CONTROLLED PASSIVE ACTUATION

Concepts for energy efficient actuation using mechanical storage elements and continuously variable transmissions

DOUWE DRESSCHER

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**CONCEPTS FOR ENERGY EFFICIENT ACTUATION USING
MECHANICAL STORAGE ELEMENTS AND CONTINUOUSLY
VARIABLE TRANSMISSIONS**

DOUWE DRESSCHER

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CONTROLLED PASSIVE ACTUATION

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DISSERTATION

to obtain
the degree of doctor at the University of Twente,
on the authority of the rector magnificus,
prof. dr. H. Brinksma,
on account of the decision of the graduation committee,
to be publicly defended
on Wednesday 9 November 2016 at 14:45

by

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SUMMARY

Energy autonomous robots need to take care of their own energy supply. In the absence of a charging station, the robot will have to replenish its energy in other ways. Alternative energy sources available to a robot in an outdoor environment may include unrefined biomass, sun and wind. As these energy sources are limited, energy efficiency of mobile robots is of key importance.

With this research, we want to make a step toward energy autonomous legged robots. For this, we have two key objectives: increase insight in the mechanisms of energy loss and contribute toward the improvement of energy efficiency.

In the first part of this thesis, the energy losses in legged robots are discussed. The active supporting of the system mass and the additional movement of the legs contribute to the high energy consumption in legged vehicles. However, how significant the contributions of the various causes of energy loss are, is not well understood. We investigate the mechanisms of energy loss in electric actuators to understand their contribution to energy (in)efficiency. As a part of this effort, a method to select a motor-gearbox combination for energy efficiency is presented and evaluated to show the possible impact. We conclude that for most robotic applications, electric actuators are the dominant cause of energy losses. Selecting a motor-gearbox combination for energy efficiency can provide a significant improvement in the efficiency. However, the energy losses in the motor-gearbox combination remain significant.

As an alternative to using only electric actuators for actuation, an actuation principle that incorporates a mechanical storage element and a Continuously Variable Transmission (CVT) in the drive-train is discussed. This actuation principle is named Controlled Passive Actuation (CPA). The envisioned benefits are twofold: first, separation of the energy supply function and the serving function enables using the electric actuator in its most efficient range of operation; and, second, by incorporating the capability to recycle energy within the mechanical domain, the energy loss in the electric actuator can be avoided.

An important component in this actuation principle is the CVT and in the

second part of this thesis, two CVT's that were designed for application in Controlled Passive Actuation are discussed. The first design is a lever with a pivot that moves along the lever. It is shown that the design is conceptually suitable for application in Controlled Passive Actuation and that the mechanical design needs improvement. The second design is based on two spheres of which the relative orientation can be changed to adapt the transmission ratio. This concept is named Dual-Hemi CVT. A prototype was built and experimental results show that it is suitable for application in CPA.

In the third part of this thesis, two versions of CPA are discussed. In the first version of CPA, a flywheel is used as mechanical storage element. By changing the rate of the transmission ratio of the Continuously Variable Transmission, the energy transfer between the source and the load is controlled. A test setup was built and experimental results show that the system can be used to achieve servoing behaviour at a limited bandwidth. In the second version of CPA, a spring is used as mechanical storage element. By changing the transmission ratio of the CVT the energy flow between the source and the load is controlled. A test setup that includes the aforementioned Dual-Hemi CVT was built and experimental results show that the system can be used to achieve servoing behaviour with limited performance.

When applying CPA, achieving good performance and high efficiency has proven to be difficult. By adding a CVT, a complex device with non-idealities that have to be dealt with is added. We have shown that we can separate the energy source function from the servoing behaviour function. We were not (yet) able to obtain good servo performance nor achieve the envisioned efficiency gains.

To conclude: to achieve robots that can move with high energy-efficiency, we need to look beyond currently available electric actuators. Either the motor steepness of electric motors needs to be increased such that they offer an energy efficient way of actuating robots when used in the traditional sense, or additional facilities need to be included in the drive to let the motor operate in its current high-efficiency region and/or reduce the energy flow through the motor. The latter can for example be achieved with CPA. A step has been made on this path but the search should continue. CVTs can help in achieving lower energy losses in the actuators, but the requirements on the CVT are high.

SAMENVATTING

Energieautonome robots moeten zelf zorgen voor hun energievoorziening. In de afwezigheid van een laadstation zal de robot moeten uitwijken naar andere energiebronnen. Ruwe biomassa, zon en wind zijn voorbeelden van alternatieve energiebronnen die beschikbaar kunnen zijn voor een robot. Omdat deze bronnen beperkt energie kunnen leveren is de energie-efficiëntie van robots een belangrijk onderwerp.

Met dit onderzoek willen we een stap maken in de richting van energieautonome robots. Hiervoor stellen we ons twee doelen: het inzicht in de mechanismes van energieverlies vergoten en een bijdrage leveren aan de verbetering van de energie-efficiëntie.

In het eerste deel van dit proefschrift worden de energieverliezen die optreden in lopende robots beschreven. Het actief dragen van het gewicht van de robot en de extra beweging die de lopende poten met zich meebrengen dragen bij aan het hoge energieverbruik van lopende robots. Het is echter niet bekend hoe groot de invloed van de verschillende oorzaken van energieverlies is. We onderzoeken de mechanismes van energieverlies in elektrische motoren om meer inzicht te krijgen in hun bijdrage aan de energie-(in)efficiëntie. Als een onderdeel van dit proces hebben we een methode ontwikkeld om een energie-efficiënte combinatie van motor met overbrenging te selecteren. We hebben deze methode getest om de mogelijke verbetering die deze energie-efficiënte combinatie brengt te laten zien. De conclusie is dat elektrische actuatoren voor de meeste robottoepassingen de dominante oorzaak voor energieverlies zijn. In de selectie van de combinatie van motor en overbrenging kan een significante verbetering in de energie-efficiëntie bereikt worden. De verliezen in de motor en overbrenging blijven echter significant.

Als alternatief voor het gebruik van elektrische actuatoren beschrijven we een actuatieprincipe dat gebruik maakt van een mechanisch opslagelement en een Continu Variabele Transmissie (CVT). Dit actuatieprincipe hebben we Controlled Passive Actuation (CPA) genoemd. We verwachten dat CPA twee voordelen zal brengen: Ten eerste worden de “energieleveren-” en de

“servogedrag-” functies gescheiden. Hierdoor kan de elektrische actuator in zijn meest efficiënte werkbereik gebruikt worden. Ten tweede wordt een mogelijkheid geboden om energie te hergebruiken in het mechanisch domein, hierdoor wordt een deel van het energieverlies in de elektrische actuator vermeden.

Een belangrijk onderdeel in CPA is de CVT. Het tweede deel van dit proefschrift beschrijft twee CVT's die specifiek voor gebruik binnen CPA ontworpen zijn. Het eerste ontwerp is een hefboom waarvan het draaipunt altijd langs de hefboom verplaatst wordt. We laten zien dat het concept geschikt is om te worden toegepast in CPA, hoewel het mechanisch ontwerp nog verbetering behoeft. Het tweede ontwerp is gebaseerd op twee bollen waarvan de relatieve oriëntatie gewijzigd kan worden om de transmissieverhouding aan te passen. Dit concept hebben we Dual-Hemi CVT genoemd. Er is een prototype gerealiseerd en experimentele resultaten laten zien dat het geschikt is voor toepassing binnen CPA.

In het derde deel van dit proefschrift worden twee versies van CPA beschreven. In de eerste versie van CPA wordt een vliegwiel gebruikt als opslagelement (bron). Door de afgeleide van de transmissieverhouding van de CVT aan te passen kan de energieoverdracht tussen de last en de bron geregeld worden. Er is een testopstelling gerealiseerd en experimentele resultaten laten zien dat het systeem gebruikt kan worden om servogedrag met een beperkte bandbreedte te bereiken. In de tweede versie van CPA wordt een veer gebruikt als opslagelement. Door de transmissieverhouding van de CVT aan te passen wordt de energieoverdracht tussen de bron en de last geregeld. Gebruikmakende van de eerder genoemde Dual-Hemi CVT is er een experimentele opstelling gerealiseerd en experimentele resultaten laten zien dat het systeem gebruikt kan worden om servogedrag met beperkte prestaties te bereiken.

In het toepassen van CPA is het lastig gebleken om goede prestaties en hoge efficiëntie te bereiken. Door een CVT aan het systeem toe te voegen wordt er een ingewikkeld apparaat met onvermijdelijk niet-ideaal gedrag toegevoegd. We hebben laten zien dat dat de “energieleveren-” en “servogedrag-” functies gescheiden kunnen worden. We zijn er (nog) niet in geslaagd om goede servo-prestaties of de voorspelde efficiëntiewinst te behalen.

Ter afsluiting: Om tot robots te komen die met hoge energie-efficiëntie kunnen bewegen, moet er voorbij de op dit moment beschikbare elektrische actuatoren gekeken worden. Of de steilheid van elektromotoren moet verbeterd worden zodat ze een efficiënte manier bieden om robots te actueren op de traditionele manier, of er moeten extra faciliteiten in de aandrijving ingebouwd worden zodat de motor in zijn huidige hoog-efficiënte werkbereik gebruikt en/of de stroom van energie door de motor beperkt kan worden. Het laatste kan bijvoorbeeld bereikt worden met CPA. Er is een stap gedaan op dit pad maar het onderzoek moet voortgezet worden. CVT's kunnen helpen om lagere energieverliezen in actuatoren te bereiken, maar de eisen voor de CVT zijn hoog.

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CHAPTER 1

INTRODUCTION

1.1 Introduction

The Netherlands - the name itself means “low countries” - is a geographically low lying country. As shown in Fig. 1.1, more than 60% of the land lies below sea level, including the densely populated “Randstad” region, which is encircled in red. Over 7,000,000 people live and work in this region, which is a little more than 40% of the Dutch population. Without effective flood defences, the parts that lie below sea level would frequently (if not permanently) be subjected to flooding. This makes it important for the Dutch to keep their flood defences in good shape and up-to-date. A major breach would be a disaster in many ways. At the time of writing, the Netherlands is protected by over 3,500 km primary water works - these keep the water of the sea, major rivers and some big lakes out - and 14,000 km of other water works.

Historically, many breaches have occurred in the Netherlands, the big flood of 1953 being the most well known. A storm tide hit the Netherlands, resulting in catastrophic flooding in the south-western delta area (Fig. 1.2). Almost 2000 people lost their lives.

In response to this catastrophe, the Netherlands government installed the

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Figure 1.1: Height map of the Netherlands. The blue areas show the low areas whereas the yellow and red parts show the higher ground. The red ellipse shows the location of the densely populated Randstad area. (c) AHN - www.ahn.nl, edited for readability

Delta Commission. They published the “Delta Plan” in 1960 which included plans for the improvement of a total of 13,000 km dykes and the creation of eleven flood defence installations as shown in Fig. 1.3a - an example is the “Measlantkering” as shown in Fig. 1.3b. The execution of the Delta Plan resulted in better protection of the areas that until then regularly flooded at high tide and the protection of the Dutch waters from salinisation. The Delta Plan was a milestone in the structural improvement of the flood defences of the Netherlands.

An important part of the flood defence system consists of dykes. Recent (2003, 2004) dyke failures have shown that knowledge about the flood-defence systems is insufficient to always prevent flooding. In response to these failures,

(a) Photo of the 1953 flood. The image shows how the flood affected the people living in the flooded area. (Image courtesy of: <https://beeldbank.rws.nl>, Rijkswaterstaat).

(b) Map showing the areas flooded during the 1953 flood. The hatched blue markings indicate the areas that were flooded. (Original image courtesy of Tino Heinicke).

Figure 1.2: The 1953 flood

(a) Eleven flood defence installations that were realised as part of the Delta plan. (Image courtesy of: Classical geographer).

(b) The Maeslantkering (Image courtesy of: www.deltawerken.com).

Figure 1.3: The Delta Plan

an effort was started to investigate if sensor technology is economically viable to increase the insight into the existing flood defences. This effort is hosted by the IJkdijk (loosely translates to Measurement dyke) foundation. As a part of this effort, the ROSE (Energy-efficient design and control of mobile RObotic SENSors) project was initiated as part of the Autonomous Sensor Systems (ASSYS) program of STW under grant 10550. The goal of the ROSE project is to develop a ‘team’ of robots that will function as an autonomous

early-warning system by acquiring data about the composition, consistency and condition of dykes. The final goal is a collaborating ‘team’ of air, ground and water-based robots. The ROSE project is divided in two parts; the first part concerns the design of algorithms for coordinating the network of robotic sensors, was carried out at the University of Groningen and is reported in (Vos, 2015). The second part is the design of the energy-autonomous vehicle which is the focus of this work.

Energy autonomy implies that the robot takes care of its own energy supply. In highly structured environments that are not too isolated from the electrical grid, this can be achieved by providing a charging station where the robot can replenish its energy. However, for a dyke inspection robot this is undesirable or even not possible. Without the presence of a charging station, the robot will have to replenish its energy in other ways. Alternative energy sources available to a robot in an outdoor environment may include unrefined biomass, sun and wind. Of these, robotic energy harvesting from unrefined biomass has not yet progressed beyond the laboratory, although the research is promising (Greenman et al., 2003; Melhuish et al., 2006; Ieropoulos et al., 2010). By contrast, equipment to harvest energy from sun and wind has been available COTS (Commercially Off The Shelf) for several decades and is continually improving. For this reason, we focus on energy autonomous robots that use solar and/or wind energy.

This section presents an exploratory study of an energy-autonomous robot that can be deployed on the Dutch dykes. We start by giving some background information on energy harvesting from sun and wind and the energy-cost of locomotion. This information is combined in an analytic expression to determine the feasible daily operational time of such a vehicle. The parameters in this expression are identified for a dyke robot in the Netherlands to which the theory will be applied.

1.2 Theoretical background

To evaluate the energy autonomy of a robot, the achievable operational (working) time per day can be used as a measure. The operational time per day can be determined from the energy that can be harvested in a day and the average power consumption of the robot by applying the following reasoning. Since we do not add/remove energy to/from the system in any other way than by harvesting, the maximum energy that the system can consume, E_{cons} , is equal to the energy that the system harvests, E_{harv} .

$$E_{\text{cons}} = E_{\text{harv}} \quad (1.1)$$

where the energy that the system consumes is equal to the product of the average power consumption during operation, $\bar{P}_{\text{total}}$, and the operational time

per day, T_{op} :

$$T_{\text{op}} = \frac{E_{\text{harv}}}{\bar{P}_{\text{total}}} \quad (1.2)$$

In the following two sections, the derivation of the harvested energy per day and average power consumption is discussed.

