_{pg,3}R_{pg,2}}{R'_{pg,2}R_{pg,1}}\right)} \frac{1}{x \sin \alpha} \quad (7.9)$$

If $\frac{R_{pg,3}R_{pg,2}}{R'_{pg,2}R_{pg,1}} > 1$, the first term of Eq. 7.9 is negative. By increasing x from 0 to ∞ , the IVT ratio can therefore vary from $-\infty$ to 1, i.e. including 0.

7.3 Prototype

A prototype was designed to implement the principle developed in Section 7.2. This prototype has an outer diameter of 220 mm and is 150 mm long.

Fig. 7.7 shows a CAD drawing of the CVT ring, planets and sun. As previously mentioned, the rollers are diabolo-shaped so that their arrangement provides

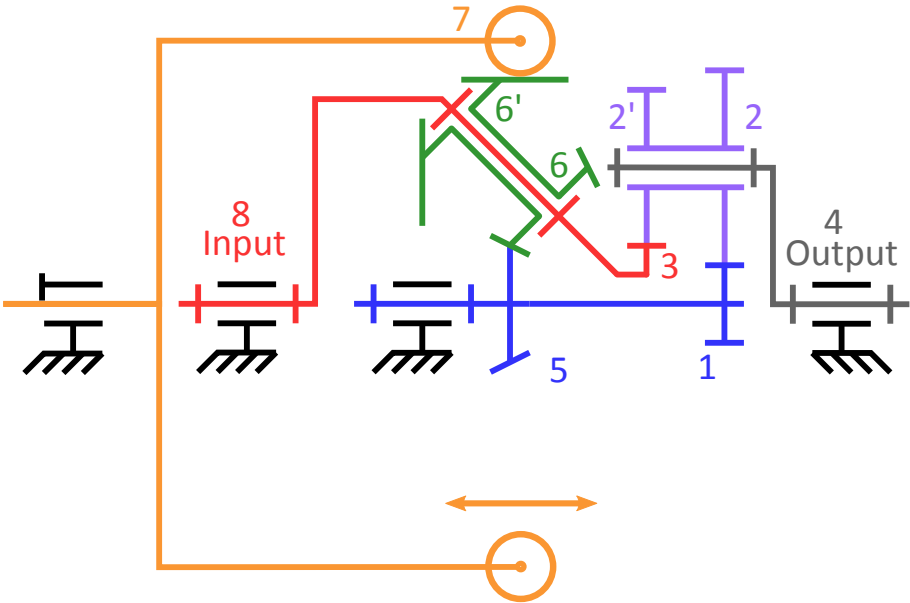


Fig. 7.6: IVT kinematic diagram. The suns are depicted in blue, the tilted conical planets in green, the CVT carrier and PG ring in red, the moving CVT ring in orange, the PG planets in purple and the PG carrier in dark grey. This color code will be preserved throughout all the figures, including those presenting the prototype CAD

a continuous circle in the inner side of the ring. As a consequence, the contact point between the ring and the conical planets moves on this circle without discontinuity, while keeping a constant distance with respect to the center of rotation. In the presented design, the CVT has 4 cones and 6 rollers. With more cones, the number of contact points would increase and so would the maximum torque capacity. However, this reduces the size of the cones and then the range of ratio variations. This amount of cones offers an ideal force distribution within the mechanism while exploiting most of the volume being available. With 6 rollers, the contact points between the cones and rollers are also ideally distributed: at most two contact points can be located at the connexion between two adjacent rollers, while in this case the two other contact points will be located in the middle of both other rollers. This configuration maintains the capacity to transmit high torques all over the circumference, and minimizes the vibrations induced by a transition of the contact points from one roller to the next. This last condition is respected when the number of cones and the number of rollers are not multiples of each other. Note finally that the number of rollers must be strictly larger than the number of cones in order to prevent that two cones contact a single roller with different radius. This would induce large friction forces during ratio variation.

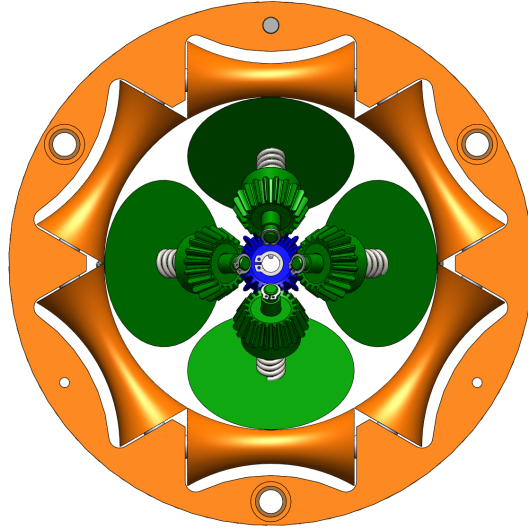


Fig. 7.7: IVT ring. The rollers are depicted in orange, the conical planets in green.

Fig. 7.8 shows CAD of the different components of the IVT and their assembly. The CVT planet carrier represented in Fig. 7.8(a) is a cage containing 8 bores being tilted by $\alpha = 45^\circ$ to host the 8 bearings of the cones shafts. The planet carrier also hosts the bores for the sun bearings.

The cones are mounted on the shafts and free to translate along their axes. Fig. 7.8(b) shows the assembly. In order to minimize the contact force variations and guarantee an equal distribution of the force between the cones and the rollers, the cones are spring loaded. The torque is transmitted through a key but the cone is not blocked along the axial degree of freedom as shown on Fig. 7.9, so that the springs can compress them on the rollers. The torque transmission between the planets and the sun is achieved by using 45° bevel gears.

The CVT ring is made up with the diabolo-shaped rollers mounted on their shafts through bearings. This allows changing the transmission ratio without recruiting friction forces, and thus with virtually no energy cost. The ring is displayed in Fig. 7.8(c). It is placed around the cones and is free to translate along the IVT longitudinal axis, thus modifying the instantaneous transmission ratio.

Fig. 7.8(d) shows the assembled CVT with an external frame and a motor to actuate the mechanism. The ring is mounted on 3 shafts through linear bearings. It is free to translate along the shaft direction but is blocked in rotation. Two gears are placed at the CVT output. One is connected to the planet carrier - i.e. the CVT input - and the second is connected to the sun - i.e. the CVT output.

Fig. 7.8(e) shows the planetary gear being connected to the two output gears of

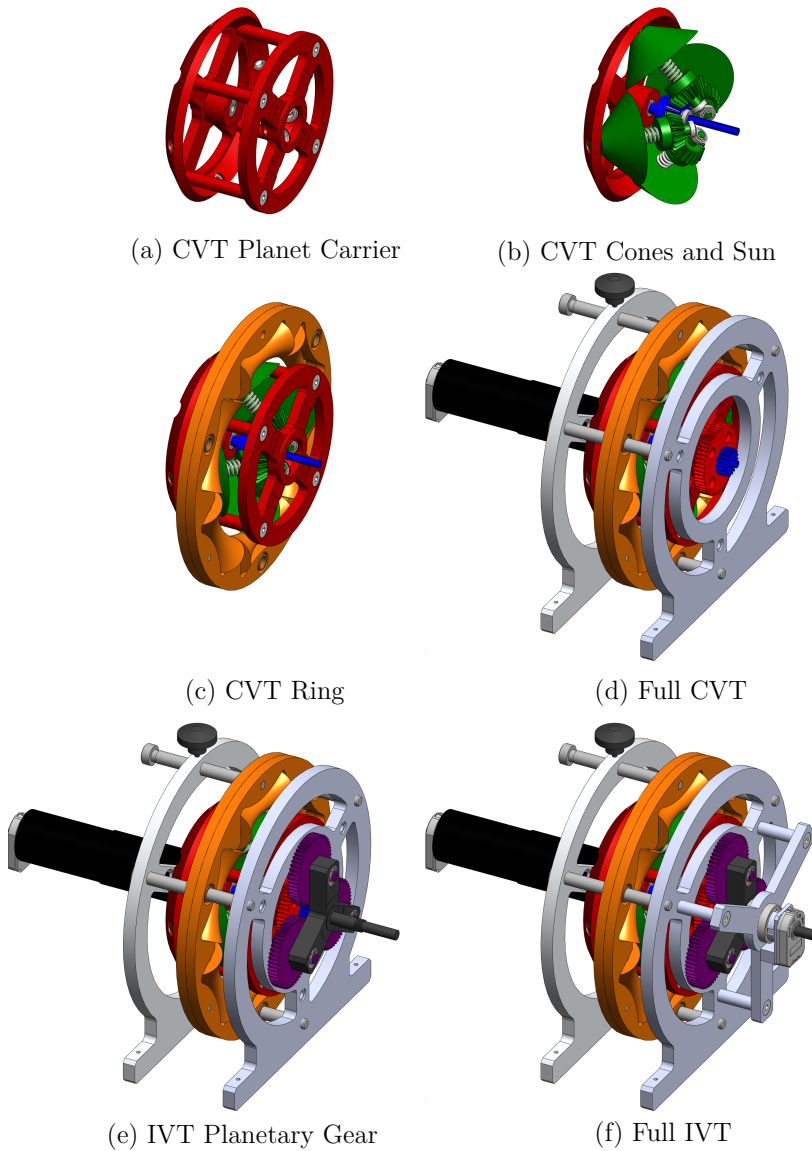


Fig. 7.8: Sequential assembly of the IVT CAD, highlighting the different components.

the CVT. The output of the IVT is made up with the planet carrier being placed around the two output CVT gears.

Fig. 7.8(f) shows the complete IVT and Fig. 7.9 shows a cut view of the IVT. This figure better emphasizes how all the components are connected and geared to each other. The figure highlights that the cones must be spring loaded and how planets are geared with the sun through tilted bevel gears.

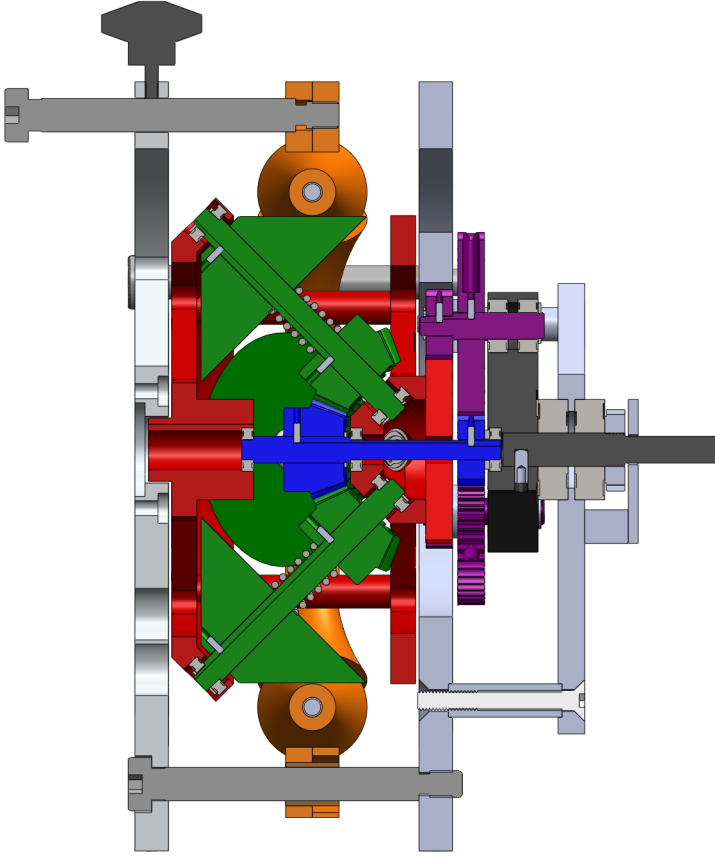


Fig. 7.9: IVT cut view

We further manufactured a prototype of this design. It is showed in Fig. 7.10.

The geometrical parameters of the actual prototype are the following:

- CVT ring radius: $R_7 = 70$ mm
- Cone angle: $\alpha = 45^\circ$
- Ring range: $x \in [21; 47]$ mm
- Bevel gears have a ratio equal to 1, i.e. $R_6 = R_5$
- Planetary gears radii: $R_1 = 10$ mm, $R_3 = 27$ mm, $R'_2 = 15$ mm and $R_2 = 22$ mm

Introducing these values in Eq. 7.9 above, the IVT ratio varies within the range $\tau_{ivt} \in [-0.58; 0.23]$, thus spanning across negative to positive values.

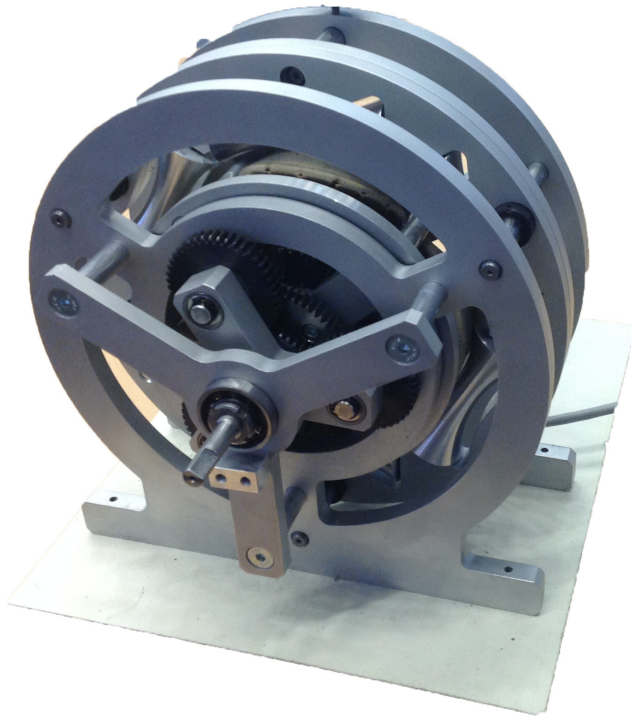


Fig. 7.10: Picture of the actual IVT prototype

7.4 Experimental Validations

A preliminary experiment with the prototype was carried out. This experiment aims at validating the IVT working principle, i.e. to show that the output velocity can indeed vary from negative to positive values while maintaining a constant input velocity. The CVT ring is moved on its whole range and the actual ratio is compared with the predicted one, provided by Eq. 7.9. A constant input velocity was provided with the motor at the input side and the output velocity was measured for different positions of the ring. In this experiment, the output was not connected to any load and the CVT ring was moved by hand.

Fig. 7.11 shows that the IVT ratio indeed changes when the ring is moved. The input velocity was controlled to be constant (around 11 rad/s) while the output velocity varied from -3.1 to 6.4 rad/s. Fig. 7.12 shows the measured ratio and the expected one. This experiment was video-taped and the recording is provided as supplemental multimedia attachment. The video further validates another very important feature of our design, namely that the transmission ratio can be changed when the input is at rest (zero velocity). The static force needed to move the ring has been measured with a force gauge and it never exceeds 1 N.

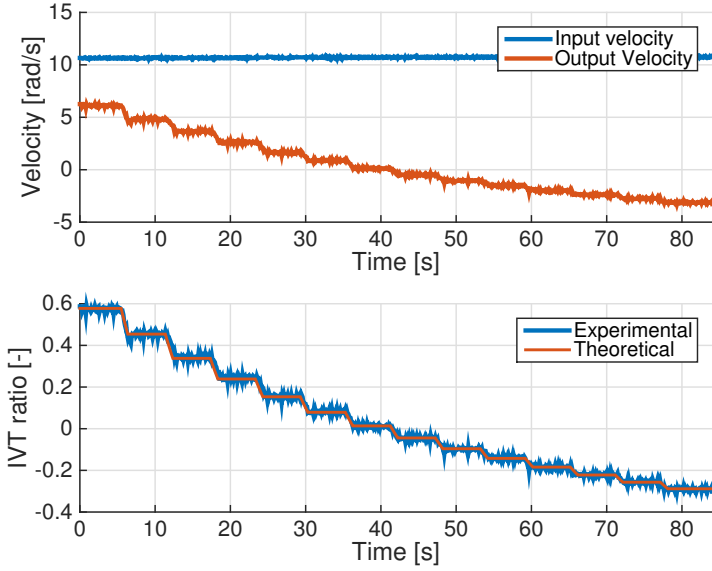


Fig. 7.11: Input and Output IVT velocities during manual changes of the ring position. The theoretical value is obtained from Eq. 7.9.

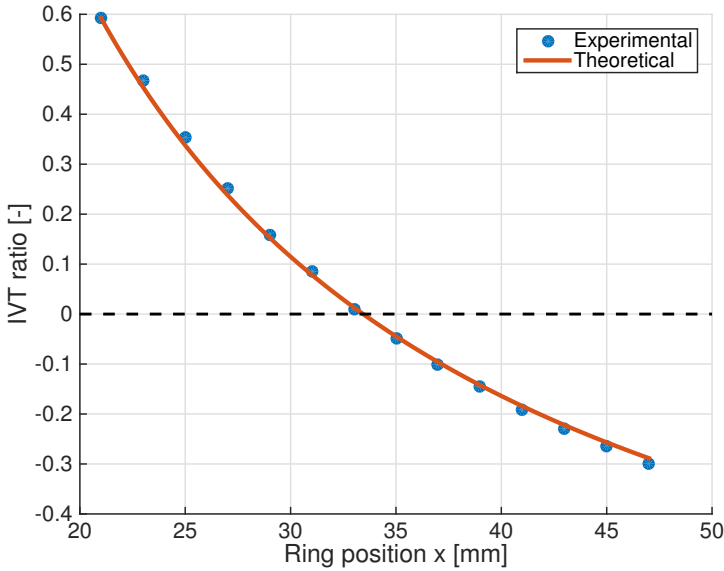


Fig. 7.12: IVT ratio variation as a function of the ring position. The theoretical value is obtained from Eq. 7.9.