1.2.1 Harvested energy per day

Two types of energy are considered for harvesting: solar energy, E_{s} , and wind energy, E_{wt} . The total amount of harvested energy, E_{harv} is equal to their sum:

$$E_{\text{harv}} = E_{\text{s}} + E_{\text{wt}} \quad (1.3)$$

From weather statistics, the average effective incident energy per square metre per day, $\bar{E}_{\text{sun}}\{Whm^{-2}\}$, can be obtained. Using this information, the average energy generated by the solar panels, $\bar{E}_{\text{s}}\{Wh\}$, can be calculated based on the surface area $S_{\text{s}}\{m^2\}$ and the efficiency $\gamma_{\text{s}}\{\}$ of the solar panel:

$$\bar{E}_{\text{s}} = S_{\text{s}}\gamma_{\text{s}}\bar{E}_{\text{sun}} \quad (1.4)$$

For the calculation of the energy that can be harvested from the wind, the average wind speed over 24 hours, $\bar{v}_{\text{wind}}\{ms^{-1}\}$, is available from weather statistics. Using this information, Betz' law (Betz, 1966) can be used to calculate the maximum average power generation under ideal conditions, using a wind turbine, $\bar{P}_{\text{w}}\{W\}$:

$$\bar{P}_{\text{w}} = 0.5\rho_{\text{air}}\bar{v}_{\text{wind}}^3 S_{\text{w}}C_{\text{p}} \quad (1.5)$$

where $\rho_{\text{air}}\{kgm^{-3}\}$ is the density of the air, $S_{\text{w}}\{m^2\}$ the effective surface area of the wind turbine (sectional area) and $C_{\text{p}}\{\}$ the power coefficient of the wind turbine. Using this, the amount of wind energy that is harvested in a day can be calculated:

$$E_{\text{wt}} = 24\bar{P}_{\text{w}} = 12\rho_{\text{air}}\bar{v}_{\text{wind}}^3 S_{\text{w}}C_{\text{p}} \quad (1.6)$$

The total energy that is harvested in a day is now equal to:

$$E_{\text{harv}} = E_{\text{s}} + E_{\text{wt}} = 24S_{\text{s}}\gamma_{\text{s}}\bar{E}_{\text{sun}} + 12\rho_{\text{air}}\bar{v}_{\text{wind}}^3 S_{\text{w}}C_{\text{p}} \quad (1.7)$$

1.2.2 Average power consumption

The total power consumption, $\bar{P}_{\text{total}}$, can be split into the power consumption of the robot's locomotion system ($\bar{P}_{\text{l}}$) and the power consumption of other equipment ($\bar{P}_{\text{o}}$):

$$\bar{P}_{\text{total}} = \bar{P}_{\text{l}} + \bar{P}_{\text{o}} \quad (1.8)$$

The power consumption of the robot's locomotion system can be calculated using a frequently used measure for the power consumption of a locomotion

system, namely the specific resistance as first described by (Gabrielli & von Karman, 1951). The specific resistance $\epsilon\{\}$ is defined as the ratio of power used for locomotion $P_1\{W\}$ and the product of the weight $m\{kg\}$, earth's gravitational acceleration $g\{ms^{-2}\}$ and the maximum speed $v_{\max}\{ms^{-1}\}$, such that:

$$\epsilon = \frac{P_1}{mgv_{\max}} \quad (1.9)$$

This can be rewritten to:

$$P_1 = \epsilon mgv_{\max} \quad (1.10)$$

such that, when given a certain specific resistance of a locomotion system for a certain maximum speed and mass, the power consumption can be calculated.

Splitting the total mass, m , into the mass of the locomotion system, m_l ; the solar panels, m_s ; the wind turbine, m_w ; and the mass of other equipment, m_o , results in:

$$P_1 = \epsilon(m_l + m_s + m_w + m_o)gv_{\max} \quad (1.11)$$

Let us assume that, when changing the surface area of a wind turbine, the change in depth is negligible. Then, the mass of the wind turbine, m_w , is approximately linearly dependent on the surface area of the wind turbine, S_w :

$$m_w = \eta_w S_w, \quad (1.12)$$

where η_w represents the planar density of the wind turbine. It is reasonable to assume that such an approximation also exists for the weight of the solar panels:

$$m_s = \eta_s S_s \quad (1.13)$$

where η_s is the density of the solar panels. Now:

$$P_1 = \epsilon(m_l + \eta_s S_s + \eta_w S_w + m_o)gv_{\max} \quad (1.14)$$

such that the total power consumption is equal to:

$$\bar{P}_{\text{total}} = \epsilon(m_l + \eta_s S_s + \eta_w S_w + m_o)gv_{\max} + \bar{P}_o \quad (1.15)$$

1.2.3 An analytic expression for the operational time per day

Combining equations 1.2, 1.7 and 1.15 results in:

$$T_{\text{op}} = \frac{S_s \gamma_s \bar{E}_{\text{sun}} + 12 \rho_{\text{air}} \bar{v}_{\text{wind}}^3 S_w C_p}{\epsilon(m_l + \eta_s S_s + \eta_w S_w + m_o)gv_{\max} + p_o} \quad (1.16)$$

which enables the relation between solar panel and/or wind turbine surface area and the operational time to be studied.

1.3 Parameter identification

Using equation 1.16, the operational time per day can be evaluated for an area of solar panel and wind turbine surface. The next step is to determine the parameters used in equation 1.16, based on the case of a dyke inspection robot in the Netherlands. In this section, weather statistics and experimental results that have been achieved for both energy harvesting and locomotion are used to give a value to the parameters. Equipment other than the locomotion system and harvesting equipment is not considered at this point; this implies $P_o = 0$ and $m_o = 0$.

1.3.1 Solar energy harvesting

As mentioned earlier, the study concerns a robot that will be deployed in the Netherlands and the average incident solar energy is used to obtain a realistic value for $\bar{P}_{\text{sun}}$. Fig. 1.4a shows the average solar energy (Whm^{-2}) that is incident per day, for each month of the year, in blue. Fig. 1.4b shows the average solar energy (Whm^{-2}) that is incident per day, for several locations in the Netherlands, in blue.

Figure 1.4: Average incident solar energy in the Netherlands. The figures are shown for a flat surface (source: (KNMI, 2010, 2011, 2012, 2013)) and an optimally tilted surface.

These numbers represent the incident energy on a flat surface and may be further increased by tilting the panels such that the surface is normal to the sun's rays (for details on the calculations, please refer to (Stine & Geyer, 2012)). By applying this correction, we obtain incident energy as shown in green. For the case study, the average of these values is used, which is equal to $4334 Whm^{-2}$.

The maximum theoretically achievable efficiency of solar cells is defined by the thermodynamic limit and is equal to 86% (Mart & Arajo, 1996). However, current levels are at 37.5% for InGaP/GaAs/InGaAs cells in a lab environment and 28.5% for GaAs (thin film) cells in commercially available modules (Green et al., 2012). Commercially available GaAs (thin film) modules can have a efficiency of 23.5%; this is 82% lower than the cell efficiency. The planar density of current state-of-the-art silicon panels is around $2.2kgm^{-2}$ (Solbian, 2013).

In this study, we look at the opportunities offered by currently commercially available modules ($\gamma_s = 0.235$ and $\eta_s = 2.2kgm^{-2}$).

1.3.2 Wind energy harvesting

To obtain the average wind speed, average measurements are used as well. Figure 1.5a shows the average wind speed (ms^{-1}) in the Netherlands, for each month of the year. Fig. 1.5b shows the average wind speed for various locations in the Netherlands.

(a) Average wind speed in the Netherlands, for each month of the year.

(b) Average wind speed in the Netherlands, for each month of the year.

Figure 1.5: Average wind speed in the Netherlands (source: (KNMI, 2010, 2011, 2012, 2013))

For the study, the average of these values is used, which is equal to $4.78 ms^{-1}$. The density of air, ρ_{air} , at $10^\circ C$ is $1.25kgm^{-3}$.

The maximum theoretically achievable efficiency at which an idealised model of a wind turbine can convert the kinetic energy of wind to useful power is defined by Betz's coefficient and is equal to 0.59 (Huleihil, 2009). Currently, the measured power coefficient is in the range 0.4-0.5 (Inglis, 1979). The problem is that currently available wind turbines are large and are not designed for mobile applications. Therefore, it is difficult to say if this number is realistic for a smaller model. However, for this study, we assume that it is realistic to have such a power coefficient for a smaller model. We use $C_p = 0.45$ and the implications of this assumption will be discussed when the results are presented.

Currently available wind turbines are not designed for mobile applications and are therefore not optimised for low mass. For mobile applications, it is desirable to have a light turbine with a planar density of (say) 10kg per square metre of surface area. It is assumed that such a wind turbine is available ($\eta_w = 10\text{kgm}^{-2}$), and the implications of this assumption will be discussed when the results are presented.

1.3.3 A locomotion system

In this section, we explore the technical means for a robot to move around on a dyke surface. We start by sketching the terrain of a dyke and use this to compare several options for locomotion systems. Based on this comparison, a type of locomotion system is selected from which a state-of-the-art example is used to identify the parameters that are required to evaluate the energy autonomy of such a system.

The terrain

The terrain of most dykes is not well defined. Apart from keeping the water out, dykes have other purposes. Some have roads on top; others are used for keeping animals like sheep. Due to these differences, the terrain on which the robot has to move may vary from dry sand to high grass to rocks, but may also include well-maintained roads. Fig. 1.6 shows a few examples.

Figure 1.6: Some terrains that can be encountered on a dyke and various levels of inclination. From left to right: Paved top layer (Image courtesy of Diederik Santema), Rocky top layer (Image courtesy of Frank van Tol), Grazing cattle (Image courtesy of: Taco Meeuwssen).

Furthermore, dyke slopes have various and often unknown levels of inclination.

Selection of a locomotion type

The topic of locomotion in difficult terrain has been surveyed before by, among others, (Hardarson, 1997; Seeni et al., 2008). Their work will be used as a

starting point. Three basic locomotion types can be distinguished: wheels, tracks and legs.

Figure 1.7: The three basic locomotion types. From left to right: An example of a wheeled robot (image courtesy of ESA - AOES Medialab), an example of a tracked robot (image courtesy of Mattracks) and an example of a legged robot (image courtesy of FZI Research Center for Information Technology)

To select a suitable locomotion system, several criteria need to be considered. First of all, the robot needs to be able to navigate the terrain as described in Section 1.3.3. Second, the environmental footprint (i.e. the potential damage that a system can do to the soil) must be considered. Third, the energy efficiency must be considered.

The performance of wheeled, tracked and legged locomotion differs depending on the terrain as explained below.

- On loose soil, tracks outperform wheels and legs (Seeni et al., 2008). They provide better traction where wheels are prone to slip, and have a larger contact surface whereas legs are prone to sinking.
- On even, structured terrains, wheels outperform tracks and legs in terms of speed (Seeni et al., 2008). However, none of the locomotion types have difficulty traversing this type of terrain.
- On uneven or rocky surfaces, legged vehicles outperform both wheeled and tracked vehicles. Wheels have proven to be able to traverse obstacles up to the height of the wheel diameter (Hardarson, 1997; Seeni et al., 2008). Tracked vehicles can traverse much larger obstacles than wheeled vehicles (Hardarson, 1997; Seeni et al., 2008). Intrinsically, legged vehicles are highly suitable to uneven terrain (Hardarson, 1997; Seeni et al., 2008). Therefore, they have better obstacle traversal capabilities (Seeni et al., 2008).
- Legged and tracked vehicles have proven to be superior to wheeled vehicles in terms of slope climbing. A typical maximum inclination of a slope that wheeled vehicles can cope with is 15° while tracked and legged vehicles have shown to have no difficulty with slopes of at least 30° (Seeni et al., 2008).

How large the environmental footprint of a locomotion system is depends on the amount and the way of interaction with the terrain. Legged vehicles have much less interaction with the environment than wheeled and tracked vehicles. Therefore, the environmental footprint of legged vehicles is the smallest (Hardarson, 1997). Tracked vehicles are notorious for being rough with the environment, especially when turning (Hardarson, 1997).

When considering ideal rolling and ideal walking, neither wheeled, tracked nor legged locomotion experience energy losses. However, in the non-ideal world they perform differently on different types of terrains. The following rough claims can be made:

- Tracks are designed to provide more traction than wheels. Conversely, they are also less efficient due to this property.
- The more even and structured a terrain is, the more rolling approaches ideal rolling. This suits the higher efficiency of wheeled locomotion.
- When the terrain is uneven and unstructured, rolling is far from ideal. However, ideal walking is, in theory, still possible. This is a more realistic scenario for this project.
- Tracked vehicles turn using skid-steering, which is highly energy-inefficient.

In the literature, wheeled locomotion is said to be much more efficient than tracked or legged locomotion (Hardarson, 1997; Seeni et al., 2008). This is concluded based on the work the researchers incorporated in their study and not beyond dispute. However, it does point out that energy-efficient walking on unstructured terrain remains an unsolved issue.

The above properties of legged, tracked and wheeled locomotion are summarised in Table 1.1. This shows that locomotion with legs is the only solution that can possibly meet all the requirements. However, the literature also pointed out that the energy efficiency of current legged robots is poor relative to wheeled and tracked robots (Hardarson, 1997; Seeni et al., 2008).

Parameter identification

Many legged robots have been developed over the years. To identify a realistic set of parameters for the locomotion system, a state-of-the-art example will be selected and utilized. In (Gregorio et al., 1997), the specific resistance of a range of “vehicles” is compared. Fig. 1.8 shows the results. The locomotion system of the Scout II is, with a specific resistance of 1 at a velocity of 1ms^{-1} , one of the most energy-efficient multi-legged walking systems currently available. Although these are experimental results from a lab environment, it is assumed that it is possible to locomote with this efficiency outside the lab, and therefore, the locomotion system of the Scout II robot is used as an example locomotion

	Moving on loose soil	Moving on rocks	Moving on even terrain	Slope climbing	Environ- mental footprint	Energy efficiency
Wheels	±	±	+	-	±	+
Tracks	+	±	+	+	-	-
Legs	±	+	+	+	+	±

Legend:

+	good performance
+/-	sufficient performance
-	bad performance

Table 1.1: A comparison of the performance of three basic locomotion types on six points that are relevant for an energy autonomous dyke robot. Colours and markings are used to grade the locomotion types as shown in the legend.

system in this case study. Therefore, ϵ is equal to 1, $v_{\max}$ is equal to $1ms^{-1}$ and m_l is equal to $25kg$ (including batteries and electronics)(Poulakakis et al., 2005). For the gravitational acceleration, g , a value of $9.81ms^{-2}$ is used.

1.3.4 Temporary energy storage

Depending on the short-term fluctuations in energy harvesting during operations, the robot needs to be equipped with temporary energy-storage capabilities in the form of batteries. For this analysis, it is assumed that the amount of battery storage with which the Scout II robot is equipped is sufficient to achieve this goal.

1.4 Exploratory study

Now that the parameters have been identified, they can be substituted in equation 1.16. This results in the following expression:

$$T_{op} = \frac{1018.49S_s + 737.2S_w}{245.25 + 21.58S_s + 98.1S_w}, \quad (1.17)$$

such that the (yearly average) operational time per day can be expressed as a function of the solar panel and wind turbine surface area. When plotting the yearly average operational time per day as a function of the surface area of the solar panel and/or wind turbine, Fig. 1.9 is obtained; where the horizontal axis represents the area of solar panels/ wind turbines and the vertical axis the operational time. For this case study, we assume that the robot can carry up to twice its top surface in solar panels; for the Scout II robot, this is $1.637m^2$ of solar panel. Furthermore, we assume a wind turbine with a radius of the

Figure 1.8: The specific resistance and maximum speed for several locomotion systems (Original image courtesy of (Gregorio et al., 1997)).

length of the robot body at most; this results in a maximum surface of $0.55m^2$ for the Scout II robot.

From this, it can be seen that it is always beneficial to include solar panels when applying a wind-turbine (Fig. 1.9b). In addition, it shows that up to a certain amount of solar panel surface area, it is beneficial to include a wind turbine when applying solar panels (Fig. 1.9a). This boundary is at $2.15m^2$ of solar panel surface area. If the solar panel surface area is smaller than $2.15m^2$, the presence of a wind turbine offers an improvement in the feasible operational time per day. However, if the maximum amount of solar panel surface is applied, the addition of a wind turbine offers only a small improvement - from 5.96 to 6.21 hours per day.

To obtain these results, we have made bold assumptions on the efficiency and weight of the wind turbine. In Fig. 1.10a and Fig. 1.10b the effect of variations in these assumptions is shown.

Fig. 1.10a shows that if the wind turbine planar density is more than $13kg/m^2$, it is no longer beneficial to include a wind turbine if the maximum area of solar panels is applied. Fig. 1.10b shows that when the efficiency of the wind turbine drops below 0.36, it is no longer beneficial to have a wind turbine when the maximum amount amount solar panels is applied.

(a) The operational time per day vs the solar panel surface area in the case of no wind-turbine (in blue) and the case of a 0.55m^2 wind turbine (in green).

(b) The operational time per day vs the wind turbine surface area in the case of no solar panels (in blue) and the case of a 1.637m^2 wind turbine (in green).

Figure 1.9: Case evaluation.

(a) Variations of the wind turbine weight per square metre η_w .

(b) Variations of the wind-turbine efficiency C_p .

Figure 1.10: A variation of Fig. 1.9 with the addition of the red line that shows how point A moves with variations of the wind turbine weight per square metre η_w (left) and wind-turbine efficiency C_p (right), the red dots represent the specific values that are given alongside multiplied by the amount of weight per square metre as used in the study and the specific values that are given alongside, respectively.

Later in this thesis, we will focus on energy-efficient locomotion. Therefore, it is relevant to look at the effect of variations in the parameters that are associated with the locomotion system. These are the mass of the locomotion system, including batteries (m_l), the specific resistance (ϵ) and the maximum speed (v).

Figure 1.11 shows the results of varying m_l . It shows that a variation in the weight of the locomotion system significantly affects the operational time

per day.

Figure 1.11: A variation of Fig. 1.9 with the addition of the red line that shows how point A moves with variations of the mass of the locomotion system m_1 , the red dots represent values of 0.5, 0.94 and 1 multiplied by the value used in the study

Varying ϵ and v directly scales the operational time.

In (Dresscher et al., 2015a), a more elaborate generalisation of the results is discussed.

1.5 Scope

From this study, it follows that a dyke robot should be a dyke walker in order to handle the variations in terrain and that even if only a few assumptions prove valid, it is realistic to have an energy-autonomous walking dyke robot in the Netherlands. From the results, we see that the mass, m_1 and specific resistance, ϵ , of the locomotion system have a significant influence on the performance of the system.

With this research, we want to contribute to decreasing the specific resistance of legged robots by reducing the energy consumption in the locomotion system. This way, we want to make a step toward energy autonomous legged robots. For this, we have two key objectives:

1. Increase insight in the mechanisms of energy loss
2. Contribute toward the improvement of energy efficiency

1.6 Thesis outline

The first step towards decreasing the energy consumption in legged robot is an analysis of the causes of energy consumption, which is discussed in chapter 2. We conclude that “For most robotic applications, electric actuators are

the dominant cause of the energy losses” and we suggest Controlled Passive Actuation to reduce these losses. Controlled Passive Actuation applies a Continuously Variable Transmission and a mechanical storage element to achieve servoing behaviour. In chapters 3 and 4, two possible designs of CVTs are discussed. In this work, we discuss two versions of Controlled Passive Actuation: Inertia-driven, which is discussed in chapter 5; and spring-driven, which is discussed in chapter 6. Chapter 7 concludes this work by piecing together the insights and contributions from the work, reflecting on the implications for the research field and recommendations for future research.

CHAPTER 2

ENERGY LOSSES IN LEGGED ROBOTS

2.1 Introduction

In chapter 1, we argued that a dyke robot should be a dyke walker in order to handle the terrain in an elegant way. The specific resistance of such a robot is of significant influence on its energy autonomy. Compared to wheeled and tracked robots, legged robots are known to have a significantly higher energy consumption (Hardarson, 1997; Seeni et al., 2008). Their surveys discuss several courses of low performance that are related to this, see the orange blobs in Fig. 2.1. They mention that legged vehicles have a *higher number of actuators*, are *heavier*, have a *low payload to weight ratio* and a *higher energy consumption* and *suffer an impact at every step*. With some reasoning, this chart can be extended with the points in the blue blobs. The arrows indicate cause and effect relations that were found in literature.

It is well known that the active supporting of the system mass and the additional movement of the legs contribute to the high energy consumption in legged vehicles, these are shown in the green blobs in Fig. 2.1. However, how significant the contributions of the various causes of energy loss are, is not well understood.

This chapter is based on Dresscher, D., de Vries, T. J. A., & Stramigioli, S. Motor-gearbox selection for energy efficiency, In press: 2016 IEEE International Conference on Advanced Intelligent Mechatronics (AIM).

Figure 2.1: A schematic overview of cause and effect relations underlying the low performance in legged locomotion. The orange blobs indicate items that were mentioned in the surveys of (Hardarson, 1997; Seeni et al., 2008), the blue blobs indicate items that were added based on reasoning and the green blobs indicate items that are suspected to have a relation in the scheme.. The arrows indicate cause and effect relations that were found in literature.

Many studies on the energy efficiency of and energy losses in legged robots can be found. Here, we discuss a selection of the literature.

For unknown reasons, most of the studies that were found only investigate mechanical aspects other than the actuator (for example: (Zielinska, 2003; Zhoga, 1998; Vohnout, 1982; Schiehlen, 2005; Marhefka & Orin, 1997; Kar et al., 2001; Gregorio et al., 1997)). We believe that it is necessary to consider the complete mechatronic system.

The authors of (Silva & Machado, 1999; Silva et al., 2002) focus on the effect of gait parameters on the power consumption. A simple model is used for the actuator losses: they take the losses as proportional to the squared torque. They do not distinguish between different causes for losses in the actuators. Their results can be combined to get an impression of the energy losses in the actuators compared to other parts of the system. However, as they only evaluate a quantity related to the actuator losses, it is not possible to compare it with the mechanical power output. In (de Santos et al., 2009), a method to minimise the energy consumption by changing the gait is discussed.

They do have a proper model for the electric energy losses but do not consider mechanical energy losses in the drive. Additionally, they do not consider energy recovery through the electric drive.

The authors of (Raby & Orin, 1999) focus on the effect of gait parameters on the overall energy consumption, but do not differentiate between different sources of energy loss.

In (Kaneko et al., 1987), a model is proposed to evaluate the efficiency of walking machines. However, in this model, actuator losses are not considered.

To evaluate the significance of the contribution of actively supporting the system mass and additional movement of the legs to the energy losses, we need to investigate the mechanisms of energy loss in electric actuators. Previous work suggests that losses in the actuators of robots are a significant source of energy consumption. In (Poulakakis et al., 2005), more than a factor two difference between the mechanical specific resistance and the electrical specific resistance of the Scout II robot is reported. This points toward significant losses in the electrical domain. In the experimental results presented in (Seok et al., 2015), 76% of the total power is lost in the windings of the electric motor and 24% is lost in mechanical friction. They conclude that, for any legged robot, the design of the drive is critical to increasing the efficiency. They expect that the electric losses can be reduced by increasing the gear ratio. They mention that the trade-off between energy losses and performance needs to be investigated.

The effect of choices in motor-gearbox combinations on the performance of a mechatronic system has been discussed before in, for example, (Van de Straete et al., 1998), (Van de Straete et al., 1999), (Cetinkunt, 1991), (Roos et al., 2006) and (Folkertsma et al., 2012). However, when looking at the performance of a mechatronic system in a traditional way, energy efficiency is not considered. As a result, we only see it mentioned in the latter ((Roos et al., 2006)) as an option for optimisation. They suggest the use of simulations to compare the energy consumption over a load cycle. In (Seok et al., 2012), the design of electric drives that are optimised for torque density and transmission transparency is discussed. As efficiency is not their (primary) goal, it is not discussed.

An alternative and complementary approach to decreasing the energy consumption by electric actuators is the inclusion of series or parallel springs. The possibilities of this approach have been shown by several authors already, including (Au & Herr, 2008), (Holgate et al., 2008), (Brown & Ulsoy, 2011), (Plooijs & Wisse, 2012) and (Rouse et al., 2013). The use of series and parallel springs in actuation brings along several additional design constraints and is not the focus of this chapter.

In this work, we present a model-based motor-gearbox selection procedure with the focus on key mechanisms of energy dissipation. Using an example, we evaluate the possible efficiency gain when selecting a motor-gearbox combina-

tion for energy efficiency.

We start with a discussion of the model structure that we use throughout this work. In Section 2.3, we build on the model to obtain criteria for the selection of an energy efficient motor and gearbox, followed by a new combined motor-gearbox selection procedure. We continue with the discussion of an example in Section 2.4 where we evaluate the efficiency gains that can be obtained by applying the aforementioned selection procedure on an existing setup. In section 2.5, the insights on the efficiency of motor-gearbox combinations are brought back to the topic of legged locomotion and conclusions are presented.

2.2 A motor-gearbox model

2.2.1 Model formulation

To identify the energy losses in legged locomotion, we look at a basic mechatronic system: a DC-motor with gearbox and an inertial load. Brushless motors are treated in the same way as DC-motors, assuming that the motor driver uses field oriented control. Based on a model of this system, a method is developed to select a motor-gearbox combination for maximum efficiency.

Fig. 2.2 shows the model that is used throughout this work. The model consists, from left to right, of the motor - including winding inductance, resistance, rotor inertia and friction (Named “Full DC Motor Model Approach” in (Verstraten et al., 2016)) - , gearbox - including gearbox inertia and friction - , and load inertia. In this study, we assume the use of a planetary gearbox, which is widely available and fully back-drivable at higher ratios. The friction in such a gearbox is dominated by the coulomb friction (Schempf & Yoerger, 1993), thus, we assume a friction model that only includes coulomb friction. In addition, we assume the gearbox to be backlash free and infinitely stiff. Friction in the load and gravitational forces are not considered at this moment.

In the model, all relations that are used are the standard constitutive relations for the elements except for the elements that represents the coulomb friction in the gearbox, τ_{cgb} . This element contains a standard coulomb friction model given by the following constitutive relation:

$$\tau = \tau_{cgb} \cdot \text{sgn}(\omega) \quad (2.1)$$

where τ_{cgb} is the coulomb friction coefficient, τ the torque and ω the rotational velocity.

²Note: Typically, $r_{gb} < 1$; a gearbox with a ratio 1:5 has $r_{gb} = 0.2$

Figure 2.2: The model that was used to obtain detailed insight in the energy losses. The model includes a DC-motor, gearbox and load. The model parameters are shown in Table 2.1.

2.2.2 A note on the gearbox model

A two-stage gearbox would normally be modelled as shown in Fig. 2.3a. This model can be reduced to the one shown in Fig. 2.3b, where the values of the elements $r_{1,2}$, $I_{1,2}$ and $\tau_{c1,c2}$ are given by Eq. 2.2, 2.3 and 2.4, respectively.

$$r_{1,2} = r_2 r_1 \quad (2.2)$$

$$I_{1,2} = I_2 + r_2^{-2} I_1 \quad (2.3)$$

$$\tau_{c1,c2} = \tau_{c2} + r_2^{-1} \tau_{c1} \quad (2.4)$$

Hence, the general gearbox model as shown in Fig. 2.2 applies both to single-stage gearboxes and multi-stage gearboxes.

Table 2.1: The model parameters that have been used in the dynamic model (Fig. 2.2) to evaluate energetic losses.

Component	Parameter
DC-motor	Inductance, L
	Electric resistance, R_e
	Motor constant, K_m
	Rotor inertia, I_r
	Viscous friction coeff. R_{vm}
Transmission	Gearbox inertia, I_{gb}
	Coulomb friction, τ_{cgb}
	Gear ratio, r_{gb}
Load	Inertia, I_ℓ

Figure 2.3: Bond-graph model of a 2-stage gearbox

2.3 Motor-gearbox selection for energy efficiency

In this section, we discuss the selection of a motor-gearbox combination for energy efficiency. We start by discussing the influence of selecting a motor on the energy efficiency, given a gearbox, then discuss the influence of selecting a gearbox on the energy efficiency, given a motor. We then continue with combining the insights in a process for the selection of a motor-gearbox combination.

2.3.1 DC-motor

Suppose that we want to achieve a certain acceleration a_d of a load I_ℓ . Suppose we have a motor that can deliver a torque τ_m and has a rotor inertia I_r and negligible friction ($R_{vm} = 0$), that we have gearbox with ratio r_{gb} , an inertia I_{gb} (specified at the output) and coulomb friction τ_{cgb} . Then the acceleration, $\dot{\omega}$, of the system load is given by Eq. 2.5.

$$\dot{\omega} = \frac{\frac{\tau_m}{r_{gb}} - \text{sgn}(\omega)\tau_{cgb}}{\frac{I_r}{r_{gb}^2} + I_{gb} + I_\ell} \quad (2.5)$$

where ω is the output velocity.