7.5 Conclusion

This chapter presented a new kind of Infinitely Variable Transmission. The transmission ratio of this IVT can be changed even when the mechanism is at rest, theoretically with no frictional forces. This mechanism is thus an ideal candidate for applications in legged robotics (humanoids and prosthetics). These applications indeed require joints providing largely fluctuating output power, but which should be ideally fed by motors working close their optimal constant power.

The design of the IVT is the combination of a CVT and a planetary gear. The CVT is a modified planetary gear where the planets are turned into tilted cones. The ring is made up with rollers and can translate along the cones in order to change the contact point radius and then the transmission ratio.

A preliminary experiment was realized on a prototype. This experiment showed that the transmission output can indeed continuously vary from negative values to positive values with a constant input velocity. This experiment thus validated the IVT principle and showed that the measured ratio corresponded to the predictions.

Next step will focus on improving the prototype, mainly regarding its mechanical efficiency. Bevel gears, which are noisy and inefficient, should be replaced by friction gears to transmit the torque between the planets and sun of the CVT. Then, the spring loading would have a positive effect on both sides of the tilted planets. Secondly, we aim at increasing the torque transmission capacity between the cones and rollers.

Continuously Variable Transmission Allowing Ratio Variations at Rest: β -Prototype

8

Publication

The material presented in this chapter is adapted from:

Ch. Everarts, L. Mooney, F. Heremans, B. Dehez, H. Herr, and R. Ronsse.
A Continuously Variable Transmission Allowing Ratio Variations at Rest.
IEEE Transactions on Mechatronics, in preparation.

This chapter presents a second prototype of the Continuously Variable Transmission (CVT) presented in chapter 7.

Chapter 7 showed that the transmission ratio can indeed continuously vary from negative values to positive values. The CVT technology was thus validated. The second prototype aims at validating the possibility to embed this technology of CVT in a transfemoral prosthesis. In order to achieve that, we should validate to possibility to reduce the size, to obtain a good efficiency and a good torque capacity.

The chapter presents details about the CVT and IVT design and dimensioning, and then reports experimental results aiming at characterising the global efficiency of the transmission.

8.1 Introduction

Human locomotion involves both high and low hip, knee, and ankle torques in both directions of movement (flexion/extension) within a single gait cycle. Similarly, these three joints display large variations in their angular velocity. This in-

duces large fluctuations in the lower limb joint power profiles, at least in the sagittal plane [74, 108]. Moreover, the power profile of these leg joints switches several times between positive and negative values over one cycle, as illustrated in Fig. 8.1 [108].

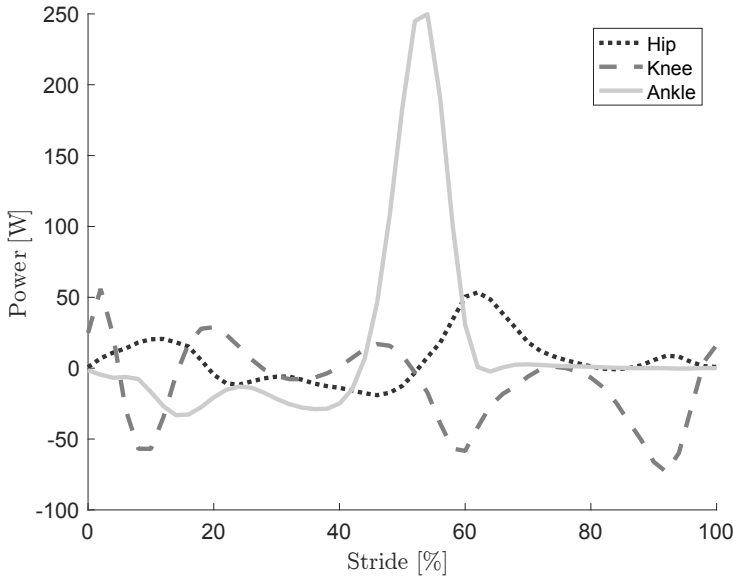


Fig. 8.1: Power profiles of the lower limb joints in the sagittal plane, as a function of the stride duration. 0 and 100% capture the time of two successive heel strikes. The transition between the stance and the swing phase occurs at approximately 60% of the stride. Data adapted from [108] for a 75 kg healthy walker.

Potentially, humanoid robots having similar legs should display dynamical behavior close to the human's one, and therefore similar power profiles. This is particularly relevant in bio-inspired locomotion projects which tend to emulate the mechanisms of human locomotion, in order to display human-like gaits [83, 95, 96]. Similarly, active lower-limb prostheses should potentially follow these ideal kinematic and dynamic patterns, in order to restore healthy walking to their human user. [2, 17, 35, 80]

An emerging method to actuate artificial leg joints relies on Series Elastic Actuators (SEA), both in humanoid legs [70] and in lower-limb prostheses [5, 8, 17]. Typically, a SEA embeds a serial compliant element between the transmission output and the load [77]. The compliant element has the ability to store and release mechanical energy and then displays a different power profile at the motor and load sides. Importantly, this holds pending a proper energy balance: the compliant element being passive, it cannot create energy. Consequently, for a cyclic profile, the power integral over the cycle will be same both on the motor and on the load side. If the stiffness of the compliant element is properly selected, the power profile provided by the actuator can be "flattened" with respect to the one transmitted to the load. For a

purely sinusoidal profile, this would be equivalent to tuning the stiffness to work at resonance, so that the motor would have only to continuously provide the average power. Consequently the selected actuator stiffness optimises this "power filtering" for a single cadence only [101].

Recent studies showed that the use of Continuously Variable Transmissions (CVT) or Infinitely Variable Transmissions (IVT) can generalise the "power filtering" described above, this time for a large range of walking cadences [28, 69, 86]. Interestingly, this induces a drastic potential improvement in the efficiency of the whole actuation chain. A CVT is a mechanical transmission whose ratio can be continuously varied. An IVT is a CVT with a ratio variation range crossing zero, i.e. taking both negative and positive values. Usually, an IVT is designed by adding a planetary gear at the output of a CVT (see e.g. [66, 67]). Indeed, the planetary gear can shift the ratio range of a CVT around zero.

In the literature, two different approaches of using CVT or IVT are presented in order to improve the actuation efficiency of artificial lower limbs. The first relies on an augmented SEA, with a CVT being placed between the motor and the series elastic element (see e.g. [69]). This concept makes the motor operating at lower torque and higher speed regimes, thus displaying a higher efficiency. Indeed, typical electrical motors display better efficiency regimes at high speed/low torque. The second also relies on augmented SEA, this times with an IVT placed between the elastic element and the load. This configuration allows to produce any desired output torque from an arbitrary level of energy in the the elastic element, pending the appropriate tuning of the transmission ratio. This concept was presented for the first time in [86]. Interestingly, for loads requiring a highly fluctuating power peak like in walking, it offers to actuate the motor with a constant regime corresponding to the *average* power of the load [28]. The motor feeds the spring with this nearly constant power, while the desired torque is applied to the load by adapting the IVT ratio. This concept is analogous to using a flywheel in applications requiring high power peaks with a constant input power. A critical difference is that the flywheel stores and releases kinetic energy, while the spring stores and releases potential energy. As direct consequence, this configuration offers to sharply reduce the size of the main actuator, and to boost its efficiency, since it operates in a almost constant power regime. The price to pay is the addition of a second (smaller) motor, in charge of actuating the IVT mechanism causing the ratio variations. In [79], we extended this concept to a biarticular actuator, namely for an above-knee active prosthesis.

In order to bridge the gap between simulations and reality, these concepts require a CVT or IVT design that is small, lightweight, efficient and - critically - with a ratio that can be changed even at rest, i.e. when the input shaft is not rotating. Indeed, walking requires to modify the output torque also when the joint is displaying zero (or very low) velocity. Various implementations of CVT and IVT exist

in the literature, see e.g. [30–32, 41, 51, 52, 64, 65, 91, 112]. The most common implementation relies on toroidal systems, like in [51, 64, 65]. This kind of CVT is remarkably efficient for automobile applications although they are complex, difficult to miniaturise, while their transmission ratio cannot be changed at rest. Indeed, changing the transmission ratio relies on sliding between two bodies and thus requires the device to be moving. CVT and IVT for smaller applications can be found, but they cannot operate continuous rotations, their range of motion being limited like in [30–32, 41, 52, 91, 112].

The present chapter reports the design of a new mechanism of CVT that was developed for robotic legged locomotion. It describes the kinematic model of this CVT which can further be turned into an IVT by adding a planetary gear at the output. The presented CVT concept is a modified planetary gear, whose planets are conical and mounted on inclined shafts, and ring is made of contiguous diabolo-shaped rollers. This configuration allows moving the contact point radius on the cones and then modifying the transmission ratio. Importantly, this movement relies on rolling and not on sliding, such that it requires virtually no energy, and can be performed even if the main shaft is at rest. This chapter is organized as follows. In Section 8.2, the transmission principle is presented. In Section 8.3, the main elements of the transmission are dimensioned. In Section 8.4, an implementation of this concept, and a prototype of the CVT is described. In Section 8.5, an experimental protocol validating the CVT prototype is reported, such as the main results showing that the output velocity can continuously vary while maintaining a constant input velocity. This experiment validates the model by showing good matching between the expected and actual transmission ratios. On top of that, efficiency of the mechanism is evaluated with different output speeds and load torques. Section 8.6 introduces the results of the same experiment for the IVT, i.e. the CVT combined with an planetary gear. Finally, Section 8.7 concludes the chapter with a discussion on the experimental results, and opens perspectives for improvements.

8.2 Working Principle

This section develops the kinematic model of the new CVT concept. This concept has already been reported in a conference proceedings [29], although the current version displays several improvements with respect the former one. All these improvements were conducted in order to obtain a smaller and more efficient mechanism.

8.2.1 Planetary Gear Transmission

Our new CVT concept can be viewed as a planetary gear with varying ratio between its ring and planets. A planetary gear (PG) is a system where one or more gears (planets) spin around a central gear (sun) and inside an outer gear (ring). The kinematic

diagram of a PG is provided in Fig. 8.2(a) and the kinematic equation characterizing this well-known transmission is:

$$\frac{\omega_{pg,1} - \omega_{pg,4}}{\omega_{pg,3} - \omega_{pg,4}} = -\frac{R_{pg,3}R_{pg,2}}{R_{pg,2'}R_{pg,1}} \quad (8.1)$$

where $\omega_{pg,1}$, $\omega_{pg,3}$ and $\omega_{pg,4}$ are the angular velocities of the sun, the ring and the planet carrier, respectively, and $R_{pg,i}$ is the pitch radius of gear i , following the nomenclature of Fig. 8.2(a).

8.2.2 Continuously Variable Transmission

The developed CVT concept is made of a PG whose planets are tilted and cone-shaped in order to induce a variation of the corresponding pitch ratio by moving the contact point between the ring and the cone surface. A kinematic diagram of this principle is provided in Fig. 8.2(b).

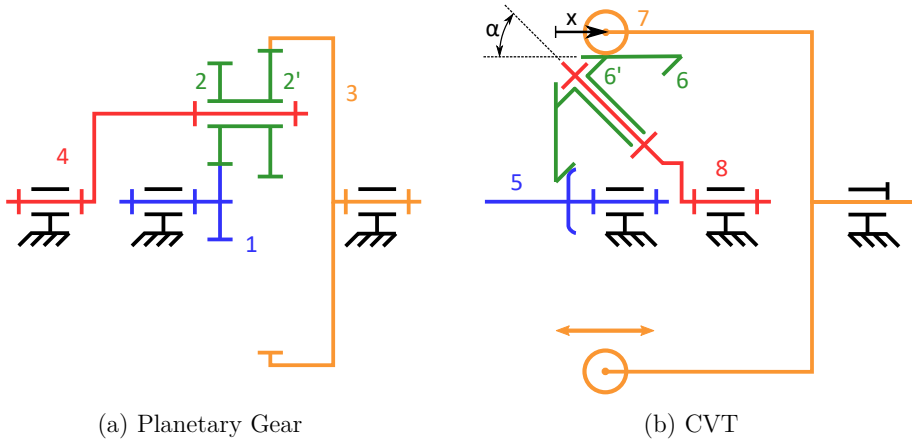


Fig. 8.2: (a) Planetary Gear kinematic diagram. The different parts of the planetary gear are called *sun* (blue, 1), *planets* (green, 2, 2'), *ring* (orange, 3) and *planet carrier* (red, 4). (b) Continuously Variable Transmission kinematic diagram. The sun (blue, 5) is dome-shaped, the tilted planets (green, 6, 6') are cone-shaped, the carrier (red, 8) contains the planets, and the ring (orange, 7) is composed of rollers. This nomenclature and color code will be preserved throughout all the figures, including those presenting the prototype CAD.

The key element of this new concept is the transmission between the ring and the planets. The planets (6') are conical and their shafts are inclined so that the outer surface of the cones are tangent to a cylinder being concentric with the sun (5) and ring (7) revolution axes. Consequently, the ring can translate along its axis, while keeping contact with the cones, in order to vary the planet pitch radius $R'_{cvt,6}$ and therefore the transmission ratio.

Since the planet radius continuously varies, the motion cannot be transmitted through gears. Indeed, an integer number of teeth for each radius cannot be found. An alternative is thus to transmit the motion between the planets and the ring by friction.

It is moreover critical to minimise the energy required to change the transmission ratio, and thus sliding movements should be avoided in this direction of motion (i.e. the one parallel to the revolution axis of the sun and ring). To achieve ratio variations by rolling instead of sliding, the ring is built up with several rollers whose axes are perpendicular to the ring axis and being diabolo-shaped, such that they form a continuous circle of radius $R_{cvt,7}$, in contact with the planets, as illustrated in Fig. 8.3. In analogy to the well-known Swedish wheels being used in mobile robotics [13], the ring is thus blocked in the direction of motion transmission and free to rotate in the ring translation direction, i.e. the ratio changing direction.

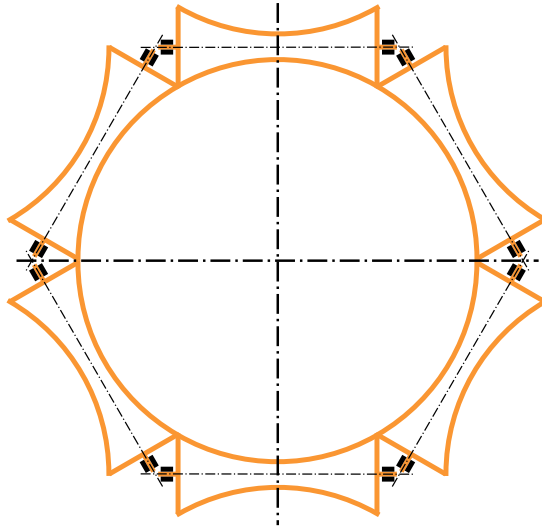


Fig. 8.3: Ring composed of several rollers whose axes are perpendicular to the ring axis.

Since the planets and sun revolution axes are not parallel to each other, the torque transmission between the planets (6) and sun (5) is achieved by friction. The sun is dome-shaped and is in contact with the base of the conical planets. The contact point distance from the sun axis and the planets axis are respectively $R_{cvt,5}$ and $R_{cvt,6}$.

The kinematic equation linking the angular speed of the sun, the ring and the planet carrier is the same as for the PG, i.e.:

$$\frac{\omega_{cvt,5} - \omega_{cvt,8}}{\omega_{cvt,7} - \omega_{cvt,8}} = - \frac{R_{cvt,7} R_{cvt,6}}{R_{cvt,6} R_{cvt,5}} \quad (8.2)$$

The key difference with the PG case is that $R_{cvt,6'}$ depends now on the ring varying axial position x and the cone inclination angle α (Fig. 8.2(b)) through :

$$R'_{cvt,6} = x \sin \alpha \quad (8.3)$$

where $x = 0$ when the contact point is at the tip of the cone.

If the ring of the CVT (7) is blocked in rotation, i.e. if the kinematic constraint $\omega_{cvt,7} = 0$ holds, and by considering the planet carrier (8) as input and the sun (5) as output, the transmission ratio τ_{cvt} can be written as following:

$$\tau_{cvt} = \frac{\omega_{cvt,5}}{\omega_{cvt,8}} = 1 + \frac{R_{cvt,7}R_{cvt,6}}{R_{cvt,6'}R_{cvt,5}} \quad (8.4)$$

$$= 1 + \frac{R_{cvt,7}R_{cvt,6}}{R_{cvt,5}} \frac{1}{x \sin \alpha} \quad (8.5)$$

Equation (8.5) shows that the CVT ratio is always positive with this mechanism. In order to convert the CVT into an IVT, a second stage must be added.

8.2.3 Infinitely Variable Transmission

The CVT described in the previous section can be turned into an IVT by adding a second stage, namely a classical PG. The kinematic diagram of this IVT is given in Fig. 8.4. The planet carrier of the CVT is connected to the planet carrier of the PG (3) and the sun of the CVT to the sun of the PG. This is equivalent to imposing the following kinematic constraints:

$$\omega_{pg,1} = \omega_{cvt,5} \quad (8.6)$$

$$\omega_{pg,4} = \omega_{cvt,8} \quad (8.7)$$

From Equations (8.1) and (8.5), characterizing the PG and the CVT, and Equations (8.6) and (8.7), providing the kinematic constraints, and with the planet carrier of the CVT as input and the ring of the PG as output, the IVT variable transmission ratio τ_{ivt} is derived as:

$$\tau_{ivt} = \frac{\omega_{pg,3}}{\omega_{cvt,8}} \quad (8.8)$$

$$= 1 - \frac{R_{cvt,7}R_{cvt,6}R_{pg,1}}{R_{cvt,5}R_{pg,3}} \frac{1}{x \sin \alpha} \quad (8.9)$$

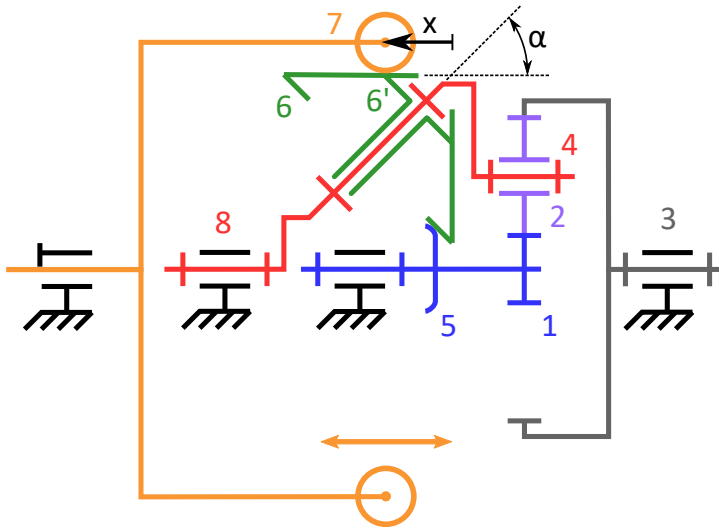


Fig. 8.4: IVT kinematic diagram. The color code is the same as of Fig. 8.2, except for the PG planets (2) in purple and the PG ring (3, output), which is displayed in dark grey.

Since there is a negative sign in front of the second term of Equation (8.9), varying the ring axial position x from 0 to ∞ will make the IVT ratio varying from $-\infty$ to 1, i.e. across 0.

8.3 Dimensioning

8.3.1 Main geometrical parameters

First of all, the amount of rollers must be strictly larger than the amount of cones, in order to prevent that two cones contact a single roller at two different contact points, thus corresponding to two different radii of the roller. This would violate the homokinetism of the corresponding roller and thus induce large friction forces during ratio variation. Next, not all contact points should be simultaneously located at the connexion between two adjacent rollers, so that the capacity to transmit high torques all over the circumference is maintained. This would further minimizes the vibrations induced by a transition of a contact point from one roller to the next. This last condition is fulfilled when the amount of roller and the amount of cones are not multiple of each other.

Here, we claim that an ideal embodiment for our CVT requires to use 4 cones and 6 rollers. With more cones, the number of contact points would increase and so would the maximum torque capacity. However, this reduces the size of the cones and then the range of ratio variation. In sum, 4 cones offers an ideal force distribution

within the mechanism while exploiting most of the volume being available. With 6 rollers, the contact points between the cones and rollers are also ideally distributed: at most two contact points are simultaneously located at the connexion between two adjacent rollers, while in this case both other contact points will be located in the middle of both other rollers.

Cone inclination has been chosen to be $\alpha = 45^\circ$. This value gives a good compromise between ratio range capacity and external diameter.

8.3.2 Forces relationships

In order to transmit torque through the mechanism by friction, a normal contact force should be applied between the different bodies. In order to produce such forces, the cones and sun are left free to translate along their axis. If a force F_s is applied along the axis of the sun, a reaction force F_{cs} is generated at the cone/sun contact point. Since the cones are also free to translate along their axis, a reaction force F_{rc} will be transmitted to the cones/rollers contact point.

This force F_s is generated through a ball ramp. It consists in connecting the sun to its shaft through balls rolling in a tilted groove. This ball ramp induces a relationship between the torque that is applied to the sun and the axial force being applied to it. Fig. 8.5 illustrates this principle. With this mechanism, the contact force between the different bodies is thus proportional to the torque that the whole mechanism is transmitting. This offers to decrease the overall friction losses in the mechanism during low torque transmission and thus to obtain a high efficiency on a large range of transmitted torques.

When a torque T_s is applied to the sun, a force is applied on the ball ramp. The force F_b that a ball can transmit is normal to the tilted groove containing it. This force F_b can be separated in a tangential component and an axial component. These two components can be linked together by the tilted angle of the groove β with the following relationship:

$$F_t = F_a \tan \beta \quad (8.10)$$

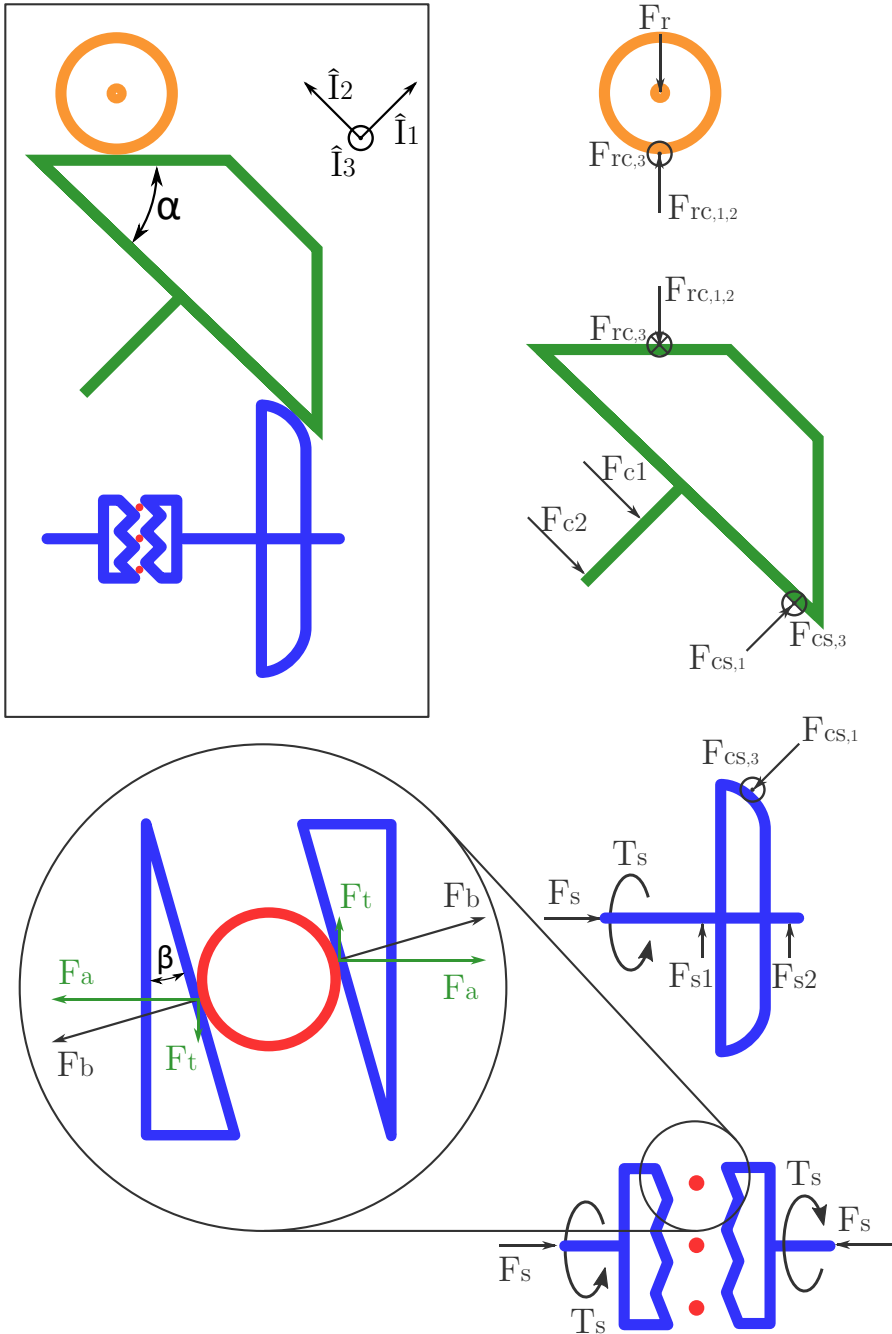


Fig. 8.5: Free Body Diagram of three parts of the CVT. F_{rc} is the contact force between the cone and roller, F_{cs} is the contact force between the cone and sun, and F_s is the axial force applied on to the sun. F_{c1} and F_{c2} are the cone bearings reaction forces. F_{s1} and F_{s2} are the sun bearings reaction forces.

If n balls are placed on a radius R_b , the total axial force F_s can be linked to the torque applied to the sun T_s by the following relationship:

$$F_s = \frac{T_s}{R_b \tan \beta} \quad (8.11)$$

The Free Body Diagram of a representative interface between one roller, one cone and the sun is depicted in Fig. 8.5. It will be used to compute the relationships between the different forces.

The force equilibrium along the $\hat{I}_1$ axis (parallel to the cone axis) provides the following relationship:

$$F_{cs,1} = F_{rc,1,2} \sin \alpha \quad (8.12)$$

$$= \frac{F_{rc,1,2}}{\sqrt{2}} \quad \text{if } \alpha = 45^\circ \quad (8.13)$$

The force equilibrium along the sun axis if the mechanism contains N cones gives the following relationship:

$$F_s = N F_{cs,1} \sin \alpha = N \frac{F_{cs,1}}{\sqrt{2}} = N \frac{F_{rc,1,2}}{2} \quad (8.14)$$

With 4 cones, Eq. (8.14) becomes:

$$F_s = 2F_{rc,1,2} \quad (8.15)$$

The torque equilibrium around the cone axis gives the following relationship:

$$F_{CS,3}R_{cvt,6} = F_{RC,3}R_{cvt,6'} \quad (8.16)$$

8.3.3 Torque Transmission

The maximum torque that the CVT can transmit depends on the friction coefficient between cones and rollers.

More preceptively, the maximum torque is found when one of these conditions is first encountered: (i) slippage between the cones and rollers, and (ii) slippage between

the cones and sun. Either one or the other will occur first, as a function of the cone radius at the contact point, i.e. the transmission ratio. Both cases are detailed below.

The torque transmitted to the planet carrier T_p is equal to the tangential force between cone and sun $F_{CS,3}$ multiplied by the sun radius $R_{cvt,5}$ plus the tangential force between cone and roller $F_{RC,3}$ multiplied by the ring radius $R_{cvt,7}$, this sum being multiplied by the number of cones. The torque transmitted to the sun T_s is equal to the tangential force between cone and sun $F_{CS,3}$ multiplied by the sun radius $R_{cvt,5}$. These relationships can be written as a function of the number of cones N_c and the Roller/Cone contact force or the Sun/Cone contact force.

Roller/Cone contact

$$T_s = F_{RC,3} N_c \frac{R_{cvt,5} R_{cvt,6'}}{R_{cvt,6}} \quad (8.17)$$

$$T_p = F_{RC,3} N_c \left(R_{cvt,7} + \frac{R_{cvt,6'}}{R_{cvt,6}} R_{cvt,5} \right) \quad (8.18)$$

Slippage occurs when the tangential force $F_{RC,3}$ becomes higher than the normal force $F_{RC,1,2}$ multiplied by the friction coefficient μ .

$$F_{RC,3} < \mu F_{RC,1,2} \quad (8.19)$$

By using Equations. (8.11), (8.14) and (8.17) in order to introduce the ball ramp angle β , this relation becomes:

$$T_s \frac{R_{cvt,5} R_{cvt,6'}}{N_c R_{cvt,6}} < \mu T_s \frac{2}{N_c R_b \tan \beta} \quad (8.20)$$

$$\tan \beta < 2 \mu \frac{R_{cvt,6} R_{cvt,5}}{R_{cvt,6'} R_b} \quad (8.21)$$

Sun/Cone contact

$$T_s = F_{CS,3} N_c R_{cvt,5} \quad (8.22)$$

$$T_p = F_{CS,3} N_c \left(R_{cvt,5} + \frac{R_{cvt,6}}{R_{cvt,6'}} R_{cvt,7} \right) \quad (8.23)$$

Slippage occurs when the tangential force $F_{CS,3}$ becomes higher than the normal force $F_{CS,1}$ multiplied by the friction coefficient μ .

$$F_{CS,3} < \mu F_{CS,1} \quad (8.24)$$

By using Equations. (8.11), (8.14) and (8.22) in order to introduce the ball ramp angle β , this relation becomes:

$$T_s \frac{1}{N_c R_{cvt,5}} < \mu T_s \frac{\sqrt{2}}{N_c R_b \tan \beta} \quad (8.25)$$

$$\tan \beta < \sqrt{2} \mu \frac{R_{cvt,5}}{R_b} \quad (8.26)$$

8.3.4 Stress dimensioning

Again, the critical points regarding stress dimensioning are the cone/roller and cone/sun contact points. When curved elastic bodies are pressed against each other, their ideal contact point degenerates as a finite contact area due to material deflection (see Fig. 8.6). These contact areas are so small that the corresponding compressive stresses tend to be extremely high. These stresses can be computed from the theory of Hertz, providing the dimension of the contact area and the maximum stress in the contacting bodies as a function of the material and curvature of the bodies at contact.

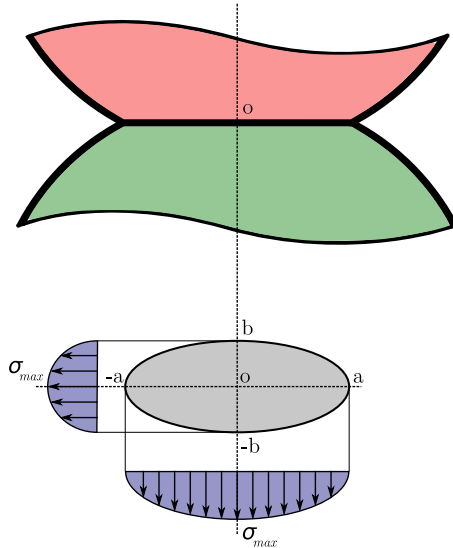


Fig. 8.6: Hertzian contact area. When curved elastic bodies (red and green) are pressed against each other, the ideal contact point degenerates as a finite contact area (grey), due to material deflection. This area has an ellipse shape with a and b the half length of its axes. Maximum stress occurs at the center of this ellipse.

The Hertzian contact parameters for a representative contact area of the CVT can be obtained by applying the following equations, taken from [57]:

$$\sigma_{max} = \frac{1.5 F_c}{\pi ab} \quad (8.27)$$

$$b = \beta \left[\frac{3 F_c}{4(B + A)} \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \right]^{1/3} \quad (8.28)$$

$$a = \frac{b}{k} \quad (8.29)$$

where, σ_{max} is the maximum stress at the center of the contact area; F_c is the normal force applied between both parts; a and b are the half-length of the long and short axes of the ellipse capturing the contact surface; ν_1 and ν_2 are the Poisson coefficients of both contacting materials; and E_1 and E_2 are their respective Young's modulus.

The parameters $(B + A)$ and $(B - A)$ are related to the main curvatures of the contacting bodies. They can be determined by the following equations:

$$B + A = 0.5 \left(\frac{1}{R_{12}} + \frac{1}{R_{11}} + \frac{1}{R_{22}} + \frac{1}{R_{21}} \right) \quad (8.30)$$

$$B - A = 0.5 \left[\left(\frac{1}{R_{12}} - \frac{1}{R_{11}} \right)^2 + \left(\frac{1}{R_{22}} - \frac{1}{R_{21}} \right)^2 + 2 \left(\frac{1}{R_{12}} - \frac{1}{R_{11}} \right) \left(\frac{1}{R_{22}} - \frac{1}{R_{21}} \right) \right]^{1/2} \quad (8.31)$$

where R_{ij} is the radius of curvature of body i in the plane j . In (8.28) and (8.29), the two coefficients β and k depend on the surface principal curvatures and can be approximated by linear relationships with respect to $\frac{B-A}{B+A}$, i.e.:

$$k = 0.95 - 0.89 \left[\frac{B - A}{B + A} \right] \quad (8.32)$$

$$\beta = 1.02 - 0.60 \left[\frac{B - A}{B + A} \right] \quad (8.33)$$

The contact stresses considerably exceed other stresses, like tension and bending. However, the parts integrity is usually not compromised even under such high stresses. This is due to the fact that the materials are under compression and not traction or shear. The maximum allowable contact stress can be determined by performing experiments that would characterize the deformation as a function of the contact geometry. This value can reach up to 5 times the yield strength for some geometries. Indeed, if small plastic deformation occurs, the contact areas rapidly increase and thus the stress decreases.