The acceleration is maximal for a certain τ_m and τ_{cgb} when:

$$r_{gb}^2(I_{gb} + I_\ell) = I_r \quad (2.6)$$

When assuming that τ_{cgb} is not influenced by r_{gb} . This is the principle of inertia matching (Pasch & Seering, 1984).

By substituting Eq. 2.6 for r_{gb} in Eq.2.5 we can obtain the maximum achievable acceleration for the system with matched inertias:

$$\dot{\omega}_{max} = \frac{\frac{\tau_m}{\sqrt{I_r}} - \frac{1}{\sqrt{I_{gb} + I_\ell}} \text{sgn}(\omega)\tau_{cgb}}{2\sqrt{I_{gb} + I_\ell}} \quad (2.7)$$

To at least be able to achieve $\dot{\omega}_{max}$, the right side of Eq. 2.7 needs to be larger or equal to the left side. Refactoring results in Eq. 2.8.

$$\frac{\tau_m}{\sqrt{I_r}} \geq 2\dot{\omega}_d\sqrt{I_{gb} + I_\ell} + \frac{1}{\sqrt{I_{gb} + I_\ell}} \text{sgn}(\omega)\tau_{cgb} \quad (2.8)$$

Assuming that we have an inertia-matched system, any motor that meets Eq. 2.8 can provide the required acceleration, $\dot{\omega}_d$.

Note that this requirement does not contain r_{gb} . Furthermore, an upper bound on τ_{cgb} could be assumed and I_{gb} is often negligible compared to I_ℓ . In this case the selection of the motor can be done without exact knowledge of the gearbox.

This gives some freedom in selecting an efficient motor, but what is an efficient motor? To answer this question, the commonly used motor steepness, S , can be used. The motor steepness relates the motor constant K_m to the winding resistance, R_e , as shown in Eq. 2.9.

$$S = \frac{K_m^2}{R_e} \quad (2.9)$$

Actually, when looking from a port-based perspective we see that it is the inverse of the mechanical representation of the electrical resistance: $\omega = \frac{R_e}{k_m^2} \tau$. The remarkable thing here is that due to the gyrating effect of the electrical motor, it is linear with torque rather than with velocity.

We also know that for a certain output torque, τ_m , a certain current, i , is required depending on the motor constant:

$$i = \frac{\tau_m}{K_m} \quad (2.10)$$

and that the electric losses of the motor, $P_{loss,m}$, are described by:

$$P_{loss,m} = R_e i^2 \quad (2.11)$$

Eq. 2.9, Eq. 2.10 and Eq. 2.11 can be combined to obtain Eq. 2.12.

$$P_{loss,m} = \frac{K_m^2}{S} \left(\frac{\tau_m}{K_m} \right)^2 = \frac{\tau_m^2}{S} \quad (2.12)$$

Eq. 2.12 shows that the motor steepness, S , can be used as a measure for the motor efficiency: the larger the steepness, the smaller the losses will be for a particular required motor torque.

So, given constraints such as Eq. 2.8 and possibly others like size and weight, a motor with maximum steepness should be selected to obtain maximum efficiency for the motor.

2.3.2 Gearbox

Now, the influence of the choice of gearbox on the energy efficiency is discussed, assuming the motor is not changed.

We have an expression for the acceleration of the load in Eq. 2.5, and we know that the maximum acceleration is given by Eq. 2.6.

If we want to achieve at least an acceleration $\dot{\omega}_{min}$, then:

$$\dot{\omega}_{min} \leq \dot{\omega} = \frac{\frac{\tau_m}{r_{gb}} - sgn(\omega)\tau_{cgb}}{\frac{I_r}{r_{gb}^2} + I_{gb} + I_\ell} \quad (2.13)$$

We can rewrite this to:

$$\dot{\omega}_{min} \left(\frac{I_r}{r_{gb}^2} + I_{gb} + I_\ell \right) - \frac{\tau_m}{r_{gb}} + sgn(\omega)\tau_{cgb} \leq 0 \quad (2.14)$$

If we multiply left and right with r_{gb}^2 we obtain:

$$\dot{\omega}_{min}(I_r + r_{gb}^2(I_{gb} + I_\ell)) - \tau_m r_{gb} + r_{gb}^2 sgn(\omega)\tau_{cgb} \leq 0 \quad (2.15)$$

If we then assume that:

1. $\dot{\omega}_{min} > 0$
2. $\tau_m > 0$ and $r_{gb} > 0$
3. worst-case coulomb friction: $\omega > 0$

Eq. 2.15 represents a concave up quadratic equation in r_{gb} with two zeros if $b^2 - 4ac > 0$ where:

$$a = \dot{\omega}_{min}(I_{gb} + I_\ell) + sgn(\omega)\tau_{cgb} \quad (2.16)$$

$$b = -\tau_m \quad (2.17)$$

$$c = \dot{\omega}_{min}I_r \quad (2.18)$$

If this is the case, the quadratic formula can be used to find the roots. As it is a concave up parabola, all r_{gb} that lie between the roots satisfy Eq. 2.14 and thus represent range for gearbox ratios with which at least $\dot{\omega}_{min}$ is achievable. This range is given by:

$$\frac{-b - \sqrt{b^2 - 4ac}}{2a} \leq r_{gb} \leq \frac{-b + \sqrt{b^2 - 4ac}}{2a} \quad (2.19)$$

This gives some freedom in the selection of a gearbox that results in the most efficient motor-gearbox system. But what is the gearbox that results in the most efficient motor-gearbox system? To answer this question, we look at the total losses in the system.

The losses in the motor are given by Eq. 2.12 and the losses in the gearbox can be expressed as:

$$P_{loss,gb} = sgn(\omega)\tau_{cgb}\omega \quad (2.20)$$

Such that the total losses sum up to be:

$$P_{loss} = \frac{\tau_m^2}{S} + sgn(\omega)\tau_{cgb}\omega \quad (2.21)$$

The motor torque, τ_m is dependent on the output torque, τ_{out} , the transmission ratio of the gearbox, r_{gb} and the coulomb friction of the gearbox, τ_{cgb} :

$$\tau_m = r_{gb}(\tau_{out} + sgn(\omega)\tau_{cgb}) \quad (2.22)$$

Eq. 2.21 and Eq. 2.22 can be combined into:

$$P_{loss} = \frac{(r_{gb}(\tau_{out} + sgn(\omega)\tau_{cgb}))^2}{S} + sgn(\omega)\tau_{cgb}\omega \quad (2.23)$$

A gearbox stage can have a ratio that is limited by a lower bound, r_{max} . Multiple stages can be combined to obtain a lower transmission ratio. To obtain the lowest total friction and inertia, the last stage of the gearbox should be tailored to create the desired total ratio while the earlier stages have the maximum stage ratio r_{max} .

When using the gearbox model discussed in Section 2.2 and assuming that:

1. the coulomb friction in a gearbox stage is independent of the ratio of that stage
2. the last stage is used to tailor the total ratio while the earlier stages have the maximum stage ratio r_{max} ,

the losses in the gearbox, motor and the complete system have shapes as shown in Fig. 2.4.

Figure 2.4: The losses in a motor gear-box combination for a given motor over a range of transmission ratios. Discontinuities are present where the number of stages of the gearbox changes.

From Fig. 2.4, we can see that for higher ratios (left in Fig. 2.4) the losses in the motor are dominant and for lower ratios the losses in the gearbox. We also see that adding an additional stage introduces a sharp increase in both the gearbox and motor losses. When decreasing the transmission ratio, the total power consumption decreases and eventually almost flattens out. Therefore, a gearbox with a lower transmission ratio decreases (or at least does not increase) the energy losses and thus, the gearbox with the lowest transmission ratio that still satisfies Eq. 2.19 is the best choice in terms of efficiency.

2.3.3 Combined motor-gearbox selection

So far, we have discussed the selection of a motor, given the gearbox and the selection of a gearbox given the motor. Ideally, the selection of the motor-gearbox combination is done in one process. As discussed in Section 2.3.1, the selection of a motor can be done without exact knowledge of the gearbox by assuming a lower bound on the efficiency of the gearbox and assuming that the gearbox inertia is negligible compared to the load. Using this knowledge,

we suggest the selection process below for the selection of a motor-gearbox combination.

1. Specify the following parameters for the system: $\dot{\omega}_d$ and I_ℓ .
2. Assuming an upper bound on τ_{cgb} and assuming I_{gb} is negligible, use Eq. 2.8 to define a lower bound for $\frac{\tau_m}{\sqrt{I_r}}$
3. Select the motor that has the highest steepness and meets the lower bound on $\frac{\tau_m}{\sqrt{I_r}}$ and possibly other bounds - for example, on size and dimensions.
4. Again assuming an upper bound on τ_{cgb} and assuming I_{gb} is negligible, calculate bounds on the transmission ratio using Eq. 2.19.
5. Within this range, select the gearbox with the lowest transmission ratio.
6. Evaluate if the assumption on the gearbox friction, τ_{cgb} , and gearbox inertia, I_{gb} are valid. If necessary, go back to step 2 and repeat the process with other assumptions.

2.3.4 Motor-gearbox selection based on gearbox efficiency

Sometimes, the gearbox maximum efficiency, η_{gb} , is provided by gearbox manufacturers rather than the coulomb friction. If this is the case, the procedure is alike but constraints are calculated differently.

Eq. 2.8 becomes Eq.2.24

$$\frac{\tau_m}{\sqrt{I_r}} \geq 2\dot{\omega}_d \frac{\sqrt{I_{gb} + I_\ell}}{\sqrt{\eta_{gb}}} \quad (2.24)$$

Eq. 2.19 is unchanged (repeated in Eq. 2.25) but the coefficients a, b and c are now given by Eq. 2.26, 2.27 and 2.28, respectively.

$$\frac{-b - \sqrt{b^2 - 4ac}}{2a} \leq r_{gb} \leq \frac{-b + \sqrt{b^2 - 4ac}}{2a} \quad (2.25)$$

$$a = \dot{\omega}_d(I_{gb} + I_\ell) \quad (2.26)$$

$$b = -\eta_{gb}\tau_m \quad (2.27)$$

$$c = \dot{\omega}_d\eta_{gb}I_r \quad (2.28)$$

In this case, we can assume a lower bound on the efficiency of the gearbox, η_{gb} .

If you have the choice use τ_{cgb} rather than η_{gb} as it is a more accurate representation of the friction.

2.4 Example

Now, we demonstrate the effect of selecting a motor for energy efficiency on the energy efficiency by applying the method described in Section 2.3.3 to a simple mechatronic setup.

2.4.1 Setup

The mechatronic setup that is used in this example is shown in Fig. 2.5. It consists of a Maxon (Maxon Motor, 2015) DC-motor-gearbox combination that can lift a load by means of a pulley. The DC-motor is controlled to follow a

Figure 2.5: A photo of the experimental setup.

motion profile as shown in Fig. 2.6. The light grey marked parts of the motion profile is where the actuator is used to deliver a static force, the white parts show where the actuator is used to create motion.

The setup has a position sensor on the motor shaft. A linear voltage amplifier is used to provide the power for operating the DC-motor.

Table 2.2 lists the physical properties of the setup. The parameters for the motor and gearbox are mostly obtained from the Maxon catalogue (Maxon Motor, 2015). The motor viscous friction, R_{vm} is estimated using the no-load current and rotational velocity. The coulomb friction force in the gearbox, τ_{cgb} , is estimated using measurements and an observer model, details can be found in section 2.4.2.

The model discussed in Section 2.2 is used, with an adaptation of the load to include the gravitational force.

Figure 2.6: The motion profile that is used as reference in the experiments and simulations. The figure shows both the position and the velocity. With this motion profile, the load is moved up and down with three different velocities (white marked parts). In between motion are parts where the actuator has to deliver a static force (light grey marked parts). In the dark grey parts no actuation is performed, they do not affect the results.

2.4.2 Model verification

The model of a DC-motor as shown in Fig. 2.2 is well established and the required parameters are provided by the manufacturer. However, for the gearbox the model is less commonly accepted and the friction parameters are determined experimentally.

A common way to approach this problem is to do parameter identification. However, we use a model with parameters based on physical phenomena and the identification of the parameters is ongoing research - interesting but not the topic of this chapter. Therefore, we have chosen a pragmatic approach: To verify that the gearbox model approximates reality sufficiently close, we do a simple experimental model verification: During the experiment, the real setup is controlled to follow the motion profile as shown in Fig. 2.6. The motor voltage, motor position and motor velocity are measured. The measured voltage is then applied to the open-loop model in simulation and the position and velocity behaviour is compared. The result is shown in Fig. 2.7.

The results show that the motion in the experiments and the open-loop simulation are quite similar. The simulated velocity matches the real one quite closely; the integral thereof, position, shows some deviation over time, yet still matches quite well. Based on these results, we conclude that the model is competent for our goal.

Table 2.2: Properties of the system shown in Fig. 2.5.

Component	Parameter	Value
DC-motor (Maxon Motor, 2015) (Maxon RE25)	Inductance, L_e	$8.32e^{-4}H$
	Electric resistance, R_e	7.73Ω
	Torque constant, K_m	$4.37e^{-2}NA^{-1}$
	Rotor inertia, I_r	$1.05e^{-6}kgm^2$
	Viscous friction R_{vm}	$1.159e^{-6}Nms$
Transmission (Maxon Motor, 2015) (Maxon GP32C)	Gearbox inertia, I_{gb}	$8e^{-8}kgm^2$
	Coulomb friction, τ_{cgb}	$4e^{-2}N$
	Gear ratio, r_{gb}	$\frac{1}{33.0625}$
Load	Load inertia, I_ℓ	$3.6e^{-4}kgm^2$
	Pulley ratio, r_p	$2e^{-2}m$
	Pulley inertia	0
	Gravitational acc., g	$-9.81ms^{-2}$
	Mass, m_ℓ	$0.9028kg$

2.4.3 Energy optimised motor selection

Now, we apply the motor-gearbox selection process described in Section 2.3.3 to select an alternative combination. In the process, gravity is assumed to be zero and this will introduce an error in step 2. Therefore, we have to verify the acceleration capabilities of the drive system after the selection process is completed to make sure that it meets our requirement. Maxon specifies the gearbox efficiency rather than the coulomb friction such that we have to use the adapted equations discussed in Section 2.3.4.