In our design, we did not perform these experiments and we decided to take σ_{max} equal to the yield strength of the material. This hypothesis is conservative since it should prevent any plastic deformation. With σ_{max} and the geometry being determined, F_c , the maximum admissible contact force between each pair of parts, can be found following Eq. 8.27.

8.4 Prototype

A prototype was designed in order to implement and validate the principle developed in Section 8.2. This prototype has an outer diameter of 110 mm and is 50 mm long. This size should roughly fit with the dimensions of a human-size artificial leg joint.

The geometrical parameters of the actual prototype are the following:

- CVT ring radius: $R_{cvt,7} = 30$ mm
- Cone angle: $\alpha = 45^\circ$
- Ring range: $x \in [8; 21]$ mm
- Sun radius: $R_{cvt,5} = 10.9$ mm
- Cone base radius: $R_{cvt,6} = 12$ mm

Introducing these values in Eq. (8.9), the CVT ratio could vary within the range $\tau_{cvt} \in [3.2; 6.8]$.

Fig. 8.7 shows a CAD drawing of the CVT ring, planets and sun. As previously mentioned, the rollers are diabolo-shaped so that their arrangement provides a continuous circle in the inner side of the ring. As a consequence, the contact point between the ring and a conical planet moves on this circle without discontinuity, while keeping a constant distance with respect to the center of rotation of the planet carrier.

Fig. 8.8 shows CAD of the different components of the CVT and their assembly. The CVT planet carrier represented in Fig. 8.8(a) contains 4 bores being tilted by $\alpha = 45^\circ$ to host bearings holding the conical planets shafts (2 in each bore). The planet carrier also hosts the bores of the sun bearings. Fig. 8.8(b) shows the sun mounted on the planet carrier.

Fig. 8.8(c) shows the assembly of the tilted cones in their bores. The CVT ring is made up with the diabolo-shaped rollers mounted on their shafts through bearings. The ring is displayed in Fig. 8.8(d). It is placed around the cones and is free to translate along the CVT longitudinal axis, thus modifying the instantaneous transmission ratio.

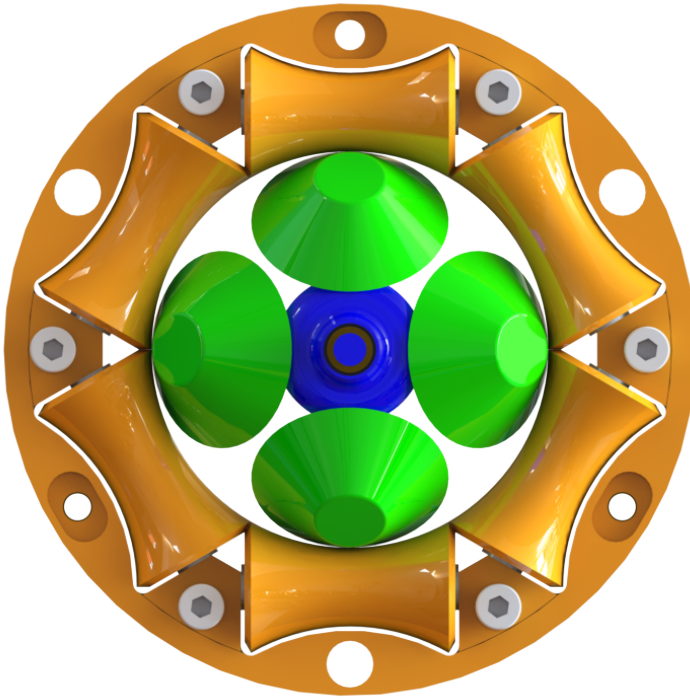


Fig. 8.7: IVT ring. The rollers are depicted in orange, the conical planets in green.

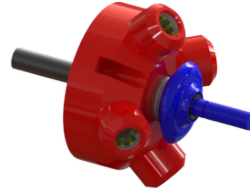
Fig. 8.9 shows a cross section of the CVT mechanism. Each bores in the planet carrier contains 2 needle bearings and a spacer. Cones are free to translate along their axes in these bearings so that the contact force is applied to the rollers. This contact force is applied trough the sun.

The sun is made of a dome connected to its shaft trough a ball ramp. This configuration is illustrated in Fig. 8.10. It consists in connecting the sun to its shaft through balls rolling in a tilted groove. This ball ramp induces a relationship between the torque that is applied to the sun and the axial force being applied to the dome. This axial force is transmitted to the sun/cones contact points in order to minimise slipping. Since the cones are free to translate along their axis, this force is also transmitted to the cones/rollers contact point. With this mechanism, the contact force between the different bodies is thus proportional to the torque that the whole mechanism is transmitting. This offers to decrease the overall friction losses in the mechanism during low torque transmission and thus to obtain a high efficiency on a large range of transmitted torques. This will be quantified in the validation experiments (see Section 8.5.2).

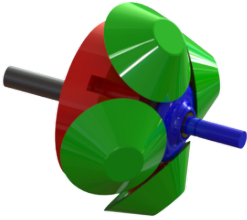
Fig. 8.8(e) shows the actuation mechanism of the CVT ratio. The ring is mounted on 3 shafts through linear bearings. It is free to translate along the shaft direction



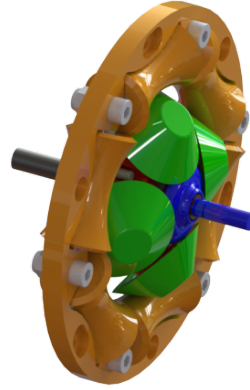
(a) CVT Planet Carrier



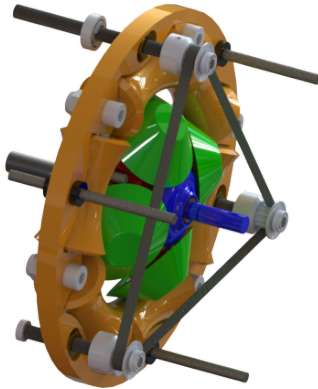
(b) CVT Sun



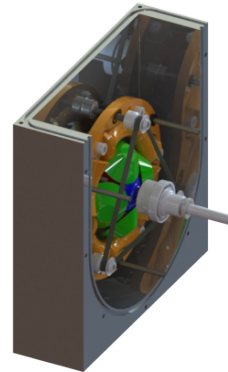
(c) CVT Cones



(d) CVT Ring



(e) CVT Ring Actuation



(f) CVT Frame

Fig. 8.8: Sequential assembly of the CVT, highlighting the different components. The color code is the same as in the previous figures.

but is blocked in rotation. In order to keep a good alignment of the ring with respect to the CVT cones, 3 synchronized leads screws actuate the ring. These lead screws are synchronized through a timing belt.

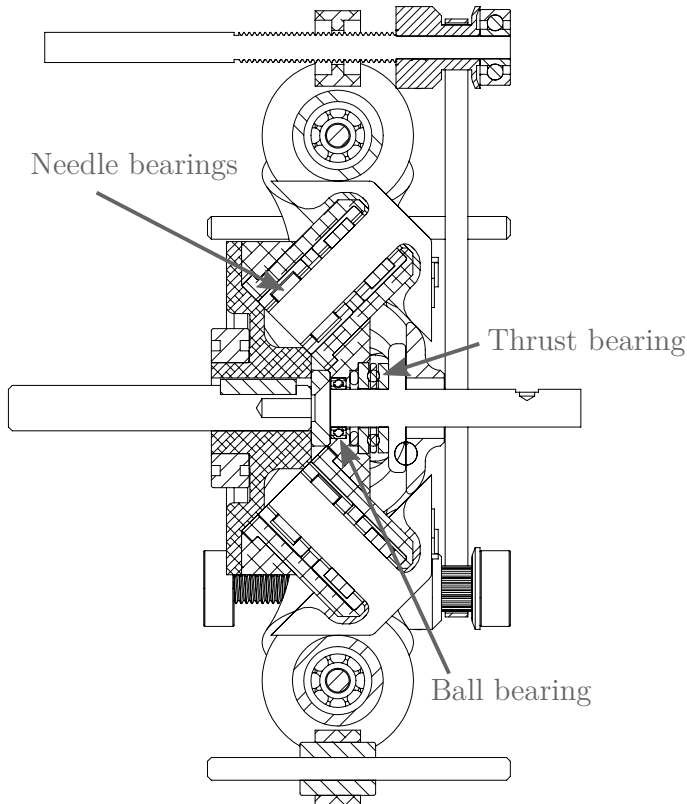


Fig. 8.9: Cross section of the CVT.

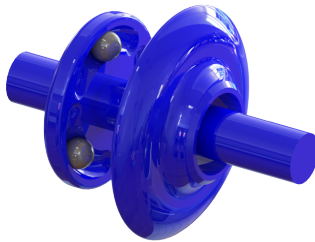


Fig. 8.10: The sun is mounted on a ball-ramp in order to adapt the contact forces as a function of the transmitted torque.

We further manufactured a prototype of this design. It is pictured in Fig. 8.11.

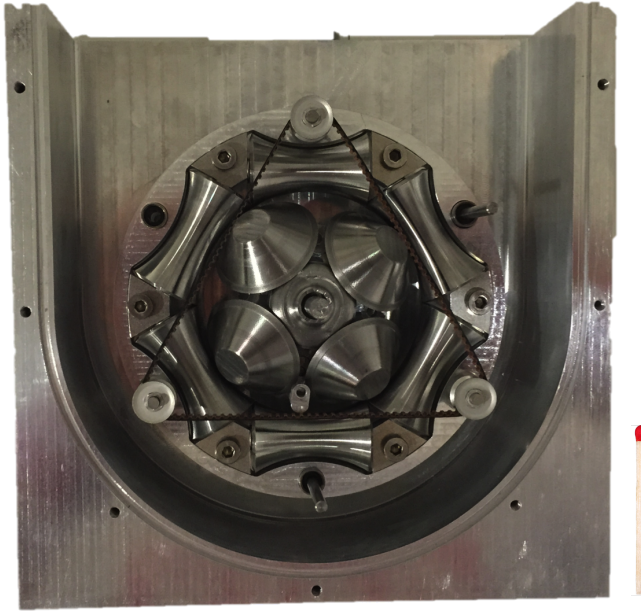


Fig. 8.11: Picture of the actual CVT prototype. Only half of the housing is shown here, in order to provide a glance to the internal mechanisms. A match is placed next to it to give an idea of dimension

Rollers, cones and sun were made with bearing steel *100Cr6*, i.e. a large yield strength ($\sigma_y = 2.4$ GPa), a Young's modulus $E = 210$ GPa and a Poisson ratio $\nu = 0.3$. These were the key parameters in the stress dimensioning since they determined the maximum admissible force between the contacting bodies. Moreover, the occurrence of seizure and abrasion was prevented by avoiding direct contact between the sun, cones and rollers. For this purpose, Traction Fluid flowed between the parts, thus lubricating and protecting the surfaces and transmitting power between them. Traction Fluids are known for having their viscosity multiplied by a factor up to 100 billion in high pressure regions (i.e. between 1 and 3 GPa). This prevents slippage and power losses. With Traction Fluid, the friction coefficient was estimated to be about $\mu = 0.1$.

The method described in Section 8.3.4 was applied to find the maximum contact force admissible between the rollers, cones and sun. This led to a maximum contact force between the cones and rollers equal to $F_{RC,1,2} = 300$ N and between the cones and the sun equal to $F_{CS,1} = 212$ N. These values were multiplied by μ and introduced in Equations (8.17), (8.18), (8.22), and (8.23) in order to obtain the maximum admissible torque on both the sun and planet carrier sides.

Fig. 8.12 shows the maximum admissible torque before facing slipping on the

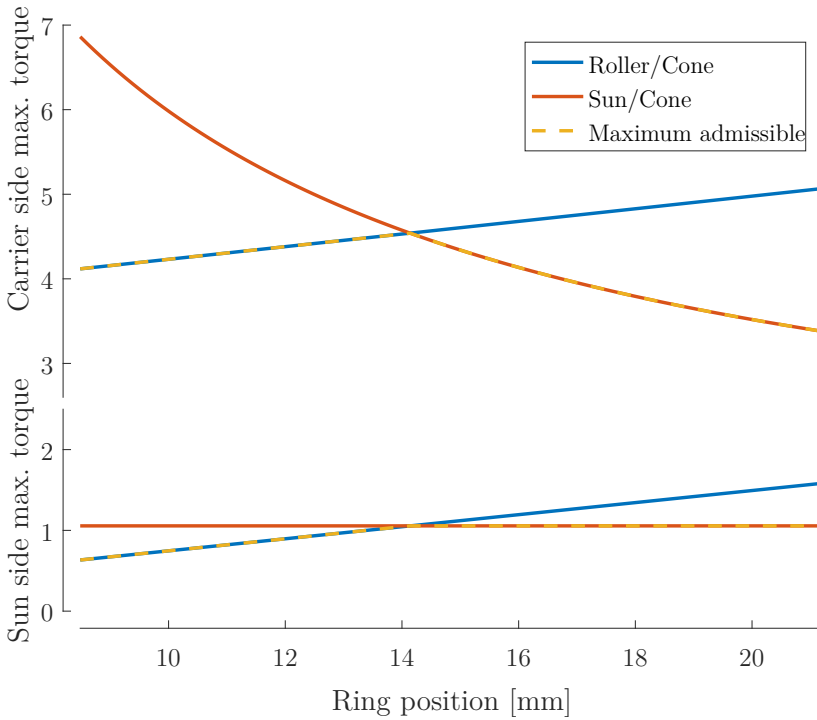


Fig. 8.12: Maximum admissible torques at both sides of the CVT, before reaching the slipping mode, and as a function of the ring position x , i.e. of the cone radius at the contact point $R_{cvt,6'}$ (see Eq. (8.3)). Blue curve shows maximal torque regarding the roller/cone contact areas. Red curve shows maximal torque regarding the sun/cone contact areas. Orange curve shows the minimum of both, i.e. the dimensioning value.

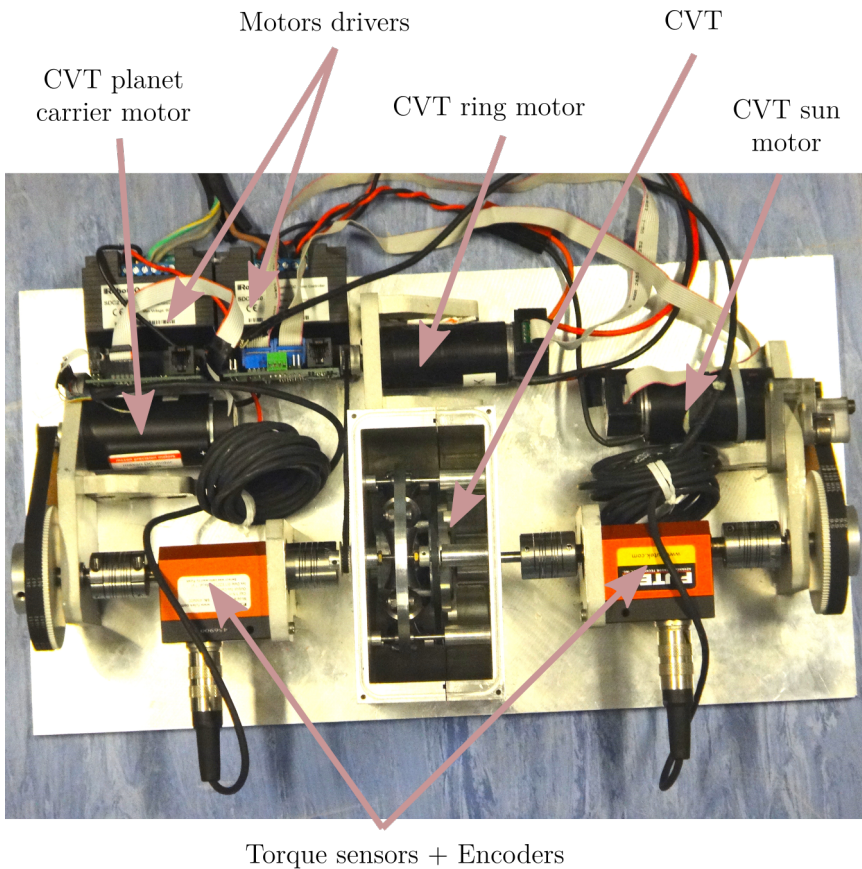
cone/sun or cone/roller contact surface. Overall, the maximum admissible torque is the minimum of both. Validation experiments will not test torques beyond this maximum.

8.5 Experimental Validation of the CVT

8.5.1 Protocol

The characterization of the prototype was performed with a test bench displayed in Fig. 8.13. A brushed DC motor (90 W, model: 148867, Maxon Motor, Sachseln, CH) was connected at each side of the CVT (sun and planet carrier) through a belt-pulley transmission with a ratio of 20/3. These motors can be used both to actuate the mechanism or to apply a resistive load. Torque sensors (model: TRS605 - 5 Nm, Irvine, CA) were connected at both side of the transmission in order to directly measure both torques without going through a transmission. These torque sensors have

also an embedded encoder, measuring the corresponding shaft velocity. The power flowing through both sides was computed by multiplying the torque and the angular velocity measured by this sensor. A brushed DC motor (90 W, model: 273753, Maxon Motor, Sachseln, CH) was used to control the transmission ratio by actuating the lead screws. All these components were mounted on an aluminium frame. Brushed DC Motor Controllers (model: SDC2160, RoboteQ, Scottsdale, AZ) were used to drive the three DC motors. An embedded hardware device (model: Beagle-Bone Black rev C, beagleboard.org) was used to collect data and control the motors. It was programmed in C++.



Planet carrier side

Sun side

Fig. 8.13: Test bench for the characterization of the CVT. Combined torque sensors + encoders are connected at both side of the transmission in order to measure torque and velocity. DC motors are connected at both side in order to actuate the CVT, or simulate its load. A third DC motor actuates the ring position in order to change the transmission ratio.

In order to characterise the mechanism, its efficiency was computed at different constant load torques. The overall efficiency being the ratio between the output and the input power was computed. Experiments were performed with both the sun and the carrier sides as input or output.

Each experiment was conducted as following: a constant torque was applied at the output while the input was controlled to follow a constant velocity. Each experimental trial lasted 20 seconds, during which the ratio was controlled to continuously change from its minimum to its maximum, and then back to its minimum again. Concretely, this was achieved by controlling the motor actuating the ratio variation with a constant positive velocity during 10 s, and then a constant negative velocity during the next 10 s. The ratio variation was assumed to be slow enough so that the CVT is in steady state and that no inertial effects impedes the efficiency. Data were recorded every 50 ms. Finally, a 6th order Butterworth low-pass filter with a cut-off frequency of 1 Hz was applied to the recorded data, in order to keep only their low-frequency variations, i.e. capturing the average efficiency of the device in these different configurations.

Experiment 1 consisted in a series of trials, conducted with the sun as input and the cone carrier as output. The input was controlled to maintain a rotational speed of 1000 rpm, 1500 rpm, 2000 rpm or 3000 rpm while the output applied a load of 0.55 Nm, 1.05 Nm or 1.55 Nm.

Experiment 2 consisted in a similar series of trials, again with the sun as input and the cone carrier as output. This time however, the output was controlled to maintain a rotational speed of 150 rpm, 300 rpm or 450 rpm while the input applied a load of 0.15 Nm, 0.25 Nm or 0.35 Nm. These values are all below the maximum admissible torque (see Fig. 8.12).

These experiments thus characterized the transmission with power flows going in both directions. Arbitrarily, Experiment 1 was said to be the positive power flow, and Experiment 2 the negative power flow.

8.5.2 Results

Fig. 8.14 shows the velocity and torque ratios of the CVT during one representative trial of Experiment 1. This figure validates the main aspect of the design, namely that the transmission ratio can indeed be varied as a function of the ring position. Velocity ratio (i.e. the input velocity ω_i divided by the output velocity ω_o) was higher than the torque ratio (i.e. the output torque T_o divided by the input torque T_i), likely because of slipping in the contact regions and friction losses occurring in the mechanism. Indeed, if slipping occurs, ω_o is lower than expected and the speed ratio is then higher. Friction losses which occur in the mechanism tend to decrease T_o and the torque ratio is then lower.

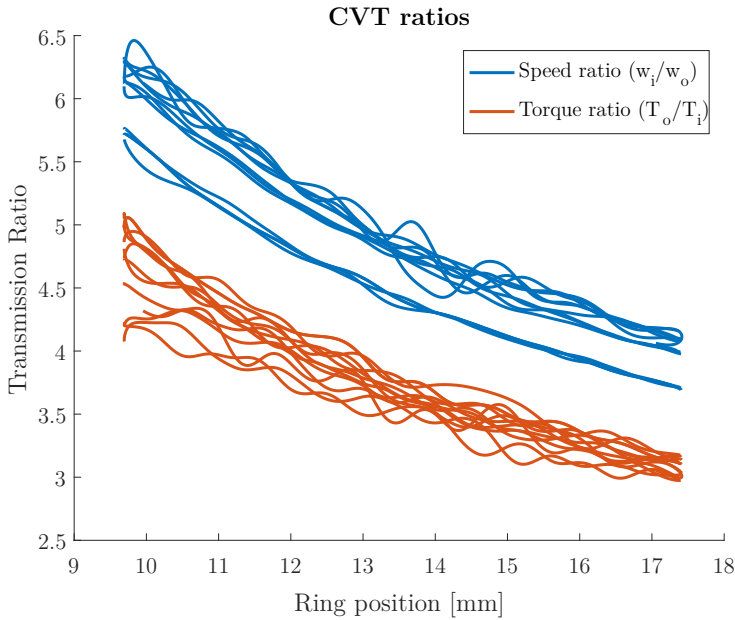


Fig. 8.14: The figure shows the ratios between the input and output velocity, the ratio between the output and input torque as a function of the ring position x for each experiments with a positive power flow.

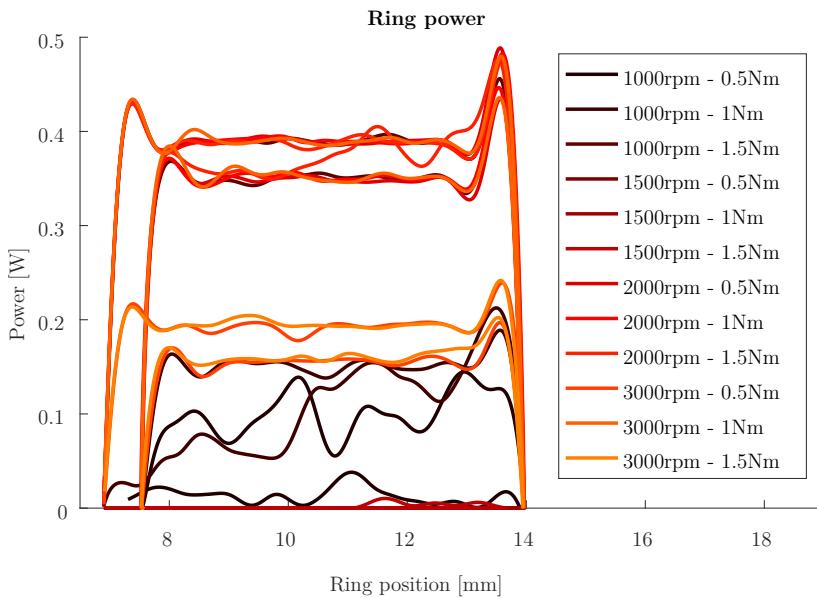


Fig. 8.15: Power consumption to move the ring back and forth during trials in the positive power flow.

Fig. 8.15 shows the power needed to move the ring. This graph shows that this power depends on the rotating speed of the sun. This power is quite low compared to the transmitted power. This low power is due to the rolling movement instead of slippage compared to existing CVTs.

However, a small difference is observed between both direction of ring movement. This difference can possibly be explained by some deflection of the cones. Indeed, when the ring reaches the base of the cone, cone bending is maximum. This is illustrated in Fig. 8.16. Therefore, with such bending, it requires less energy to move the ring further towards the base of the cone (negative slope) than towards its tip (positive slope). This possibly explains differences in power as a function of the direction of ratio variation.

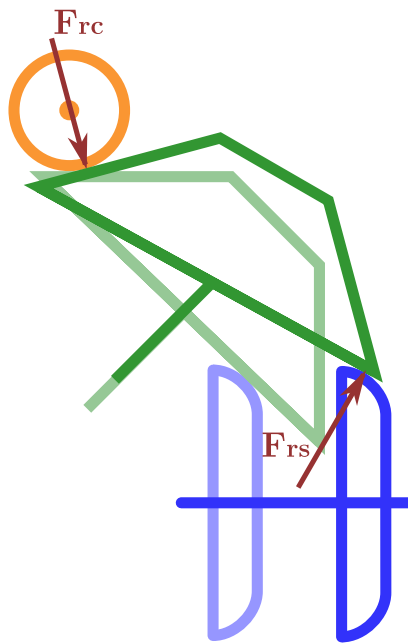


Fig. 8.16: Cone deflection when loaded. When the ring reaches the base of the cone, it likely deflects due to the force applied by the sun to the cone base F_{cs} and the force applied to the cone edge by the rollers F_{rc} . The deflection is magnified for visual purpose.

Fig. 8.17 shows the computed efficiencies, as a function of the ring position, for all the trials. This efficiency varies between 70% and 85% for the positive power flow and between 80% and 95% for the Negative power flow. Efficiency seems to increase when speed increases.

Fig. 8.18 shows the efficiency for all the different torques and velocities with power flow in both directions. Interestingly, the efficiency stays quite constant while

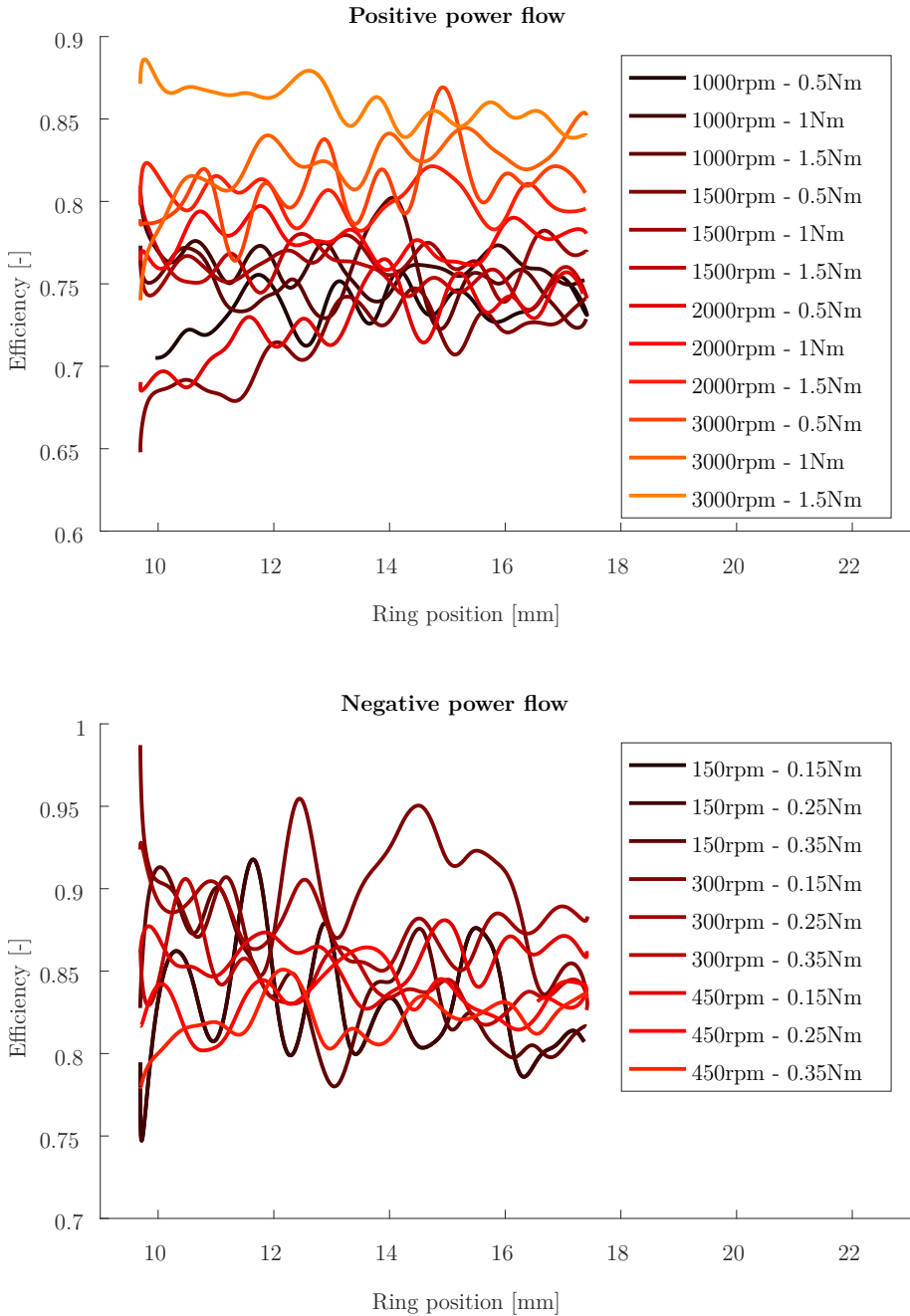


Fig. 8.17: Efficiencies of the CVT for all the trials in positive and negative power flows as a function of the ring position. It shows that the efficiency is higher with a negative power flow in the mechanism.

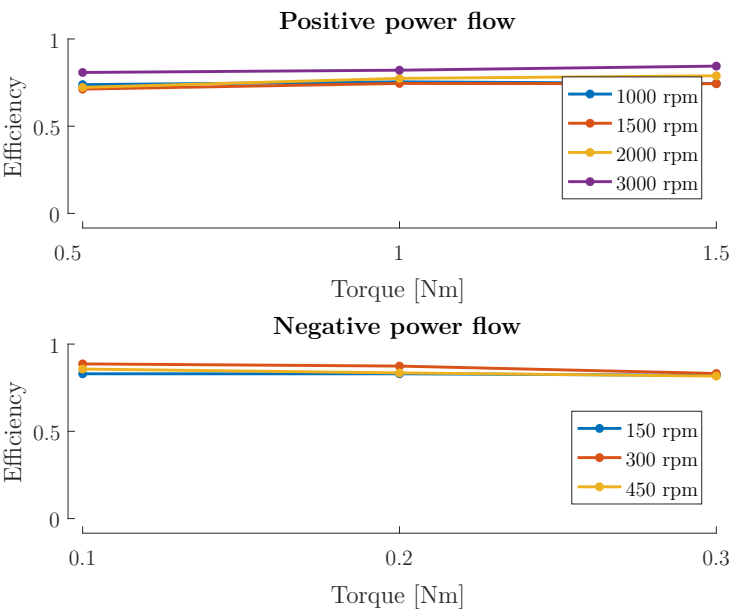


Fig. 8.18: Mean efficiency calculated during the 20 s trials for different torques and angular velocities. Left graph shows the result for a positive power flow and the right graph for a negative power flow.

the input velocity and the load torque vary across a large range. In contrast, with a conventional transmission, higher efficiency is typically reached when the maximum torque is applied. This validates the design of integrating a ball ramp modulating the contact forces as a function of the load torque. Indeed, decreasing the contact forces for a smaller load torque directly impacts the friction force in the contact surfaces and the bearings. These friction force are thus proportional to the torque being transmitted. Our mechanisms thus shows high efficiency even when the transmitted torque is lower.

Fig. 8.19 shows the losses power in the mechanism as a function of the total transmitted power. The slope of these graphs remains almost constant on the whole graph. It means that the efficiency is not influenced by the power passing through the mechanism. Again, this results validates the use of a ball ramp to minimise losses at power or torque.

Fig. 8.19 also shows that high power can be transmitted in the mechanism. During the first set of experiments, we reached 1.5 kW and during the second set, we reached 1 kW. This power is higher than the power level observed in human joints. This validates, in terms of power capacity, the possibility to use this mechanism in a prosthesis.

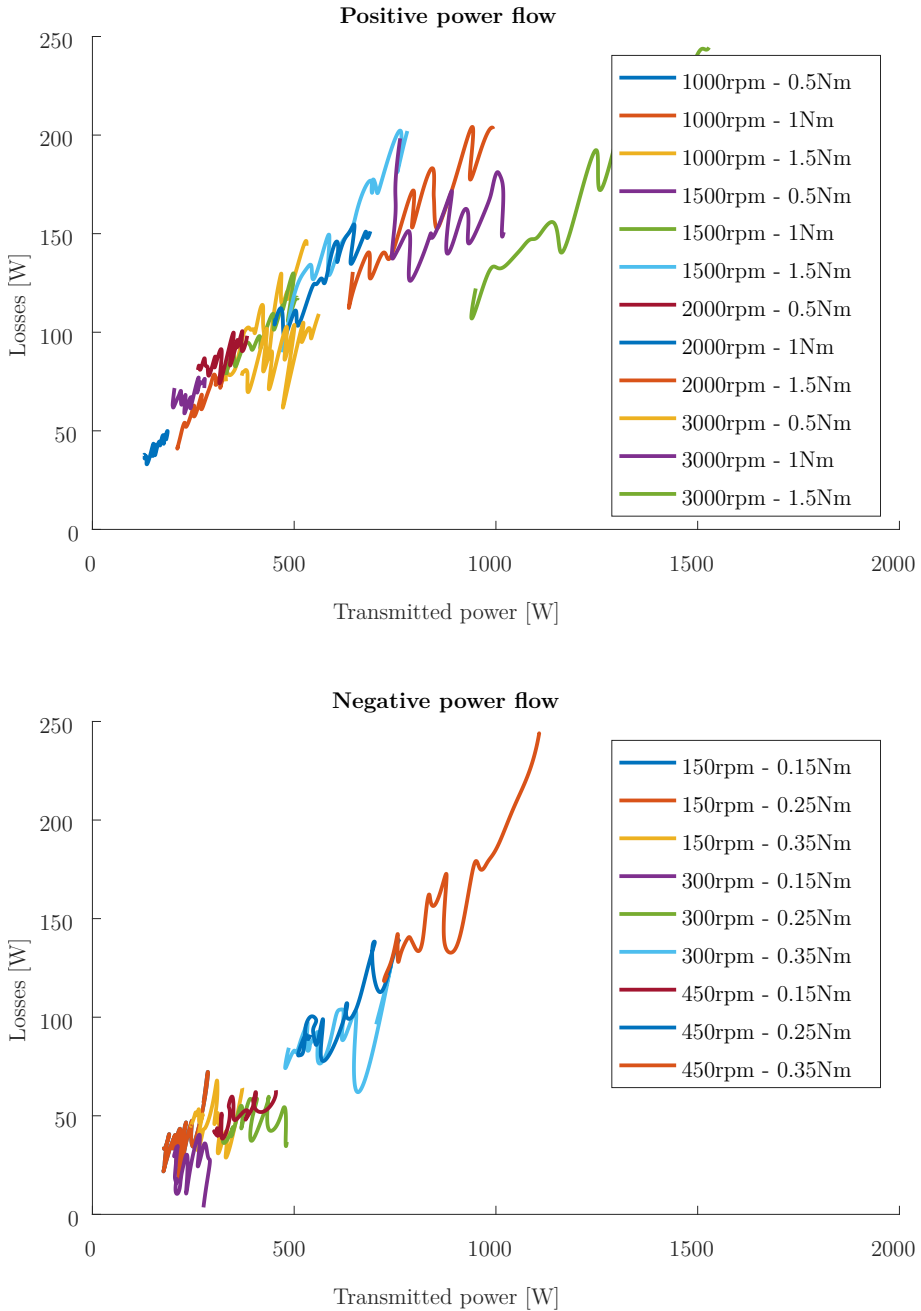


Fig. 8.19: Power losses in the mechanism as a function of the transmitted power. Slope of the curves seems quite constant meaning that transmitted power has low influence on the mechanism efficiency.