1. The relevant parameters for the system are: $\dot{\omega} = 1.41ms^{-2}$, $I_\ell = 3.6e^{-4}kgm^2$.
2. We assume $\eta_{gb} \geq 0.7$ and assume that I_{gb} is negligible. Eq.2.24 results in: $\frac{\tau_m}{\sqrt{I_r}} \geq 3.2$.
3. We set an additional bound on the motor weight to be less or equal to the weight of the motor we currently use: $m_m \leq 130g$. The motor with the highest steepness that lies within the bounds on m_m and $\frac{\tau_m}{\sqrt{I_r}}$ is the Maxon EC 45 Flat, 50W motor. This motor has the following parameters: $\tau_m = 9,69e^{-2}Nm$, $I_r = 1.35e^{-5}kgm^2$, $S = 0.001525$.
4. Eq. 2.25 results in: $0.023 \leq r_{gb} \leq 0.311$.
5. The gearbox from the maxon GP32C series with the largest transmission ratio within this range has: $r_{gb} = 0.03$, $\eta_{gb} = 0.75$ and $I_{gb} = 0.8e^{-7}kgm^2$.

Figure 2.7: A comparison of the open loop simulated motion with the motion in the experiment.

6. Now we check the assumptions: $\eta_{gb} = 0.75 > 0.7$ and $I_{gb} = 0.8e^{-7} \text{kgm}^2 \ll r_p^2 m_\ell = 0.36e^{-4}$, the assumptions are valid.

The maximum acceleration of the system is evaluated using Eq. 2.29.

$$\dot{\omega} = \frac{\frac{\eta_{gb}}{r_{gb}r_p}\tau_m + m_\ell g}{\frac{\eta_{gb}}{r_{gb}^2 r_p^2}I_r + \frac{1}{r_p^2}I_{gb} + m_\ell} = 3.87 \text{ms}^{-2} > 1.41 \text{ms}^{-2} \quad (2.29)$$

Properties of the load, r_p , m_ℓ and g , can be found in Table 2.2.

2.4.4 Comparison of the energy consumption

Using the model of the setup discussed in Section 2.2 and Section 2.4.1, simulations were done to compare the energy consumption with both motor-gearbox combinations.

The original motor

In the simulations, the electric energy that is supplied to the motor, the energy that is recovered by using the motor as a generator, the energy that is stored in the various storage elements (L , I_r , I_{gb} and I_ℓ in Fig. 2.2) and the energy that is dissipated in the various dissipative elements (R_e , R_{vm} , τ_{gb} in Fig. 2.2) was calculated. The result is shown in Fig. 2.8.

In Fig. 2.8, the supplied energy is the energy that flows from the controller to the setup and the recovered energy is the energy that flows from the setup to the controller. Energy “dissipation” that occurs in the derivative action of

the controller and energy “storage” that occurs in the proportional action of the controller is included in the supplied and recovered energy flows. It is not part of the dissipated and stored energy.

Figure 2.8: An overview of the energy in the system during the simulation. The figure shows supplied, recovered, dissipated and stored energy.

In Fig. 2.8, we see that of the energy that is supplied to the system, only 28% is recovered and 72% is dissipated. To get a better impression on where this energy is dissipated in the system, Fig. 2.9 shows a detailed image of the energy that is dissipated in the various parts of the system.

The results show that the losses occur predominantly in the windings of the motor (66%) the second major source of dissipation is mechanical friction in the gearbox (28%). This confirms the results of (Seok et al., 2015).

The energy optimised choice

As the selected combination selected in Section 2.4.3 has the same gearbox as the original combination, we use the same model and only change the motor parameters. The results are shown in Fig. 2.10.

In Fig. 2.10, we see that of the energy that is supplied to the system, now 60% is recovered and 40% is dissipated and that the dissipated energy is reduced by 50%. This shows that, although energy losses in electric actuators will still be significant for this type of application, considerable improvements can be achieved with a motor-gearbox selection aimed for efficiency.

Note that the newly proposed motor has the same weight as the original motor but a slightly different form factor: the original motor has a diameter of 25mm and a length of 55mm where the new proposition has a diameter of 45mm and a length of 21mm.

Figure 2.9: A detailed image of the dissipation in the various parts of the system. The “Total” dissipated energy is equal to the “Dissipated” energy in Fig. 2.8.

2.5 Conclusion

In Section 2.1, we introduced an incomplete scheme of relations between characteristics of legged locomotion and low energy efficiency. This was followed with a discussion the influence of the selection of a motor-gearbox combination on the energy efficiency of the drive for an application. We presented a method to select a motor-gearbox combination for energy efficiency and tested this method on an example to show the possible impact. Based on our findings, we draw the following conclusions:

1. The inverse motor steepness is the mechanical representation of the electrical resistance and is a measure for the efficiency of the motor. For maximum efficiency, a motor with maximum steepness should be selected.
2. A lower gearbox ratio results in a better drive efficiency.
3. For most robotic applications, electric actuators are the dominant cause of energy losses.
4. Selecting a motor-gearbox combination for energy efficiency can provide a significant improvement in the efficiency. However, the energy losses in the motor-gearbox combination remain significant.

Incorporating the significance of the losses in electric actuators, Fig. 2.1 can be extended to Fig. 2.11.

In a legged locomotion system, the support the relatively high system mass with the legs requires constant torques and the additional movement of the legs

Figure 2.10: An overview of the energy in the system during the simulation. The figure shows supplied, recovered, dissipated and stored energy.

introduces start-stop behaviour in the system. As electric actuators are energy-inefficient in start-stop behaviour and providing constant torques, this lies at the basis of the typically low energy efficiency of legged locomotion systems.

Figure 2.11: A schematic overview of cause and effect relations underlying the low performance in legged locomotion. The orange blobs indicate items that were mentioned in the surveys of (Hardarson, 1997; Seeni et al., 2008), the blue blobs indicate items that were added based on reasoning and the green blobs indicate items that are suspected to have a relation in the scheme and the purple blobs are added based on the results presented in this work. The arrows indicate cause and effect relations that were found in literature.

CHAPTER 3

A NOVEL CONCEPT FOR A TRANSLATIONAL CONTINUOUSLY VARIABLE TRANSMISSION

3.1 Introduction

A Continuously Variable Transmission, or CVT, is a transmission that allows “shifting” of the transmission ratio over a continuous range. When a CVT has a transmission range that includes positive ratios, negative ratios and a zero ratio, it is called an Infinitely Variable Transmission (IVT). Since an IVT is no more than a CVT with a specific property, we only use the term CVT, as it encapsulates the entire class (The range of attainable transmissions is treated as a property of a CVT).

CVT’s have the potential to change robotics. When used in a drive train in combination with a mechanical storage element such as a spring or inertia, they could significantly reduce energy consumption as suggested in (Stramigioli & Dijk, 2008; Stramigioli et al., 2008) and chapters 5 and 6 of this thesis. However, so far this advantage has been shown in theory only.

To apply the concept of a CVT to its full potential in robotics, it is desirable for a CVT concept to have at least the following properties:

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1. The transmission range needs to include positive ratios, negative ratios and a zero ratio.
2. The transmission needs to be reversible with no constraint on the applied forces.
3. Changing the transmission ratio needs to be kinematically decoupled from the transmission such that:
 - (a) Changing the transmission ratio does not directly change the state of the system. In other words, changing the transmission ratio should not perform work on other parts of the system.
 - (b) It is possible to change the transmission ratio when the system is loaded (non-zero forces on “motor” and “load”).

In addition, the CVT needs to be designed such that:

1. The transmission is efficient.
2. Changing the transmission ratio costs little energy.

CVT's have been around for a while and many designs have been proposed (see, for example, (Beachley & Frank, 1979; Ansink, 2008)). In addition, some work on the development of Variable Stiffness Actuators contains a form of CVT (see, for example, (Groothuis et al., 2012; Jafari et al., 2010, 2011; Kim & Song, 2010)).

In this work, we propose a new CVT concept that is aimed to have the properties as listed above. The concept is based on a lever with a movable pivot.

Such a lever could be applied in a Controlled Passive Actuation Scheme (Discussed in detail in chapters 5 and 6) as shown in Fig. 3.1. As we will show in this chapter, the transmission ratio can be changed by moving the pivot along the lever. In Controlled Passive Actuation, this can be used to modulate the energy transfer between the spring and the mass that are shown in Fig. 3.1.

Figure 3.1: A schematic representation of the IVT concept within a Controlled Passive Actuation scheme.

The concept of a lever with movable pivot, has been done before by, for example, (Groothuis et al., 2012; Jafari et al., 2010, 2011; Kim & Song, 2010).

However, the concept that we propose is different in the sense that the pivot always moves along the lever. We present a proof of concept realisation and we test it for the properties above.

In Section 3.2, we provide the a description of the CVT concept. In Section 3.3, the proof of concept realisation is discussed and in Section 3.4 physical properties and experimental results with the proof of concept realisation are used to show that the concept has the required properties. Section 3.5 concludes the chapter.

In this work, we use the following notation for a velocity, v , of a frame a with respect to a frame b , expressed in a frame c : ${}^c v_a^b$.

3.2 The CVT concept

In this section, we describe the CVT concept. By using a set of constraint relations, we show that the concept can meet the desirable properties of a CVT as listed in Section 3.1 in theory and we transform the required properties of a CVT to a set of constraints that a design of a realisation of the concept needs to include.

The CVT concept is a lever with a movable pivot as shown in Fig. 3.2. The pivot point, p_3 , can move along the lever and external forces can be applied in the interaction points, p_1 and p_2 .

Figure 3.2: A schematic representation of the IVT concept that is proposed in this chapter. In the image, both the lever arm and the movable pivot can be identified. In the figure, three points are indicated: points p_1 and p_2 are fixed with lever on different locations: “interaction point 1” and “interaction point 2”, respectively. Point p_3 is fixed to the pivot. External forces, F_1 and F_2 , can be applied on the interactions points.

The forces applied on the interaction points can be decomposed in components co-linear to the lever and components perpendicular to the lever. By moving the pivot, the relative length of the lever arms is changed and, as a result, the transmission ratio between the components of the interaction forces that are perpendicular to the lever. As will be explained later in this chapter, the forces co-linear to the lever are passively opposed. When we write F_i in

this chapter, without further specification, we are referring to the component that is perpendicular to the lever.

Figure 3.3 shows a top view of the concepts with five frames and five distances that will be referred to later in this chapter. Frame ψ_{0a} is chosen to rotate with the lever to simplify the expressions that follow.

Figure 3.3: A schematic representation of the IVT concept, including relevant coordinates. In the figure, we have 5 frames: Frame ψ_0 is a world fixed or inertial frame, frame ψ_{0a} has de same origin as frame ψ_0 , but rotates with the lever. Frames ψ_1 , ψ_2 and ψ_3 are fixed in p_1 , p_2 and p_3 , respectively.

The system as shown in Fig. 3.2 and Fig. 3.3 has two degrees of freedom - rotation around the pivot, θ , and translation of the pivot along the lever, $\dot{\ell}_p$ -, and six constraint relations. In the reasoning below we use relations (3.1)...(3.4) as constraint relations.

$${}^0v_2^0 = {}^0v_1^0 + \begin{pmatrix} -l \sin(\theta) \\ l \cos(\theta) \end{pmatrix} \dot{\theta} \quad (3.1)$$

$${}^1v_3^1 = \begin{pmatrix} \dot{\ell}_a \\ 0 \end{pmatrix} = \begin{pmatrix} \dot{\ell}_p \\ 0 \end{pmatrix} \quad (3.2)$$

$${}^{0a}v_{x1}^{0a} = 0 \quad (3.3)$$

$${}^{0a}v_{y3}^{0a} = 0 \quad (3.4)$$

Relation (3.1) describes that ψ_1 and ψ_2 are connected with a (rigid) beam. Relation (3.2) describes that the velocity of ψ_3 , with respect to ψ_1 (In the x-direction of frame ψ_{0a}), is equal to the velocity of the pivot along the lever ($\dot{\ell}_p$) and that the pivot is fixed to the lever in the y-direction of frame ψ_{0a} . Relation (3.3) describes that the lever can not move in the x-direction of frame ψ_{0a} . Relation (3.4) describes that the pivot can not move in the y-direction of frame ψ_{0a} .

Using relations (3.1)...(3.4) it can be shown that the system has the required properties 1...3 of an CVT as named in Section 3.1:

1. Using relations 3.1...3.4, the transmission ratio between perpendicular components of the velocity of p_1 and p_2 , ${}^{0a}v_1^{0a}$ and ${}^{0a}v_2^{0a}$, respectively can be determined to be:

$${}^{0a}v_2^{0a} = r {}^{0a}v_1^{0a} = \frac{\ell_p - \ell_1}{\ell_p - \ell_2} {}^{0a}v_1^{0a} \quad (3.5)$$

For $0 \leq \ell_p < \ell_2$, the transmission range spans $-\frac{\ell_1}{\ell_2} \leq r < \infty$ in a continuous way.

2. Dual to this, the relation between F_2 when applying a force F_1 is:

$$F_2 = r^{-1} F_1 = \frac{\ell_p - \ell_2}{\ell_p - \ell_1} F_1 \quad (3.6)$$

The relation for F_1 when applying a force F_2 is:

$$F_1 = r F_2 = \frac{\ell_p - \ell_1}{\ell_p - \ell_2} F_2 \quad (3.7)$$

In both situations, there is no constraint on the applied forces.

3. There is no kinematic relation between v_1 and v_2 , and $\dot{\ell}_p$.

When the design has constraints (3.1)...(3.4), it will realise properties 1...3 as named in Section 3.1.

3.3 Description of the setup

To experimentally verify the new CVT concept, a mechanical implementation that satisfies constraints 3.1...3.4 was made. In this section, we describe the design and realisation of the implementation.

3.3.1 Concept design

The design of the test setup is shown schematically in Fig. 3.4. It consist of a lever with a movable pivot where the pivot is constrained to move along the lever.

Since the arm of the lever is rigid, constraint 3.1 is met. The constraint for the pivot to move along the lever is equal to constraint 3.2. To provide constraints 3.3 and 3.4, conceptually two wheels are stacked as shown in Fig. 3.4.

Constraint 3.4 is created by the idealised motion of a completely rigid wheel with a pure contact point. A wheel has a central axis and a symmetry around that axis. If the wheel is touching a plane and purely rolling on that plane,

Figure 3.4: An illustration of the CVT concept that is proposed in this work. The image shows two views of a lever with a movable pivot. Fig. 3.4a shows how a pivot in the x-y plane is created with two wheels. With this construction, we implement constraints (3.3) and (3.4) (Fig. 3.4b).

the relative motion can be described as being constrained by equation 3.8. I.e. the velocity of the wheel is constrained to be in the direction in which it is oriented, in this case the orientation of the lever. This means that velocity perpendicular to the lever is constrained to 0 in point 3.

$${}^{0a}v_y^{0a} = 0 \quad (3.8)$$

Constraint 3.3 is created by the stacked wheels. Take the velocities as defined in Fig. 3.5.