8.6 Infinitely Variable Transmission

Experiments were also conducted while turning the CVT into an IVT, by adding a planetary gear at the output side (see Section 8.2.3 for the details). The ring of this PG had a radius of $R_{pg,3} = 30$ mm and the sun a radius of $R_{pg,1} = 12$ mm. A movie is provided as supplementary material, illustrating the functioning of the IVT. In particular, it shows that the transmission ratio can also be changed when the input shaft is not rotating, a key feature of our design.

Furthermore, a third set of experiments was realized with the cone carrier as input and the second stage of the planetary gear as output. The input was controlled to be actuated with a speed of 500 rpm, 1000 rpm and 1500 rpm while the output applied a constant load torque of 0.5 Nm, 0.75 Nm or 1 Nm. When ratio switched sign, also did the load torque in order to provide a resistive torque during the whole trials.

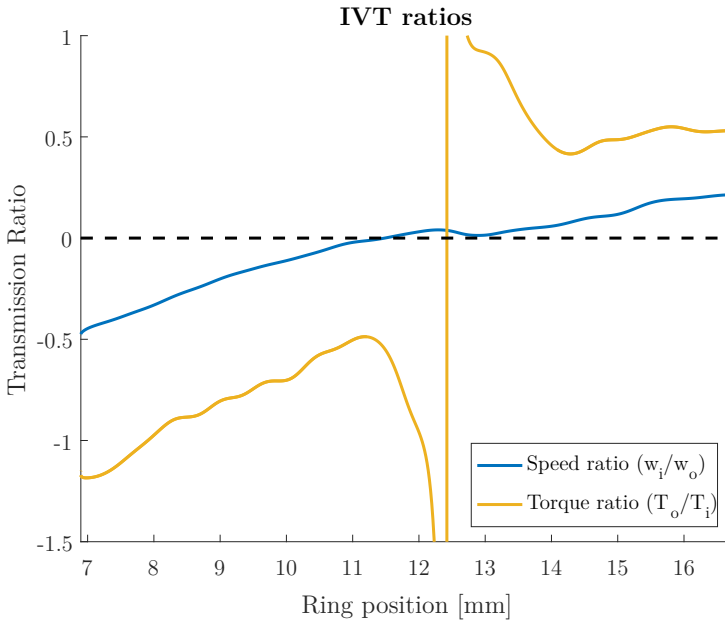


Fig. 8.20: The figure shows the ratios between the input and output velocity, the ratio between the output and input torque, and the expected ratio of the IVT during Experiment 3, as a function of the ring position x . Input was controlled to keep a constant speed of 500 rpm while a constant resistive torque of 0.5 Nm was applied at the output.

Fig. 8.20 shows the velocity and torque ratios of the IVT during one representative trial of Experiment 3. This figure validates that the transmission ratio can indeed be varied as a function of the ring position from negative to positive values. There is a range around zero where the transmission is not back-drivable anymore, so that

the measured input torque were irrelevant. The corresponding torque ratio being computed is thus not relevant either.

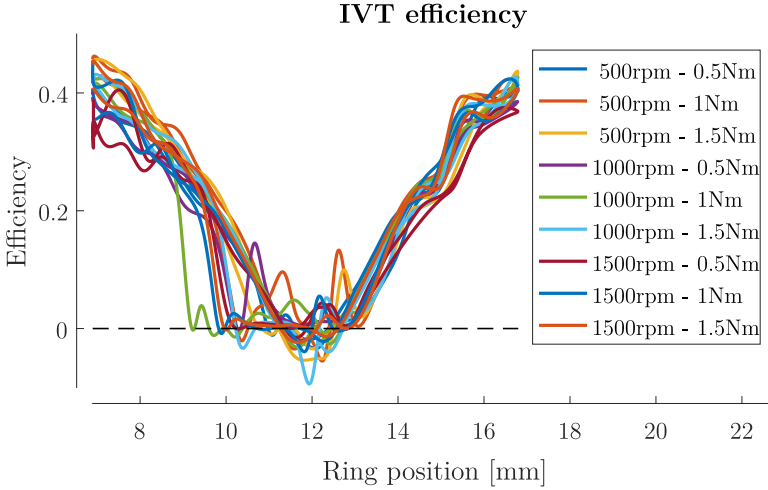


Fig. 8.21: Efficiency of the IVT throughout all experimental trials.

Fig. 8.21 shows the efficiency computed for each of the trials. It highlights that the efficiency was varying as a function of the ratio, following a V-shaped curve. More precisely, the efficiency decreased when the ratio went closer to zero. This was due to the fact that no power can be transmitted to the output if the transmission ratio is zero, while the input does dissipate power due to friction into the mechanism. Other contributions further characterised this dramatic drop of efficiency around a transmission ratio of 0 (see e.g. [66, 67]).

8.7 Conclusion

This chapter presented a new kind of Continuously Variable Transmission. The transmission ratio of this CVT can be changed even when the mechanism is at rest, with a very low energetic cost since it produces no frictional forces, in theory. This mechanism is thus an ideal candidate for applications in legged robotics (humanoids and prosthetics). These applications indeed require joints providing output power with large fluctuations, but should be ideally fed by motors working close their optimal constant power.

Our CVT is a modified planetary gear whose planets are turned into tilted cones. The ring is made up with rollers and can translate along the cones in order to change the contact point radius and then the transmission ratio. This CVT can handily be turned into an IVT by combining it with a planetary gear.

Experiments were conducted with a prototype. These experiments showed that

the CVT transmission ratio can indeed continuously vary by keeping an efficiency between 0.7 and 0.9.

Future steps will focus on improving the prototype, mainly regarding its mechanical efficiency. In order to increase the speed efficiency and avoid slipping, contact forces should be increased, for instance by changing the tilt angle of the grooves in the ball ramp. Future experiments will be conducted with different ball ramp in order to find the optimal angle in terms of efficiency. Moreover, increasing the contact forces might decrease the torque efficiency, so that a trade-off will indeed have to be found.

Secondly, we aim at increasing the torque transmission capacity between the cones and rollers. Now the sun, cones and roller are made of bearing steel. This steel can be hardened and grounded and a surface coating could be applied in order to increase the maximal normal force allowed and thus the maximum torque that the whole mechanism could transmit.

Last but not least, a larger prototype might be assembled in order to conduct experiments with a device for which fabrication tolerances will play a less significant role regarding efficiency.

General Conclusions

9

In this thesis we introduced a new concept of lower limb prostheses based on Continuously Variable Transmissions (CVT). This concept allows to manage the mechanical power flows through the device in both directions. Consequently the power motor does not face sharp load power fluctuations but smooth and almost constant power. However, to achieve that, a CVT allowing ratio variation at rest was needed. On top of that, it has to be small and lightweight enough to be embedded in a prosthesis. The second part of the thesis introduced a new kind of CVT principle fulfilling these requirements.

9.1 Summary

Chapter 3 showed that SEA and parallel springs can reduce energy consumption of the prosthesis by tuning springs stiffnesses. The outcome of the optimisation algorithm showed that these stiffnesses tend to follow the angle-torque characteristic of a sound limb. However, motors still have to produce power peaks. This is mainly due to the fact that with passive spring, energy transfers cannot be controlled. Indeed, these transfers depends on the joint kinematic and dynamic. A different approach to control energy flows was presented in Chapters 4 and 5. It relies on Continuously Variable Transmissions.

Chapter 4 investigated the use of Variable Stiffness Actuators for the design of an active ankle prosthesis, with two main objectives. First, we introduced the concept of Variable Stiffness Actuators (VSA) to circumvent the limitation of SEA. We explored potential benefits of changing the actuator stiffness not only across the cycles, but also within the cycle. This permitted to decrease the motor power peak by a factor 17, nearly corresponding to the minimum theoretical value, i.e. the total net energy produced by the joint during a walking cycle, divided by the cycle duration. This was achieved by means of a VSA rendering a continuously variable stiffness through a Continuously Variable Transmission. Second, we proposed two control

strategies for the motor feeding the spring energy. The first one was very simple, since it required the motor to be fed with a constant voltage. We showed that this strategy is stable over a wide range of input voltages, owing to an intrinsically stable torque-to-power relationship typical in DC motors. A major drawback of this approach is that it stabilized in a region where the motor efficiency is very poor, giving rise to an inefficient electromechanical conversion. This was improved by our second controller, which included a dynamical term stabilizing the spring elongation around a desired value. This permitted to improve the electromechanical conversion efficiency by about 25%. Our results tend to validate the use of a VSA based on a Continuously Variable Transmission for the design of an active ankle prosthesis.

Chapter 5 extended the concept of Chapter 4 to a tansfemoral prosthesis. It relied on two CVTs modulating the torque transmitted to both the ankle and knee joints, a mechanical differential conveying the energy flows in both directions (production/absorption), and a single compliant element (spring) filtering the power profile between the load and the motor sides. The chapter reported a full modeling of this actuation principle, a preliminary approach for closed-loop control (tracking torque references at both joints), and simulation results of three representative locomotion tasks: level-ground walking, and stair ascending/descending. These tasks featured proper functioning of our prosthesis in different configurations: stair ascending (resp. descending) required to produce (resp. absorb) energy at both joints, while level-ground walking was in-between. Although the simulated power profiles revealed a significant filtering across the spring, energy balances also revealed that the overall device efficiency requires to be improved.

Chapters 4 and 5 presented CVT based prosthesis morphologies. However, no CVT technology was specified in these chapters. Chapter 6 overviewed existing technologies of CVT but none of them displayed the required specs to make it embeddable in a prosthesis. An efficient use of CVT in walking applications requires changing the ratio with almost no energy cost, even at rest, i.e. when the input shaft is not rotating. We decided thus to focus on developing a new CVT technology which can fulfil these requirements. Chapters 7 and 8 covered the prototypes we developed during this thesis.

Chapter 7 presented a new technology of Continuously Variable Transmission. The transmission ratio of this CVT can be changed even when the mechanism is at rest, theoretically with no frictional forces. This mechanism is thus an ideal candidate for applications in legged robotics. The CVT is a modified planetary gear where the planets are turned into tilted cones. The ring is made up with rollers and can translate along the cones in order to change the contact point radius and then the transmission ratio. Preliminary experiment was realized on a prototype. This experiment showed that the transmission output can indeed continuously vary from negative values to positive values with a constant input velocity. This experiment

thus validated the CVT principle and showed that the measured ratio corresponded to the predictions.

Chapter 8 presented a second prototype of the CVT developed in Chapter 7. This second prototype aimed at validating the possibility to embed this technology of CVT in a transfemoral prosthesis. In order to achieve that, it validated the decrease of its size, its good efficiency, and its relevant torque capacity. The chapter presented details about the CVT and IVT design and dimensioning, and then reported experimental results aiming at characterising the global efficiency of the transmission. Experiments were conducted and showed that the CVT transmission ratio can indeed continuously vary by keeping an efficiency between 70 % and 90 %. This proved the possibility to embed the design in a transfemoral prosthesis.

9.2 Research questions

In the introduction, we detailed four main questions to be addressed in the context of this PhD thesis. These questions all targeted the analysis and improvement of energy efficiency of lower limb prostheses. By improving efficiency, smaller motors and smaller batteries can be used, eventually reducing the cost and the weight of the prosthesis.

The main questions that we wanted to answer in this thesis were the following:

- *Is it possible to design active lower limb prostheses with motor providing power flow in only one direction during walking?*

In Chapters 3, 4, and 5, we showed that power can keep the same sign in a lower limb prosthesis if we introduce clever mechanisms. Chapter 3 showed that springs can absorb a part of the power fluctuations and keep the power flow in the motors with the same sign during almost the entire cycle. Chapters 4 and 5 introduced a concept of prosthesis based on CVT where the power flow of the motor is almost stationary and corresponds to the net work needed at the joint. This flow keeps then always the same sign during the gait cycle.

- *Is it possible to decrease power and torque peaks in lower limb prosthesis motors?*

In Chapters 3, 4, and 5, we showed that power and torque peaks can be drastically reduced in the power motor of lower limb prostheses compared to direct drive. By introducing springs in parallel and in series with the power motor, power and torque peaks decrease. However, motors still have to produce power peaks. This is mainly due to the fact that with passive spring, energy transfers cannot be controlled. Chapters 4 and 5 introduced the concept of CVT based prosthesis. The power motor feeds a spring with an almost constant power while a CVT acts like a valve to release the needed energy to the

joints. With this approach, motor power is almost constant and fluctuations are supplied by the spring.

- *Is it possible to transfer energy between joints in a lower limb prosthesis?*

Chapter 3 showed that with uncontrolled power flows, energy transfer is not desirable. However, some existing prostheses (e.g. CYBERLEGS Beta-Prosthesis) transfer energy from the knee to the ankle through a controlled mechanism. In this thesis, we presented a new concept to transfer the energy from the knee to the ankle, where a single spring is shared for both joints. Chapter 5 explained this principle: energy is stored in a spring and each joint is connected to this spring through a CVT. The CVT acts like a energy valve letting energy going in and out. Since both joints are connected to the same spring, all their energy storage is shared. The ankle can then consume the energy harvested in the spring from the knee.

- *Is it possible to design a new mechanism improving the global energy efficiency of a lower limb prosthesis? Is it possible to make it small and lightweight enough to embed it in a lower limb prosthesis?*

Chapters 7 and 8 presented a new technology of Continuously Variable Transmission. This technology was developed with the aim to embed it in a lower limb prosthesis. The transmission ratio of this CVT can be changed even when the mechanism is at rest. This mechanism is thus an ideal candidate for applications in legged robotics. Chapters 8 validated to possibility to bring its size down to the specs of lower-limb prosthesis, and to obtain a good efficiency and a good torque capacity. The CVT transmission ratio can indeed continuously vary by keeping an efficiency between 70 % and 90 %. This proved the possibility to embed the design in a transfemoral prosthesis.

9.3 Discussion and Perspectives

The new concept of prosthesis introduced in this thesis is promising in theory. However, to be efficient, a good CVT is needed. By good CVT, we mean a CVT (i) which has high energy efficiency, (ii) which can be turned into an IVT, (iii) which is miniaturisable and lightweight, and last but not least (iv) which is able to change ratio at rest. The CVT technology we developed in this thesis can fill these requirements, although there is still a lot of elements requiring validation. First of all, the hypothesis about IVT efficiency should be adapted in the model presented in Chapter 5. Indeed, in the model, the efficiency is assumed constant and equal to 60 %. However, Chapter 8 showed that the efficiency of the IVT is not constant with the ratio but follows a V-shape as a function of the transmission ratio. A efficiency model similar to the one presented by Mantriota [66, 67] should be applied. Secondly, the actuation concept described in Chapter 5 should be tested with the CVT prototype

presented in Chapter 8. A test bench would validate or not the proposed prosthesis concept. This bench, would make possible the implementation of different control strategies and verify the theoretical results presented in the thesis.

9.3.1 Comparison with existing prosthesis

The prosthesis concept is good in theory but still has to be validated with a prototype. A prosthesis prototype should be designed and challenged in comparison with existing top of the line prostheses.



Fig. 9.1: CSEA knee connected to a BiOM ankle prosthesis from the MIT Biomechanics group [80].

A first comparison can be done with the CSEA-knee combined with the BiOM ankle of MIT [80] regarding weight. This prosthesis is illustrated in Fig. 9.1 and has a mass $m_{MIT} = 4.7$ kg without battery. For our IVT prosthesis, an estimation of the weight of the different elements can be done and is reported in Table 9.1. These values are estimated by taking the weight of similar parts as in the MIT prosthesis. The total weight without battery is estimated to $m_{IVT} = 5$ kg.

The MIT prosthesis consumes 70J/step while theoretically, the IVT prosthesis should consume 39J/step. The battery density of the MIT prosthesis is $\rho_{bat} = 180$ Wh/kg = 648 kJ/kg. Consequently, the relationship between the number of steps that the prosthesis can achieve with one full charge and its total mass is given by Equations (9.1) and (9.2).

Table 9.1: Weight estimation of the elements of the IVT prosthesis

Element	Mass [g]	Quantity
Strucutre	500	1
IVT	800	2
Spring	300	1
Motor	50	1
Ball screw	330	2
Joint	295	2
Minor parts	800	1
Electronics	50	1
Foot	450	1

$$M_{MIT} = m_{MIT} + \frac{70 \, n_{step}}{\rho_{bat}} \tag{9.1}$$

$$M_{IVT} = m_{IVT} + \frac{39 \, n_{step}}{\rho_{bat}} \tag{9.2}$$

Fig. 9.2 illustrates these relationships. For a small number of steps, the MIT prosthesis is lighter. However, for a larger number of steps, the IVT prosthesis becomes lighter due to the fact that the battery is smaller. The crossing point of the curves occurs at about 6700 steps of autonomy.

This comparison relies on crude hypothesis about the weight of the IVT prosthesis. In order to obtain a more reliable comparison, a CAD of the prosthesis should be realised. With this CAD, all the frame and elements would be designed and their mass would be more accurately evaluated.

The results show that our prosthesis might be more favourable regarding weight, under some conditions. Indeed, this new kind of prosthesis can be advantageous for long distance walkers. Indeed, for people working all day sitting on a chair, the autonomy is less important since they can easily change and charge their battery. However, for a person which stand and walk all day long, a longer autonomy is important. On top of that, our prosthesis bring other potential disadvantages. Indeed, CVTs add complexity, make noise and are expensive.

9.3.2 Other actuation systems to be investigated

Chapter 8 showed that when we turn the CVT into a IVT, the efficiency and torque capacity decrease. However a CVT alone cannot provide torque in both directions with the technology proposed in the thesis. One solution to overcome this problem could be to add a parallel spring working against the CVT. In that case, the torque seen at the joint would be the difference between the CVT torque and the parallel

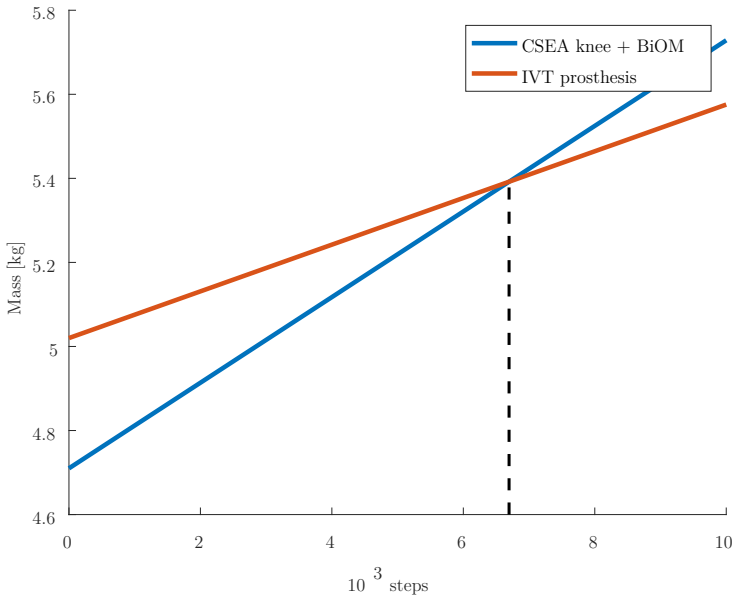


Fig. 9.2: Weight comparison of the MIT prosthesis and IVT prosthesis depending on the number of steps the battery can supply. Without battery, the IVT prosthesis is heavier but its weight increases more slowly as the autonomy requirement increases. After 6700 steps of autonomy, the IVT prosthesis becomes lighter.

spring torque. By dimensioning elements correctly, the CVT could apply a torque only in one direction. This method increases the torque provided by the CVT, but it is counteracted by the fact that the CVT alone has a higher torque capacity than the IVT.

Nowadays, motors and power electronics become more and more efficient. Another approach to control energy transfer in an ankle-knee prosthesis could be electrical. We can imagine SEA or SEAPS prostheses which store and release electrical energy in a capacitor or a battery. In this case, the battery is the storage element and the power electronics are the valves releasing or absorbing the needed energy. In this case, mechanical elements can be removed and the weight can probably significantly decrease.

In our design, we used a planetary gear to turn the CVT into an IVT. However, another solution could have been to use a second CVT. A kinematic diagram of this mechanism is represented in Fig. 9.3. By connecting in series two CVTs through their suns (blue, 3) and sharing the same ring (orange, 1), i.e. two rows of rollers connected kinematically, an IVT can be obtained. Cones (green, 2 and 4) need to be placed in the opposite direction in a way that when the ring is moving, cone radius increase for one of the CVT et decrease for the other. One of the planet carrier (red, 5) is fixed, the ring (orange, 1) is the input and the second planet carrier (red, 6) is the

output. Since the ring is shared, the position x of the ring for one CVT corresponds to the position $X_0 - x$ for the second CVT.

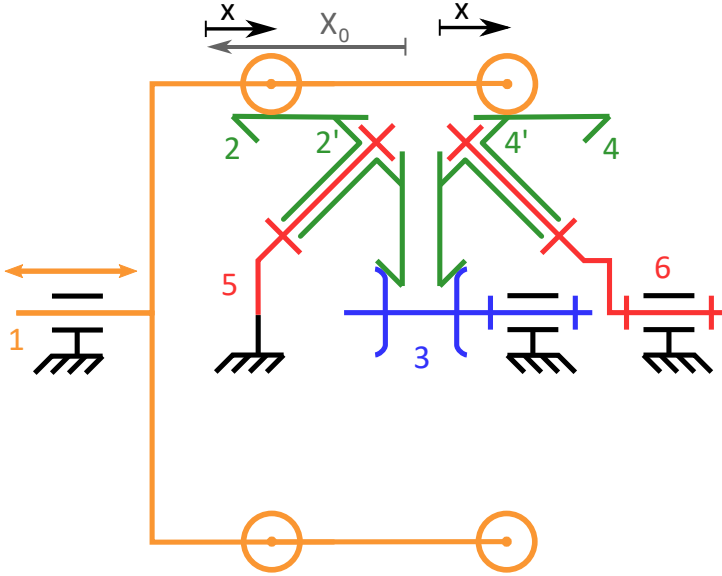


Fig. 9.3: IVT composed of two CVTs connected in series. The suns (blue, 3) are dome-shaped and connected together, the tilted planets (green, 2, 2', 4, 4') are cone-shaped and in opposite direction on both CVTs, the carrier (red, 1, 6) holds the planets, and the ring (orange, 1) is composed of two rows of rollers and is shared between both CVTs.

Equation (9.3) represent the relationship between input and output speeds of this mechanism, i.e. τ_{ivt} .

$$\tau_{ivt} = \frac{\omega_6}{\omega_1} = \frac{1 - \frac{x}{X_0 - x}}{1 + x \frac{R_3 \sin \alpha}{R_2 R_1}} \quad (9.3)$$

This relationship shows that if $\begin{cases} x = \frac{X_0}{2} & \text{then } \tau_{ivt} = 0 \\ x < \frac{X_0}{2} & \text{then } \tau_{ivt} > 0 \\ x > \frac{X_0}{2} & \text{then } \tau_{ivt} < 0 \end{cases}$

By using the same dimensions than the CVT presented in Chapter 8 and $X_0 = 20$ mm, the ratio evolves as pictured in Fig. 9.4 for x varying between 5 and 15 mm.

9.3.3 Comparison with existing CVT technologies

Regarding the CVT technology itself, it has a lot of similarities with existing solutions. For instance, like for most existing CVTs power transmission is achieved by

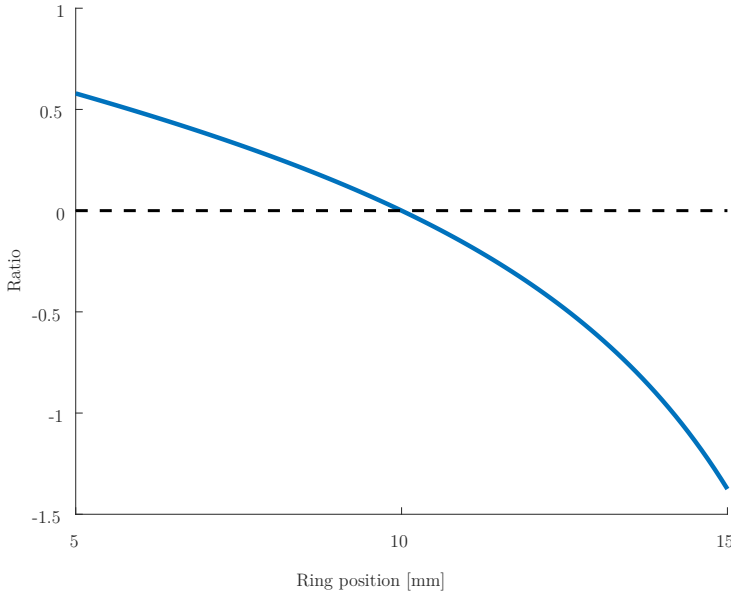


Fig. 9.4: Gear ratio of the IVT composed of two CVTs as a function of the ring position.

friction. Indeed, to obtain an infinite amount of possible ratios, tooth cannot be used to transmit mechanical power.

Like existing CVTs, our design can provide different ratios by moving the contact point radius between two elements. One of the drawback of existing CVTs is that changing the ratio requires for the mechanism to overcome friction (which, on the other hand, is needed to transmit power) and make elements slips at their contact points. This is possible only if the transmission is moving fast enough. Consequently, it is impossible to change the ratio at rest or at low speed of the input shaft. This drawback is not present in our design. Indeed, to change the ratio, the mechanism just requires to overcome rolling friction which is negligible compared to viscous friction of the existing mechanisms. This property allow to use this CVT for walking applications. Indeed, it requires changing the ratio with almost no energy cost, even at rest, i.e. when the input shaft is not rotating.

Concerning the implementation of the technology, a lot of things still have to be validated. Future steps should focus on improving the prototype, mainly regarding its mechanical efficiency. In order to increase the speed efficiency and avoid slipping, contact forces should be increased, for instance by changing the tilt angle of the grooves in the ball ramp. Future experiments should be conducted with different ball ramps in order to find the optimal angle in terms of efficiency. Moreover, increasing the contact forces might decrease the torque efficiency, so that a trade-off

will indeed have to be found.

A second point to be improved is the torque transmission capacity between the sun, cones, and rollers. Now the sun, cones and roller are made of bearing steel. This steel can be hardened and grounded and a surface coating could be applied in order to increase the maximal normal force allowed and thus the maximum torque that the whole mechanism could transmit. Tribology studies could be done to infer what is happening at the contact surface. Mechanical wear should also be investigated in order to estimate the durability of the prototype.

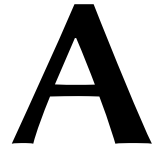
Last but not least, a larger prototype might be assembled in order to conduct experiments with a device for which fabrication tolerances will play a less significant role regarding efficiency.

The CVT technology has been patented and various applications are possible. Indeed, different markets could be interested by this technology. We are thinking, of course, at the vehicle industry but also smaller application like robotics or bigger like wind mills.

Part III

Appendix

Human walking gait: biomechanical data



The purpose of this section is to provide the reader with an overview of the gait cycle, to make the detailed description which follows a little easier to follow. First the mechanisms governing the flat ground human walking cycle are described. Then the biomechanical data (i.e. position, velocity, torque and power) of the different joints are analysed. The cycle is illustrated by Fig. A.5 to A.13.

A.1 Gait cycle

This section is based on the work of Willems et al. [107] who made a complete synthesis on current knowledge about human walking. This section covers the gait cycle kinematics described in their paper.

As described by Winter [108], a normal walking gait cycle is defined as the time interval between two successive contacts with the ground by the same foot, arbitrarily the right one in the rest of this section.

Arbitrarily, the gait cycle is divided into seven different phases, captured in Fig. A.1. The main subdivisions for each lower limb are the stance phase (about 60 % of the gait cycle) and the swing phase (the remaining 40 %). They respectively correspond to the portion of the gait cycle when the foot is in contact with the ground and when it is not [107].

By convention the cycle begins with the first double stance of the right step. At a normal speed of 4 km/h this period goes from 0 % to about 15 % of the cycle (Fig. A.2 A). During this period, the body weight is transferred from the left to the right leg. The first single support phase on the right leg takes place from about 15 % to 50 % of the cycle (Fig. A.2 B). During this phase the left leg swings forward. Then, during the second double support period, from about 50 % to 65 % of the

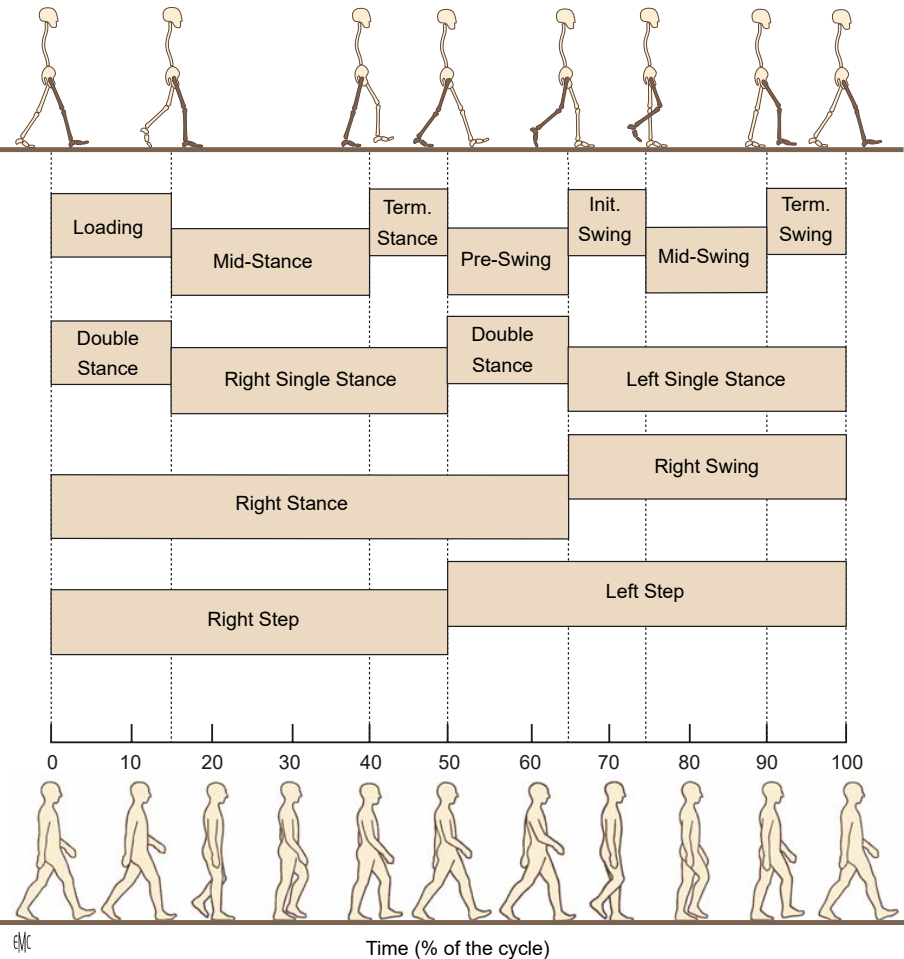


Fig. A.1: Gait cycle division (adapted and translated from [107])

cycle (Fig. A.2 C), the weight is transferred from the right to the left leg. The cycle finally ends up with the second single support period on the left leg, i.e. from about 65 % to 100 % of the cycle (Fig. A.2 D). During this last phase, the right leg swings forward.

A.2 Lower limb joints biomechanical data

During gait, movements occur in all three planes: sagittal, frontal and transverse (see Fig. A.3). Fig. A.4 displays the corresponding joint angles, velocities, torques and powers in three major lower limb joints: hip, knee and ankle, and only in the sagittal plane, which is predominant in walking [108–110].