Figure 3.5: Relevant velocities in the explanation of the constraint that is created by conceptually stacking two wheels. In the illustration, the lever “stick” is shown with a movable pivot that consists of a cage and a set of stacked wheels. Rotational velocities of the wheels and the velocity of the pivot, along the lever, relative to the lever and relative to frame 0a are indicated.

The constraint is created as follows. From the design we have the relations

3.9, 3.10 and 3.11.

$$\dot{\ell}_a = \omega_1 r_1 \quad (3.9)$$

$$\omega_1 = \frac{r_2}{r_1} \omega_2 \quad (3.10)$$

$$\omega_2 = \frac{{}^{0_a}v_{x3}^{0_a}}{r_2} \quad (3.11)$$

Where r_1 is the radius of the upper wheel and r_2 the radius of the lower wheel. Combining these, results in Eq. 3.12.

$$\dot{\ell}_a = r_1 \frac{r_2}{r_1} \frac{{}^{0_a}v_{x3}^{0_a}}{r_2} = {}^{0_a}v_{x3}^{0_a} \quad (3.12)$$

Combining Eq. 3.12 with Eq. 3.2 gives:

$$\dot{\ell}_a = {}^1v_{x3}^1 = {}^{0_a}v_{x3}^{0_a} - {}^{0_a}v_{x1}^{0_a} = {}^{0_a}v_{x3}^{0_a} \quad (3.13)$$

Such that:

$${}^{0_a}v_{x1}^{0_a} = 0 \quad (3.14)$$

which is the constraint that is required: the velocity of the lever is constrained to be zero in the direction co-linear to the lever and the (components of) forces co-linear to the lever are passively opposed. (for a reference to the locations of the frames, please refer to Fig. 3.3).

For a realisation, rotation in the x-z plane of frame ψ_{0a} also needs to be constrained. This can effectively be done by supporting the ends of the lever “stick” as is shown in Fig. 3.6.

Figure 3.6: Supports can effectively be used to constrain rotation in the x-z plane.

3.3.2 Realisation

Based on the concept design as presented in section 3.3.1, a setup has been designed and realised. Fig. 3.7a shows a sketch of the gears and wheels in the transmission that is used to obtain the effect of the double wheel as discussed in section 3.3.1. The following components are shown:

1. Two wheels are used for stabilisation. (Blue)
2. The differential to enable rotation of the pivot (Purple)
3. The rack and pinion that is the contact with the lever (Red)
4. An extra set of gears to make the ratio $\frac{{}^{0a}v_{x3}{}^{0a} - {}^{0a}v_{x1}{}^{0a}}{{}^{0a}v_{x3}{}^{0a}}$ equal to 1. (Green)
5. A DC-motor with gearbox and position sensor to position the pivot (Orange)

Fig. 3.7b shows a picture of the internals of the pivot, the colours indicate the correspondences with the stages as shown in Fig. 3.7a.

(a) A schematic drawing of the transmission that is used to create the effect of the stacked wheels.

(b) A detailed picture of the internals of the pivot.

Figure 3.7: A photo and drawing of the internals of the pivot, the colours indicate the corresponding components. A transmission is used to create the effect of the stacked wheels as discussed in section 3.3.1.

The complete setup is shown in Fig. 3.8.

The physical properties of the setup have been characterised and are shown in Table 3.1. The friction that is experienced while moving the pivot is highly position dependent and non-linear. Fig. 3.9 shows the average relation between velocity and friction force, with standard deviations for several measurement points.

3.4 Evaluation of properties and design criteria

Using the properties of the setup and experimental results, the setup was evaluated for the properties and critical design points as presented in Section 3.1. In this section we present and discuss the results for each property and critical design point.

Figure 3.8: Pictures of the setup that was realised to evaluate the CVT concept. The following components are shown: The linear guides that form the lever, supports for stabilising the lever in the (x,y)-z plane and the pivot.

Figure 3.9: The friction force that is experienced while moving the pivot along the lever. The figure shows the means and standard deviations for several measurement points.

3.4.1 Property 1: “The transmission range needs to include positive ratios, negative ratios and a zero ratio”

The transmission ratio, r , is equal to:

$$r = \frac{\ell_p - \ell_1}{\ell_p - \ell_2} \quad (3.15)$$

When choosing that $\ell_2 > \ell_1$, the transmission range is as shown in Table 3.2.

When furthermore assuming that $\ell_1 = 0.5\ell_2$, the transmission ratio, r , can be plotted as a function of the pivot position ℓ_p as shown in Fig. 3.10.

This shows that the transmission range spans a continuous range for $-\infty < r < \frac{\ell_1}{\ell_2}$. To verify if the setup indeed satisfies this, the transmission ratio was

Property	Value	Unit
$\Delta \ell_p$	$0 \dots 36.5$	<i>cm</i>
$\Delta \ell_1$	$0 \dots 41$	<i>cm</i>
$\Delta \ell_2$	$0 \dots 41$	<i>cm</i>
Pivot friction	See Fig. 3.9	
Eff. pivot mass	4.5	<i>kg</i>
Lever friction		
- Viscous	0.1	$Nmsrad^{-1}$
- Coulomb	0.4	Nm
Lever inertia	$0.0515 + 1.21l_p^2$	kgm^2
Max. force on pivot	16	N
DC motor + Gearbox		
- Max. cont. torque	363.6	mNm
- Max. peak. torque	1118.3	mNm
- Max. rot. velocity	19	$rads^{-1}$

Table 3.1: Properties of the experimental setup

Transmission ratio	pivot position
$r = \frac{\ell_1}{\ell_2}$	$\ell_p = 0$
$\frac{\ell_1}{\ell_2} > r > 0$	$0 < \ell_p < \ell_1$
$r = 0$	$\ell_p = \ell_1$
$r < 0$	$\ell_1 < \ell_p < \ell_2$
$r = -\infty$	$\ell_p = \ell_2$
$r > 0$	$\ell_p > \ell_2$

Table 3.2: The theoretical transmission range of the CVT.

Figure 3.10: The theoretical transmission range of the CVT when assuming $\ell_1 = 0.5\ell_2$.

measured by recording the movement of the lever with the pivot at several key

positions: $\ell_p = 0, 0.5\ell_1, \ell_1, \frac{\ell_1+\ell_2}{\ell_2}$ and ℓ_2 . The results of this measurement is shown in Fig. 3.11.

Figure 3.11: The measurement results measurements of the transmission ratio for several key pivot positions. From left to right: $\ell_p = 0, 0.5\ell_1, \ell_1, \frac{\ell_1+\ell_2}{\ell_2}$ and ℓ_2 . In the figure, the movement of the two interface points ℓ_1 and ℓ_2 of the lever is shown in purple, the filtered transmission ratio measurement in light orange with its mean in dark orange. The black lines show the theoretical values. For the most right result, the transmission ratio is $-\infty$ and thus not shown in the figure.

As expected, the figure shows that the range spans a range as shown in Table 3.2. It can also be seen that the measured transmission ratio deviates from the theoretical values in some cases. Possible reasons for this are positioning inaccuracy and sensor noise.

3.4.2 Property 2: “Power transmission is possible both when applying a constant force on the output and when applying a constant force on the input”

In Section 3.4.1 we have shown that power transmission is achieved for direct transmission, for all direct transmission ratios except two. For pivot positions $\ell_p = \ell_1$ ($\ell_a = 0$) and $\ell_p = \ell_2$ ($\frac{\ell}{\ell_a} = 1$), no power transmission is achieved. Now we test the power transmission for inverse transmission.

The inverse transmission ratio is:

$$r^{-1} = 1 - \frac{\ell}{\ell_a} \quad (3.16)$$

It is theoretically back-drivable when applying a constant force to the output when $\frac{\ell}{\ell_a} \neq 1$ and $\ell_a \neq 0$. As with the direct transmission ratio, this has been

verified with experiments. The inverse transmission ratio was measured for the same key positions ($\ell_p = 0, 0.5\ell_1, \ell_1, \frac{\ell_1+\ell_2}{\ell_2}$ and ℓ_2), the result is shown in Fig. 3.12.

Figure 3.12: The measurement results measurements of the inverse transmission ratio for several key pivot positions. From left to right: $\ell_p = 0, 0.5\ell_1, \ell_1, \frac{\ell_1+\ell_2}{\ell_2}$ and ℓ_2 . In the figure, the movement of the two interface points ℓ_1 and ℓ_2 of the lever is shown in purple, the filtered inverse transmission ratio measurement in light orange with its mean in dark orange. The black lines show the theoretical values. For the middle result, the transmission ratio is $-\infty$ and thus not shown in the figure.

As expected, the figure shows that the transmission is back drivable for all transmission ratios except two. For pivot positions $\ell_p = \ell_1$ ($\ell_a = 0$) and $\ell_p = \ell_2$ ($\frac{\ell}{\ell_a} = 1$), no power transmission is achieved as expected.

3.4.3 Property 3: “Changing the transmission ratio needs to be kinematically decoupled from the transmission.”

As it is also defined in two parts, the evaluation of this property is also discussed in two parts, namely “Changing the transmission ratio does not directly change the state of the system” and “It is possible to change the transmission ratio when the system is loaded”.

Changing the transmission ratio does not directly change the state of the system.

To validate this property, the velocity of the pivot p_{pivot} and of the interface points p_1 and p_2 is measured while alternating 1) moving the pivot along the

lever and 2) exciting the lever in a periodic manner. To confirm this property, the pivot should have zero velocity when exciting the lever and the interface points should have zero velocity when moving the pivot. The results are shown in Fig. 3.13.

Figure 3.13: The velocities of the pivot p_{pivot} and of the interface points p_1 and p_2 while alternating 1) moving the pivot along the lever and 2) exciting the lever in a periodic manner.

The measurements show that for all motion but between $17s$ and $20s$, the velocity of the pivot is zero when exciting the lever and the velocity of the interaction points of the lever is zero when moving the pivot.

The zoom on the period from $17s$ to $20s$ shows an irregularity of the velocity of the pivot. During this irregularity, the velocity of the interaction points suddenly increases unexpectedly. This behaviour is expected when one of the wheels slips.

It is possible to change the transmission ratio when the system is loaded

This property is validated by measuring and comparing the motor in two situations:

1. The lever is not loaded
2. The lever is loaded with a force of $10N$ collinear to the lever. This is a significant force within the working range of the lever (max. $16N$ on the pivot, see Table 3.1.)

This experiment is done for moving the pivot from $\ell_p = 0$ to $\ell_p = \ell_2$ and for moving the pivot from $\ell_p = \ell_2$ to $\ell_p = 0$. The results of the measurements are shown in Fig. 3.14.

(a) Moving the pivot from $l_p = 0$ to $l_p = l_2$. (b) Moving the pivot from $l_p = l_2$ to $l_p = 0$.

Figure 3.14: Current consumption while loaded and while not loaded during movement of the pivot. The lighter lines show the measured current and the darker lines show the average.

The measurements show that there is an increase in the magnitude of the current both when moving the pivot from $l_p = 0$ to $l_p = l_2$ and when moving the pivot from $l_p = l_2$ to $l_p = 0$. In an ideal transmission we would not expect this but in reality, the transmission is loaded by the external force and for this reason, the motor current is expected to be higher in the loaded condition due to increased friction in the transmission.

However, if the external load would load the reconfiguration actuator, the magnitude of the current would increase for moving the pivot from $l_p = l_2$ to $l_p = 0$ but decrease when $l_p = 0$ to $l_p = l_2$. In the first situation, the external load opposes the actuation action while it helps the actuation action in the second situation.

Since we see an increase in the magnitude for both situations, we conclude that the external load does not load the reconfiguration actuator. The increase in the required current in the loaded is only due to increased friction in the

transmission.

3.4.4 Critical design point 1: Efficiency of the transmission

Now that the required properties of the CVT are verified, we continue with evaluating the critical design points: the efficiency of the transmission and the energy cost of moving the pivot.

Based on the parameters for the lever friction as given in Table 3.1, the efficiency for a certain rotational velocity of the lever around the pivot, $\dot{\theta}$ and torque on the pivot τ can be calculated. The friction force can be expressed as:

$$\tau_f = R_v \dot{\theta} + \text{sgn}(\dot{\theta}) \tau_c \quad (3.17)$$

where, R_v is the viscous friction and τ_c is the coulomb friction torque. Both can be found in Table 3.1. The efficiency, η , can be expressed as:

$$\eta = \frac{\tau - \tau_f}{\tau} = \frac{\tau - (R_v \dot{\theta} + \text{sgn}(\dot{\theta}) \tau_c)}{\tau} \quad (3.18)$$

In Fig. 3.15, the efficiency of the transmission is shown for two rotational velocities and a range of torques is shown.

Figure 3.15: The efficiency of the lever for two velocities and the allowable range of input forces.

Note that when expressing the transmission ratio as a function of external forces F_1 and F_2 , the efficiency is also dependent on the transmission ratio as the relation between F_1 , F_2 and τ changes with the transmission ratio.

3.4.5 Critical design point 2: Energy cost of moving the pivot

By multiplying the friction force with the velocity in Fig. 3.9, the power lost in friction is calculated. This results in Fig. 3.16. The figure shows the power

that is required to move the pivot at a certain velocity.

Figure 3.16: The power that is required to move the pivot with a certain velocity.

3.5 Conclusions & Future work

In this work, we started with a set of properties and critical design points for CVTs. From that, we moved to the novel concept of a variable lever arm CVT where the pivot always moves along the lever. We showed that this concept has the required properties in theory. We briefly discussed a realisation of this concept which has been evaluated on the properties and critical design points. From this evaluation, we see that it indeed has the required properties. We expect that the performance on the critical design points can be further improved by improving the mechanical realisation. In order to apply this concept successfully, the maximum force on the pivot (See Table 3.1) needs to be increased by either increasing the normal force or using different materials for creating the friction contact between the wheel and the surface on which the wheel rolls. Improving the mechanical realisation and in this, increasing the maximum force on the pivot and efficiency, is a topic for future work.

CHAPTER 4

POWER SPLIT BASED DUAL HEMISPHERICAL CONTINUOUSLY VARIABLE TRANSMISSION

4.1 Introduction

A Continuously Variable Transmission, or CVT, is a transmission that allows continuous “shifting” of the transmission ratio over a continuous range. CVT’s have been around for a while and many designs have been proposed (see, for example, (Beachley & Frank, 1979; Ansink, 2008; Lahr, 2009)). In addition, some works on the development of Variable Stiffness Actuators contain a form of infinitely or continuously variable transmissions (see, for example, (Groothuis et al., 2012; Jafari et al., 2010, 2011; Kim & Song, 2010)). Roughly speaking, mechanical CVT’s can be divided into four categories: Traction based CVT’s, Belt CVT’s, Ratcheting CVT’s and Variable lever arm CVT’s. In (Beachley & Frank, 1979) and (Lahr, 2009), an overview/ review is discussed of several existing types of CVT’s.

In this work, we propose a new traction-based CVT concept, the Dual-Hemi CVT. Traction based CVT’s use a rolling frictional contact to transmit power. For many of these designs, the action that changes the transmission ratio requires to push through the same frictional contact that is also used to transmit power. Thus, changing the transmission ratio requires a high amount of work.

This chapter is based on Dresscher, D., Naves, M., de Vries, T. J. A., Buijze, M., & Stramigioli, S. Power split based dual hemispherical continuously variable transmission, Manuscript submitted for publication

There are some interesting examples where this is not required, the most notable being (Peshkin et al., 2001) and (Everarts et al., 2015). In most traction-based CVT's, changing the transmission ratio is kinematically decoupled from the transmission and energy can be transferred in both directions. However, most traction based CVT designs have a transmission ratio range that spans a positive (or negative) range only (also here, (Peshkin et al., 2001) is an exception). However, the transmission ratio range can be extended using the power-split principle (Beachley & Frank, 1979; Bottiglione & Mantriota, 2011), where a differential or a planetary gear is used to extend the range of ratios, introducing extra losses and possibly consequences for the back-drivability (Bottiglione & Mantriota, 2011).

To apply the concept of a CVT to its full potential in robotics - for example as described in (Stramigioli et al., 2008), (Everarts et al., 2012) and chapters 5 and 6 of this thesis - , we believe that a CVT should have at least the following properties:

1. To use the CVT to transfer energy both from the source to the load and vice versa, as well as the option not to transfer energy, the transmission range needs to include positive ratios, negative ratios and a zero or infinite ratio.
2. Building on point 1, the system needs to be back-drivable under all conditions.
3. For the actuation energy not to be delivered by the reconfiguration actuator, changing the transmission ratio needs to be kinematically decoupled from the transmission such that:
 - (a) Changing the transmission ratio should not perform work on other parts of the system.
 - (b) It is possible to change the transmission ratio when the system is loaded (non-zero forces on "Energy source" and "Load").

In addition, to achieve a high level of energy efficiency, the CVT needs to be designed such that:

4. The transmission has high efficiency.
5. Changing the transmission ratio requires little work.

The Dual-Hemi CVT is aimed to have these properties.

In Section 4.2, we give some background on the power-split principle. Section 4.3 and 4.4 cover the concept and important design details, respectively, of the Dual-Hemi CVT. The prototype realisation is discussed in Section 4.5 and in Section 4.6 important properties of the prototype are presented. This is followed by a discussion and exploration on two design choices in Section 4.7. Section 4.8 concludes the work with the most important findings.

4.2 Power Split based CVT's

The first requirement set to use a continuous variable transmission to its full potential in robotic applications, is a range of ratios that includes the positive ratios, negative ratios and zero or infinite ratio.

When mechanical CVT's are considered, this spread in ratios including the positive and negative ratios inherently requires a transition region where input and output are unrelated (a zero or infinite ratio). For traction based CVT's, this region typically requires a sliding contact as rotational velocity of input and output are decoupled for this ratio. As an example, the CVT design given by Peshkin can be considered (Peshkin et al., 2001). As this sliding contact in the transition region typically causes high levels of wear, friction and energy losses, such a system is impractical in many applications. In order to prevent the necessity of this unfavorable transition region, while obtaining a mechanical system with the required range of ratios, the power split principle can be used.

Power-split based continuously variable transmissions (PS-CVT) are typically built with a regular continuously variable transmission (CVT), a planetary gear system with differential properties (PG) and optionally a mechanical transmission with a fixed ratio (FR). A schematic overview of a power-split based CVT is given in figure 4.1.

Figure 4.1: Schematic overview of a PS-CVT

By using the power split principle, a CVT can be used which allows only a single sided (for example positive) transmission ratio for constructing a system with a ratio range that also includes the negative ratios and a zero or infinite ratio.

When applying the power-split principle, care should be taken regarding

the extra losses introduced by the extra gears and the effect that this may have on the reversibility (Bottiglione & Mantriota, 2011).

4.3 dual-hemispherical CVT concept design

In this section an innovative concept for a continuously variable transmission (CVT in Fig. 4.1) named dual hemispherical CVT (DH-CVT) is described². This concept exchanges power between input and output by a pure rolling contact of two spherical elements resulting in a high operating efficiency. Furthermore, force required to adjust the transmission ratio does not require to break through a frictional contact or preload, enabling an energy efficient change of transmission ratios.

By adjusting the angle at which both spheres are aligned, the transmission ratio can be alternated as the effective distance between the frictional contact and the axis of rotation of each sphere changes, resulting in a change in transmission ratio. As the alignment can be adjusted by a pure rolling motion of both hemi-spherical elements, a change in transmission ratio can be realised with minimal adjustment force. To enable this change in transmission ratio by a pure rolling motion, the intermediate distance between the center of each friction wheel is equal to the sum of the radii of both friction wheels over the entire configuration range. A schematic overview of this concept is given in figure 4.2a, where the red spherical element is held in a stationary configuration, while the blue spherical element “rolls” over the red one in order to change the effective transmission ratio.

Changing the transmission ratio is kinematically decoupled from the transmission, as is shown in Fig. 4.2b. In this image, changing the transmission ratio is a rotation around the x axis while the transmission works with a translation along the x-axis. Due to this property, changing the transmission ratio does not directly change the state of the system and it is possible to change the transmission ratio when the system is loaded.

Comparing the Dual-Hemi CVT with the most notable alternative CVT principles as described in (Peshkin et al., 2001) and (Everarts et al., 2015) shows that, in their working principle, they all use the principle of changing an effective radius in a distinctively different way. In (Peshkin et al., 2001) a sphere is used of which the motion is constrained by several rollers. By the configuration of the rollers, the sphere is constrained to roll around an axis and by rotating two of the rollers, the constraints and thus the axis of rotation can be changed. Dependent on the axis of rotation, various effective radii are created with respect to the two driving rollers - thus changing the transmission ratio. In (Everarts et al., 2015), the CVT is created by moving a ring gear over

²The initial idea of the Dual-Hemi CVT is by Peter Rutgers, DEMCON, Enschede, the Netherlands.

Figure 4.2: A conceptual drawing of the Dual-Hemi CVT. To change the transmission ratio, the blue hemisphere rolls over the red one around the x-axis of the frame ψ_1 in the point of contact (black dot in the figure).

conic planet gears such that the effective radius of the planets is changed - thus changing the transmission ratio.

4.3.1 Dual-Hemi CVT Transmission ratio

The transmission ratio of the DH-CVT depends on the contact angle between both hemispheres. This ratio is equal to the ratio between the effective radius R'_1 and R'_2 as illustrated in figure 4.3. This effective radius is defined by the distance between the axis of rotation and the contact point of each spherical element. The radius of each sphere is set equal to minimize local contact stress at the contact region (Hertz, 1882) ($R_1 = R_2$) and furthermore the spheres are configured such that at a configuration angle ϕ of 45 degrees, the transmission ratio between input and output equals one.

The resulting transmission ratio r_{cvt} as function of the configuration angle ϕ is then given by the geometrical tangent of the configuration angle ϕ according to

$$r_{cvt} = \frac{\omega_4}{\omega_3}, \quad r_{cvt} = -\frac{R'_1}{R'_2} = -\frac{1}{\tan(\phi)}. \quad (4.1)$$

If spherical friction wheels are considered, the configuration angle could theoretically be set over a range of 0 to 2π radians, resulting in a range of transmission ratio's from $-\infty \leq r_{cvt} \leq \infty$, see figure 4.4.

However, as discussed in Section 4.2, a transmission ratio of $-\infty$ at $\phi = 0 + k\pi$ rad or 0 at $\phi = \frac{\pi}{2} + k\pi$ rad which decouples the output from the input will result in a sliding contact between both friction wheels (input is

Figure 4.3: The effective radii of the Dual-Hemi CVT.

fixed, output is free to rotate or vice versa). As this sliding contact will reduce efficiency and create high levels of wear, this configuration should be avoided. A configuration angle of $0 < \phi < \frac{\pi}{2}$ with the corresponding transmission ratio of $-\infty < r_{cvt} < 0$ is considered where the allowable range of configuration angles should be sufficiently distant from 0 and $\frac{\pi}{2}$ rad in order to prevent slip. This will be discussed in more depth in Section 4.4.2.

4.3.2 PS-CVT transmission ratio

To enable the required bi-direction powerflow and the corresponding transmission ratio range including positive and negative ratios, the power split principle is used as described in section 4.2. The transmission ratio of the power-split CVT (PS-CVT) is directly related to the transmission ratio of the CVT, PG and FR as illustrated in figure 4.1. The transmission ratio of the CVT (r_{cvt}), PS-CVT (r_{ps}) and the remainder gears with constant ratio (r_{fr}) are related to the angular velocity of each component (ω_1 to ω_7) identified by the numbers 1

Figure 4.4: The complete transmission ratio range of the DH-CVT

to 7 from figure 4.1.

$$r_{ps} = \frac{\omega_7}{\omega_1} \quad (4.2)$$

$$r_{cvt} = \frac{\omega_4}{\omega_3} = -\frac{1}{\tan(\phi)} \quad (4.3)$$

$$-r_{fr} = \frac{\omega_3}{\omega_2} = \frac{\omega_5}{\omega_4} \quad (4.4)$$

Moreover, the kinematic relation between the rotation velocity of the sun gear (ω_s), carrier (ω_c) and the annulus gear (ω_a) of the planetary gear system is given by (Bottiglione & Mantriota, 2011):

$$\omega_s + \omega_a \mathcal{X} = \omega_c(1 + \mathcal{X}) \quad (4.5)$$

with $\mathcal{X} = \frac{R_a}{R_s}$ where R_a is the radius of the annulus gear and R_s the radius of the sun gear. In order to obtain the widest transmission ratio range including positive and negative ratios, the carrier is connected to component 6, the annulus to component 2 and, consequently, the sun to 1. This results in a kinematic relation for the planetary gear system according to

$$\omega_1 + \omega_2 \mathcal{X} = \omega_6(1 + \mathcal{X}). \quad (4.6)$$

Finally, from equations 4.2, 4.3, 4.4, and 4.6 and by considering $\omega_5 = -\omega_6 = \omega_7$, the transmission ratio of the PS-CVT is given by:

$$r_{ps} = \frac{r_{fr}^2}{-r_{fr}^2(1 + \mathcal{X}) + \mathcal{X} \tan(\phi)}. \quad (4.7)$$

When typically planetary gear systems are considered with $2 < \mathcal{X} < 6$ and

a ratio of $r_{fr} = -1$ is selected, this equation reduces to

$$r_{ps} = \frac{1}{-(1 + \mathcal{X}) + \mathcal{X} \tan(\phi)}. \quad (4.8)$$

The resulting transmission ratios for typical values of $\mathcal{X}$ are given in figure 4.5.

Figure 4.5: Transmission ratio of PS-CVT for typical values of $\mathcal{X}$, the planetary gear ratio.

Following from figure 4.5 the transmission allows a bi-direction powerflow and the corresponding required positive and negative transmission ratios for properly chosen values of r_{fr} and $\mathcal{X}$ when the configuration angle is set over a range $0 < \phi < \frac{\pi}{2} rad$.

4.4 Dual-hemi CVT design details

In order to experimentally validate the characteristics of the power split based dual hemispherical continuously variable transmission, a prototype has been built. To obtain a practical volume, the maximum allowable radius of the hemispherical friction wheels is set at $60mm$. The ratios for the planetary gear ($\mathcal{X}$) and fixed ratio transmissions (r_{fr}) are shown in Table 4.1. The resulting range of transmission ratios according to equation 4.7 is given in figure 4.6.

Table 4.1: Planetary gear and fixed ratio transmission ratios

Parameter	Value
$\mathcal{X}$	4
r_{fr}	-26/27

Figure 4.6: Transmission ratio of PS-CVT and DH-CVT

A schematic image of the PS-CVT is shown in Fig. 4.7.

Figure 4.7: A conceptual drawing of the Dual-Hemi CVT (left) in combination with a planetary gear (right). The fixed ratio transmissions are not shown.

4.4.1 PS-CVT torque distribution

In order to determine the maximum allowable output torque of the PS-CVT, the internal torque distribution (τ_1 to τ_7) is of interest. The ratio of torques belonging to the planetary gear system are directly related to its geometric relations and can be calculated as the dual of Eq. 4.6, which results in Eq. 4.9 and Eq. 4.10.

$$\frac{\tau_2}{\tau_1} = \mathcal{X} \quad (4.9)$$

$$\frac{\tau_6}{\tau_1} = \mathcal{X} + 1 \quad (4.10)$$

Furthermore, the ratio of torques for the CVT is dependent on the configured transmission ratio and can be calculated as the dual of 4.3, which results in Eq. 4.11.

$$\frac{\tau_4}{\tau_3} = -\tan(\phi) \quad (4.11)$$

The torque ratio of the remainder gear with constant transmission ratio can be calculated as the dual of Eq. 4.4 which results in Eq. 4.12.

$$\frac{\tau_3}{\tau_2} = \frac{\tau_5}{\tau_4} = -\frac{1}{r_{fr}} \quad (4.12)$$

By combining equation 4.9 to 4.12, the internal torque distribution proportionally to the input torque (τ_1) can be determined according to

$$\tau_2 = \mathcal{X}\tau_1 \quad (4.13)$$

$$\tau_3 = -\frac{\mathcal{X}}{r_{fr}}\tau_1 \quad (4.14)$$

$$\tau_4 = \tan(\phi)\frac{\mathcal{X}}{r_{fr}}\tau_1 \quad (4.15)$$

$$\tau_5 = -\tan(\phi)\frac{\mathcal{X}}{r_{fr}^2}\tau_1 \quad (4.16)$$

$$\tau_6 = (\mathcal{X} + 1)\tau_1 \quad (4.17)$$

As the resulting output torque τ_7 is the resultant from the torque τ_5 and τ_6 , the output torque is given by:

$$\tau_7 = -(\tau_5 + \tau_6) = \frac{\mathcal{X}\tan\phi - r_{fr}^2(\mathcal{X} + 1)}{r_{fr}^2}\tau_1 = \frac{1}{r_{ps}}\tau_1 \quad (4.18)$$

For designing the DH-CVT, the transmitted torque through the CVT (τ_3 or τ_4) is of main interest. According to equation 4.14, the torque which passes through the CVT is directly related to $\mathcal{X}$ and r_{fr} and the provided input torque τ_1 .