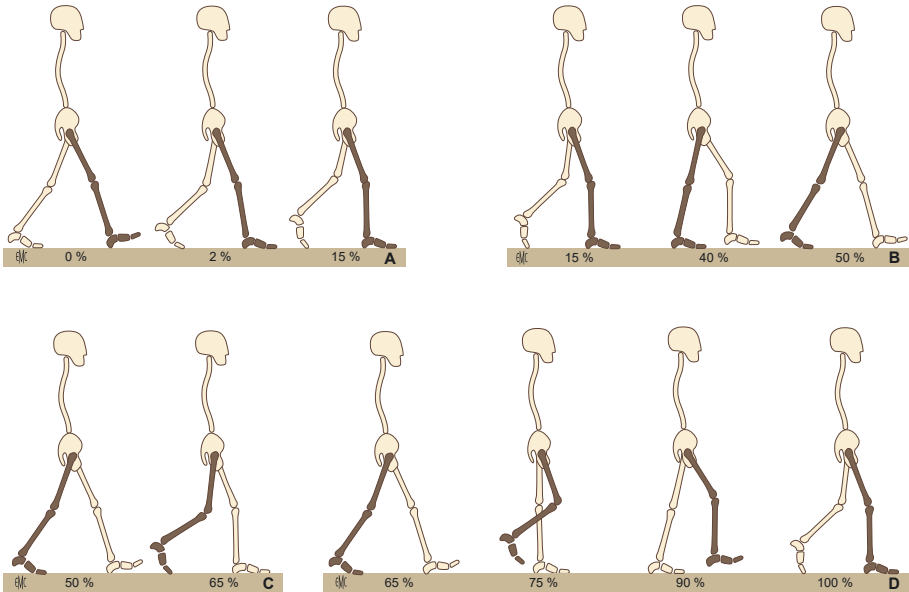


Fig. A.2: Gait cycle division (adapted from [107]). **A** Double stance. **B** Right single stance. **C** Double stance. **D** Right swing.

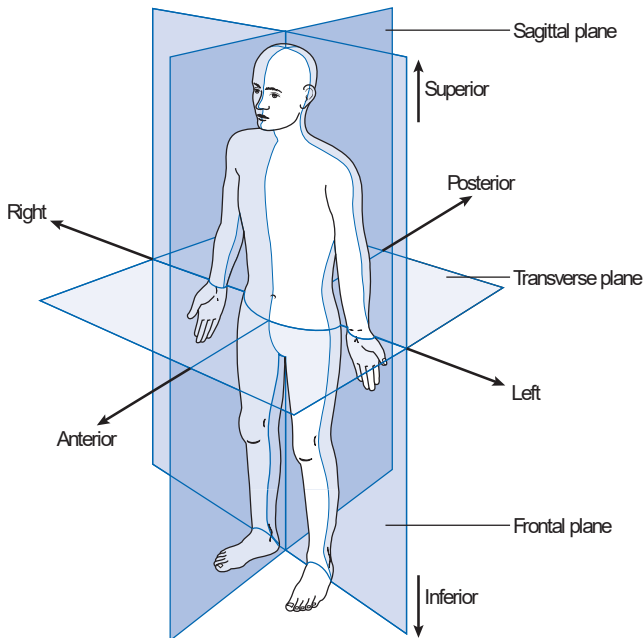


Fig. A.3: The anatomical position, with three reference planes and six fundamental directions [106].

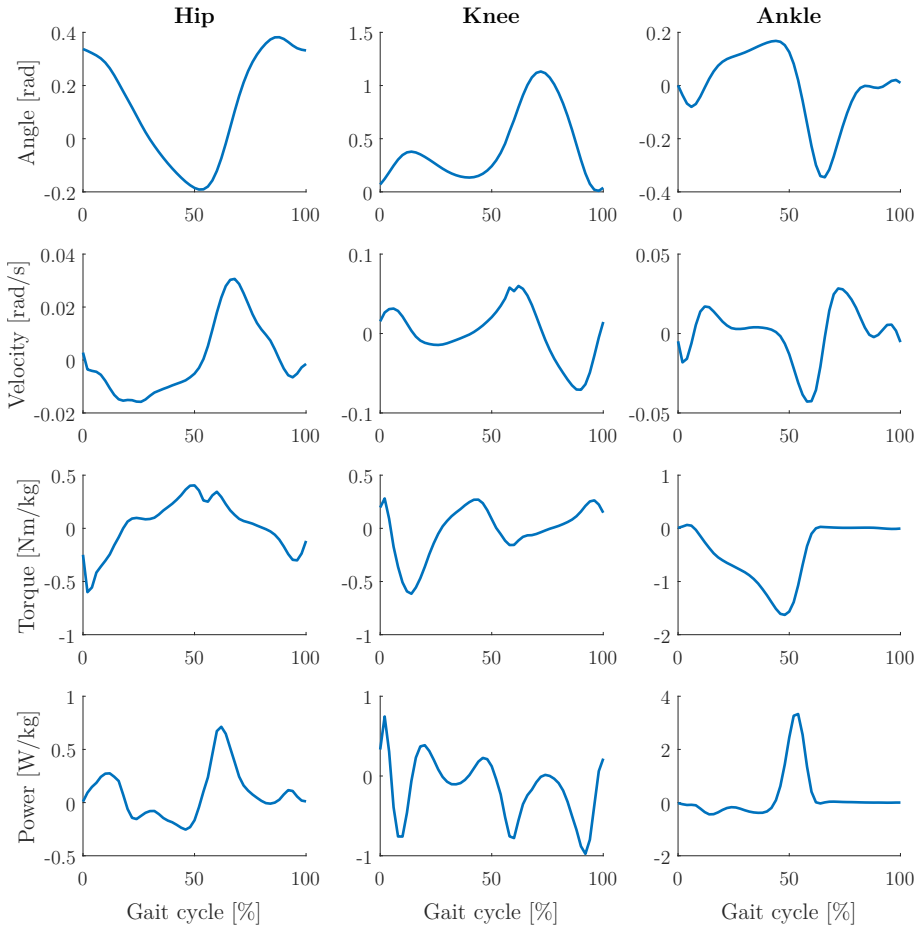


Fig. A.4: Joint positions and velocities during a single gait cycle in the sagittal plane: right hip (flexion positive), knee (flexion positive) and ankle (dorsiflexion positive). Internal joint normalised torques during a single gait cycle in the sagittal plane: right hip (flexion positive), knee (flexion torque positive) and ankle (dorsiflexion torque positive). Joint normalised powers during a single gait cycle in the sagittal plane: right hip, knee and ankle. Power generation is positive, absorption is negative [108].

A.2.1 Joints kinematics

The kinematics description is based on the book "*Gait analysis: an introduction*" by Whittle [106].

Hip The maximum flexion of the hip is reached approximately at the middle of the swing phase and lasts until initial contact. During the loading response, the hip joint begins to extend until peak hip extension is reached at the time of opposite initial contact. Joint motion then reverses in the direction of flexion,

continues to flex during toe off through the whole swing phase until initial contact.

Knee The knee shows two flexion and two extension peaks during each gait cycle. The knee extends rapidly at the end of the swing phase, achieves straight position just before initial contact, and then starts flexing again during the loading response, initiating the stance phase flexion. The knee joint reaches its flexion peak in the stance phase at between 15% and 20% of the gait cycle, close to the time of heel rise, after which it starts to extend again. By the time of opposite initial contact the knee is already flexing, reaching the peak in the swing phase. One fraction of the knee flexion is due to simultaneous hip flexion: the leg is like a articulated double pendulum, so when the hip flexes, the shank is left behind, due to its inertia, resulting in knee flexion. The peak of knee flexion is usually between 60° and 70° before the feet are adjacent, by which time the knee starts to extend again rapidly until full extension prior to initial contact.

Ankle At the time of initial contact, the ankle is generally close to its neutral position at 0° . After initial contact, the ankle plantarflexes, bringing the forefoot down onto the ground. During mid-stance, the tibia moves forward over the foot, and the ankle joint becomes dorsiflexed. Before opposite initial contact, the ankle angle again changes, a major plantarflexion taking place until just after toe off. During the swing phase, the ankle moves back into dorsiflexion until the forefoot has cleared the ground (around feet adjacent), after which something close to the neutral position is maintained until the next initial contact.

A.2.2 Joints torques and powers

Initial contact (Fig. A.5) At the time of initial contact, there is an internal extension torque at the hip. As the hip joint moves in the direction of extension, muscles contract concentrically and generate power. The knee shows an internal flexion torque to prevent hyperextension at the end of the swing phase. As the knee starts to flex, a short-lived power generation occurs. Little torque or power exchange occurs at the ankle until just after initial contact. The heel strike involves an absorption of energy by the elastic tissues of the heel and by compliant materials in footwear, very little of which could be recovered later in the stance phase. The amount of energy lost to the environment as sound and heat in this way is probably fairly small.

Loading (Fig. A.6) The hip shows an internal extension with power generation during the loading response and the knee shows an internal flexion torque with power generation. At the ankle, the posterior placement of the force vector (Fig. A.6) produces an external plantar flexion torque. In the normal individual, this is resisted by an internal dorsiflexor torque absorbing power and permitting the foot to be lowered gently to the ground.

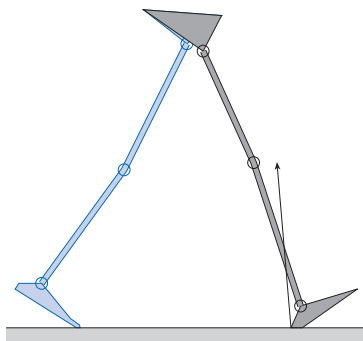


Fig. A.5: Initial contact: position of right leg (gray), left leg (blue) and ground reaction force vector during the heel strike transient. This illustration also applies to terminal foot contact [106].

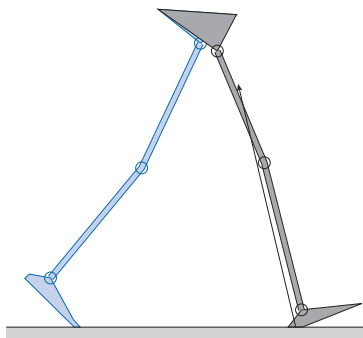


Fig. A.6: Loading response: position of right leg (gray), left leg (blue) and ground reaction force vector 20 ms after initial contact [106].

Opposite toe off (Fig. A.7) At opposite toe off, the hip continues to have an internal extension torque with power generation. At the knee, the force vector lies behind the joint (Fig. A.7) producing an external flexion torque. This is opposed by an internal extension torque absorbing power. The line of the ground reaction force begins to move forwards along the foot (Fig. A.7), causing the internal dorsiflexion torque at the ankle to become smaller and then to reverse, to become a plantar flexion torque. Little power exchange occurs at the ankle at this time.

Mid-stance (Fig. A.8) During mid-stance, the internal extension torque at the hip declines and disappears, to be replaced by a torque in the opposite direction. At the knee, the force vector remains behind the joint, producing an external flexion torque, opposed by an internal extension torque, due to. As the direction of knee motion changes from flexion to extension, power generation takes place. The ankle shows an increasing internal plantar flexion torque throughout mid-stance and into terminal stance, as the force vector moves into the forefoot. It is absorbing power.

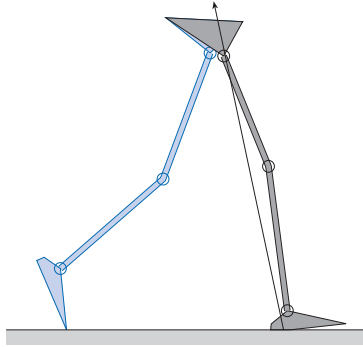


Fig. A.7: Opposite toe off: position of right leg (gray), left leg (blue) and ground reaction force vector [106].

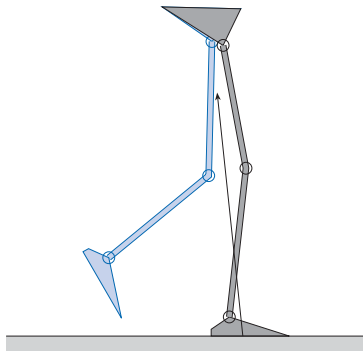


Fig. A.8: Mid-stance: position of right leg (gray), left leg (blue) and ground reaction force vector 100 ms after opposite toe off [106].

Heel rise (Fig. A.9) At heel rise there is a small but increasing internal hip flexion torque with a resulting power absorption. At the knee, the internal torque has reversed to become flexion. According to Perry [74], this occurs because the upper body moves forward faster than the tibia. If the ankle joint were totally free, the forward motion of the body would simply dorsiflex the ankle. However, Internal torque slows down the forward motion of the tibia so that as the femur moves forwards, an external extension torque is generated at the knee, opposed by an internal flexion torque. Only small and variable power exchanges occur at the knee around heel rise. At the ankle, the internal dorsiflexion torque continues to increase with power absorption.

Opposite initial contact (Fig. A.10) A peak of hip internal flexion torque occurs around opposite initial contact. As the direction of hip motion reverses from extension to flexion, power absorption is replaced by power generation. During terminal stance, flexion of the knee brings the joint in front of the force vector (Fig. A.10), re-

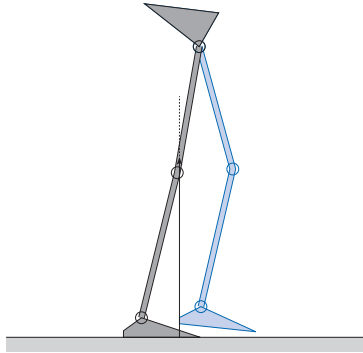


Fig. A.9: Heel rise: position of right leg (gray), left leg (blue) and ground reaction force vector [106].

versing the external torque from extension to flexion and hence hanging the internal torque from flexion to extension. It results in power absorption. At the ankle, the force vector is well in front of the joint at opposite initial contact (Fig. A.10). The resulting high external dorsiflexion torque is opposed by a correspondingly high internal plantar flexion torque (Fig. A.10). The result is a large generation of power, which is the highest power generation of the entire gait cycle. The immediate effect of this power generation is to accelerate the limb forward into the swing phase.

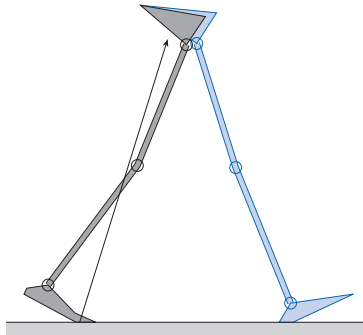


Fig. A.10: Opposite initial contact: position of right leg (gray), left leg (blue) and ground reaction force vector [106].

Toe off (Fig. A.11) Around toe off, the hip still shows an internal flexion torque. Since the hip is flexing at this time, power generation occurs. During pre-swing and initial swing, hip flexion causes the knee to flex, the double pendulum resulting in an external flexion torque at the knee, opposed by an internal extension torque to limit the speed at which the knee flexes and absorbs power. At the ankle, the internal plantar flexion torque reduces rapidly during pre-swing as the magnitude of the ground reaction force declines, falling to zero as the foot leaves the ground at

toe off. The ankle power generation peak also declines to around zero during this period.

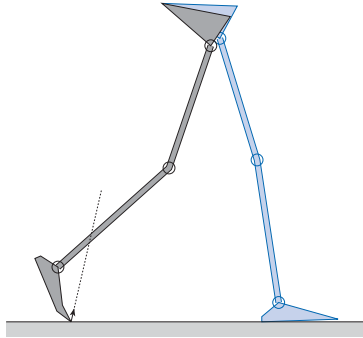


Fig. A.11: Toe off: position of right leg (gray), left leg (blue) and ground reaction force vector [106].

Feet adjacent (Fig. A.12) As the hip moves into flexion, from opposite initial contact, through pre-swing and initial swing until the feet are adjacent, an internal flexion torque is present. Hip flexion, in response to this moment, results in the highest peak of power generation at the hip, the power being used to accelerate the swinging leg forward. The resulting kinetic energy is later transferred to the trunk, as the swinging leg is decelerated again at the end of the swing phase. Between toe off and feet adjacent, the knee continues to show a small internal extension torque which prevents the knee from flexing too rapidly in response to the external flexion torque transferred from the hip. While the knee is still flexing, power absorption occurs. Only very small torque and power exchanges are seen at the ankle, since only the weight of the foot is involved.

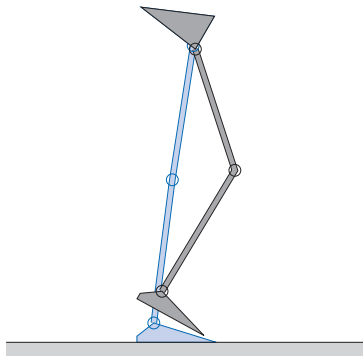


Fig. A.12: Feet adjacent: position of right leg (gray) and left leg (blue) [106].

Tibia vertical (Fig. A.13) At the hip, by the time of tibia vertical, an increasing internal extension torque is seen. This torque probably permits the transfer of mo-

momentum from the swinging leg to the trunk, recovering some of the kinetic energy imparted to the leg in initial swing. Since the hip angle is essentially static during terminal swing, very little power exchange occurs at the joint itself. The knee demonstrates an increasing internal flexion torque with power absorption. This occurs in response to an external extension torque, generated by the inertia of the swinging shank, which would hyperextend the knee if it were not checked. The ankle moment remains negligible, with very little power exchange.

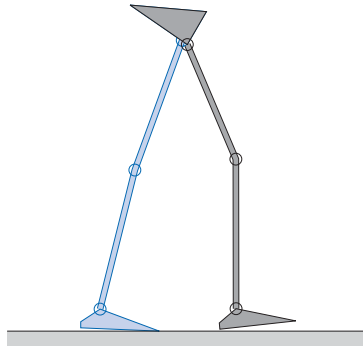


Fig. A.13: Tibia vertical: position of right leg (gray) and left leg (blue) [106].

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