Furthermore, the internal torque contribution of τ_3 , τ_4 and τ_7 is then given according to equations 4.13...4.17 which varies with configuration angle and input torque. The primary torques of interest over the configuration range are given in figure 4.8

Figure 4.8: Internal torques: τ_3 , τ_4 and τ_7

4.4.2 Design of the friction wheels

To design the friction wheels, allowable ranges of transmission ratios and input torques that can be used without failure/slip of the CVT have to be determined. To do this, we assume that the output is fixed ($\omega_7 = 0$) such that the blue sphere is fixed. The force that the friction patch needs to hold, F_w , is the force that is exerted on the blue sphere by the red sphere. This force is equal to the torque exerted on the red sphere, τ_3 divided by the effective radius of the red sphere, $R'_1 = R_1 \sin(\phi)$, and is equal to:

$$F_w \geq \frac{\tau_3}{R_1 \sin(\phi)} \quad (4.19)$$

which results in

$$F_w \geq \frac{\mathcal{X}\tau_1}{R_1 \sin(\phi)r_{fr}} \quad (4.20)$$

when combined with equation 4.14.

The required preload in order to obtain the required friction contact is then given by

$$F_n = \frac{F_w}{\mu} \geq \frac{\mathcal{X}\tau_1}{R_1 \sin(\phi)r_{fr}\mu}, \quad (4.21)$$

For practical reasons, hardened steel (100Cr6) is used as material. For a typical uncontaminated steel to steel contact, μ equals to about 0.2. Furthermore, the maximum allowable preload F_n is bounded by the maximum allowable contact stress in order to prevent failure in the contact region. The Hertzian contact pressure between two spherical elements is given by (Hertz, 1882):

$$P_{max} = \sqrt[3]{\frac{3F_n E^{*2}}{\pi^3 R^{*2}}} \quad (4.22)$$

with

$$R^* = \frac{R_1}{2} \quad \text{and} \quad E^* = \frac{E}{2(1 - \nu^2)}. \quad (4.23)$$

where R represents the sphere radius and E and ν the E-modulus respectively the Poisson ratio of 100Cr6 ($E = 210\text{GPa}$, $\nu = 0.3$). P_{max} is the maximum allowable contact stress which we limited to 1.5 GPa.

When combining equation 4.21 and 4.22, the maximum allowable input torque τ_{1max} is directly related to configuration angle ϕ according to

$$\tau_{1max}(\phi) = \frac{P_{max}^3 \pi^3 R_1^3 \mu r_{fr}}{12 \mathcal{X} E^{*2}} \sin \phi \quad (4.24)$$

Given $\mathcal{X}$ and r_{fr} according to Table 4.1, the resulting maximum allowable input torque versus configuration angle is illustrated in figure 4.9.

Figure 4.9: Maximum allowable input torque τ_1

As shown in figure 4.9, slip is inevitable at a configuration of $\phi = 0\text{rad}$, as this would require an infinite preload or a zero input torque. In order to prevent slip, the configuration angle should always remain sufficiently distant from $\phi = 0\text{rad}$.

Based on this figure, a minimum configuration angle of $\phi = 0.35\text{rad}$ is selected, resulting in an maximum allowable input torque of 2 Nm. In order to obtain a symmetric range of transmission ratios, a range of angles from 0.35rad to 1.17rad is selected.

The resulting maximum attainable output torque, for a considered input torque of 2 Nm, is given in figure 4.10 and can be determined by equation 4.18, resulting in an output torque of $-7 < \tau_7 < 7$ Nm and the corresponding range of transmission ratios of $-3.5 < r < 3.5$.

4.5 Realisation

4.5.1 Drive train

mechanism, the two hemispheres can be found. The bearing system of the blue hemisphere is suspended and preloaded by disc-springs in order to obtain the required preloading force F_n between both hemispheres. As this preload will

induce small deformations of the hemispheres, face-gears are used to prevent miss-alignments of the gears. These face-gears are connected to shafts that run exactly through the center of the spheres. For the red hemisphere, the shaft is connected to the ring gear of the planetary gear, for the blue hemisphere the shaft is connected to “gear 1”. The pitch diameter of “gear 1” is set equal to the diameter of the the hemispheres, as is the pitch diameter of “gear 2”, which is connected to the planet cage of the planetary gear. Doing so ensures that “gear 1” can roll over “gear 2” as the blue hemisphere rolls over the red one to change the transmission ratio. The “output” shaft is connected to “gear 3”, which has the same diameter as “gear 2”. The “input” shaft is connected to the sun gear of the planetary gear.

4.5.2 Reconfiguration mechanism

The reconfiguration mechanism is shown in Fig. 4.12. By using the “reconfig-

Figure 4.12: An image showing the drive train and reconfiguration mechanism of the dual-hemi CVT. With respect to Fig. 4.11, the image includes the reconfiguration “motor”, reconfiguration “lever” and reconfiguration “gears”.

uration actuator”, the “lever” is rotated around point “A” with a timing belt. The “lever” has a length of twice the radius of the hemispheres. Referring to Fig. 4.3, it corresponds to the dash-dotted line between the centers of the red hemisphere and the blue sphere. The “alignment gears” have a pitch radius with is equal to the radius of the spheres and are an extra measure to take care that the blue hemisphere rolls neatly over the red hemisphere when the “lever” is adjusted.

In order to reduce the required work for the reconfiguration mechanism, a gravity compensator (not shown in the figure) is added to compensate for the gravitational forces acting on the reconfiguration mechanism.

Fig. 6.4 shows a picture of the setup.

Figure 4.13: A photo of the Dual-Hemi CVT. Photo by Wanda Tuerlinckx.

4.6 Characterisation of the setup

In this section, we discuss the characterisation of some important parameters of the system. We start with discussing the transmission system based on a simplified lumped parameter model of the systems where each lumped parameter represents a quantity that will be characterised. Then, we discuss the properties of the reconfiguration mechanism based on which we can say something about the achievable bandwidth.

4.6.1 The transmission system

Fig. 4.14 shows a lumped IPM of the transmission system.

In this model, the total inertia of the system is lumped in three elements $I_1 = 0.24gm^2$, $I_{2,3} = 2.44gm^2$, $I_{4,5,6,7} = 3.79gm^2$. The inertia is obtained from the CAD model of the CVT.

We are interested in the friction behaviour as seen at the output, therefore we carry all friction elements across the transmissions to lump them together

Figure 4.14: Simplified IPM of the transmission system

at the output in an element R_t . This process is discussed in detail next.

Friction

Based on observations, we assume a friction model that involves stiction and coulomb friction as shown in Fig. 4.15.

Figure 4.15: Friction model

or:

$$\tau_f = \begin{cases} \tau_{-coulomb}, & \text{if } \omega < 0 \\ \tau_{-stiction}, & \text{if } \omega \lesssim 0 \\ \tau_{+stiction}, & \text{if } \omega \gtrsim 0 \\ \tau_{+coulomb}, & \text{if } \omega > 0 \end{cases} \quad (4.25)$$

As the friction is modelled as a lumped element at the output of the CVT, it is expected to have a dependence on the transmission ratio, r_{ps} . Looking at Fig. 4.1 while assuming stiction-coulomb friction at the points 1...7 and using Eq. 4.13...4.18, a lumped friction element at the output would be of the

following form:

$$\begin{aligned}\tau_{fout} = & r_1\tau_{f1}\text{sgn}(\omega_1) + r_2\tau_{f2}\text{sgn}(\omega_2) \\ & + r_3\tau_{f3}\text{sgn}(\omega_3) + r_4\tau_{f4}\text{sgn}(\omega_4) \\ & + r_5\tau_{f5}\text{sgn}(\omega_5) + r_6\tau_{f6}\text{sgn}(\omega_6) \\ & + r_7\tau_{f7}\text{sgn}(\omega_7)\end{aligned}\quad (4.26)$$

where

$$r_1 = \frac{\mathcal{X} \tan \phi - r_{fr}^2(\mathcal{X} + 1)}{r_{fr}^2} = r_{ps}^{-1} \quad (4.27)$$

$$r_2 = r_{ps}^{-1} \frac{1}{\mathcal{X}} \quad (4.28)$$

$$r_3 = -r_{ps}^{-1} \frac{r_{fr}}{\mathcal{X}} \quad (4.29)$$

$$r_4 = r_{ps}^{-1} \frac{r_{fr}}{\mathcal{X} \tan(\phi)} = \frac{1}{r_{fr}} \frac{r_{ps}^{-1}}{r_{ps}^{-1} + (\mathcal{X} + 1)} \quad (4.30)$$

$$r_5 = -r_{ps}^{-1} \frac{r_{fr}^2}{\mathcal{X} \tan(\phi)} = -\frac{r_{ps}^{-1}}{r_{ps}^{-1} + (\mathcal{X} + 1)} \quad (4.31)$$

$$r_6 = r_{ps}^{-1} \frac{1}{\mathcal{X} + 1} \quad (4.32)$$

$$r_7 = 1 \quad (4.33)$$

and where, τ_{fx} is given by Eq. 4.25.

We can also deduce:

$$\omega_x = r_x \omega_{out} \quad (4.34)$$

such that:

$$\text{sgn}(\omega_x) = \text{sgn}(r_x) \text{sgn}(\omega_{out}) \quad (4.35)$$

With this, we can rewrite Eq. 4.26 to:

$$\begin{aligned}\tau_{fout} = & (|r_1|\tau_{f1} + |r_2|\tau_{f2} + |r_3|\tau_{f3} \\ & + |r_4|\tau_{f4} + |r_5|\tau_{f5} + |r_6|\tau_{f6} + \tau_{f7}) \text{sgn}(\omega_{out})\end{aligned}\quad (4.36)$$

which is of the form:

$$\tau_{fout} = a \frac{r_{ps}^{-1}}{r_{ps}^{-1} + (\mathcal{X} + 1)} + b r_{ps}^{-1} + c \quad (4.37)$$

with different coefficients, a, b and c for:

$$r_{ps}^{-1} < -(\mathcal{X} + 1) \quad (4.38)$$

$$-(\mathcal{X} + 1) < r_{ps}^{-1} < 0 \quad (4.39)$$

$$0 < r_{ps}^{-1} \quad (4.40)$$

Since $\mathcal{X} + 1 = 5$ and $-3.5 < r_{ps}^{-1} < 3.5$, only the bottom two of these are reached in practice, such that we can expect different coefficients for:

$$r_{ps}^{-1} < 0 \quad (4.41)$$

$$0 < r_{ps}^{-1} \quad (4.42)$$

Summing up, we expect to find different parameters for the following four situations:

1. $\omega_{out} < 0$ and $r_{ps}^{-1} < 0$
2. $\omega_{out} < 0$ and $r_{ps}^{-1} > 0$
3. $\omega_{out} > 0$ and $r_{ps}^{-1} < 0$
4. $\omega_{out} > 0$ and $r_{ps}^{-1} > 0$

For both stiction and coulomb friction.

Parameter identification: stiction To identify the parameters in the relation between stiction and the transmission ratio, we fixed the output of the PS-CVT with a torque sensor ($\omega_7 = \omega_{out} = 0$). We applied a constant, known, torque on the input ($\tau_1 = \tau_{in}$), varied the transmission ratio over the full range ($r_{ps}^{-1} = -3.5 \dots 3.5$). The torque that is measured at the output is equal to:

$$\tau_{out} = r_{ps}^{-1} \tau_{in} - \tau_{fout} \quad (4.43)$$

As τ_{out} , r_{ps} and τ_{in} are known, the friction torque τ_{fout} can be calculated. Since $\omega_{out} \approx 0$, we measure stiction.

The result of this measurement is shown in Fig. 4.16 together with the fitted curves. In this graph a positive τ_{fout} corresponds to a positive output velocity.

The coefficients (a, b, c) of the fitted curves are given by:

$$\begin{array}{ll} 0.31, 0.54, -0.05 & \text{for } \omega_{out} < 0 \wedge r_{ps}^{-1} < 0 \\ -5.06, 1.67, 0.16 & \text{for } \omega_{out} < 0 \wedge r_{ps}^{-1} > 0 \\ -0.45, -0.13, 0.02 & \text{for } \omega_{out} > 0 \wedge r_{ps}^{-1} < 0 \\ 1.34, -0.96, -0.04 & \text{for } \omega_{out} > 0 \wedge r_{ps}^{-1} > 0 \end{array} \quad (4.44)$$

In the measurement result, an hysteresis effect is present. The shape of the hysteresis varies with the rate at which the transmission ratio is changed. We have the following hypothesis on the cause of this effect: For different relative orientations of the spheres, a different pre-tension is exerted on the spheres due to deformational effects. Related to this, the spheres have a different extent of deformation and the springs that provide the pre-tension have a different

Figure 4.16: Stiction, measurements and fitted curves.

state. As the state of the springs change, the sphere has to move slightly. Special bearings have been applied to allow this but they only work well with sufficient rotation. In the case of these experiments there is no rotation at all since the goal is to measure stiction, making it difficult for the sphere to move. When the transmission ratio is changed with a higher rate, the relative orientation also changes with a higher rate such that the sphere has less time to move. This can cause a variation in pre-tension which causes a variation in friction.

Unfortunately, we did not have the means to test this hypothesis.

Parameter identification: coulomb friction To identify the parameters in the relation between the coulomb friction and the transmission ratio, a servo motor was fixed to the output of the CVT and driven at a constant velocity ($\omega_7 = \pm\omega_{out}$), both positive and negative. A torque sensor was used to measure the torque that was applied to the system to maintain this velocity, this is equal to the friction torque plus torques required to speed up and slow down internal parts of the CVT. The contribution of the latter is insignificant ($< 0.00354Nm$) compared to the friction torque. The transmission ratio was varied over the full range ($-3.5 \dots 3.5$). Since $\omega_{out} \neq 0$, we measure coulomb friction.

The results are shown in Fig. 4.17.

Figure 4.17: Coulomb friction, measurements and fitted curves.

The coefficients (a, b, c) of the fitted curves are given by:

$$\begin{array}{ll}
 -0.07, 0.06, -0.56 & \text{for } \omega_{out} < 0 \wedge r_{ps}^{-1} < 0 \\
 3.99, -0.27, -0.69 & \text{for } \omega_{out} < 0 \wedge r_{ps}^{-1} > 0 \\
 0.06, -0.10, 0.35 & \text{for } \omega_{out} > 0 \wedge r_{ps}^{-1} < 0 \\
 -3.20, 0.26, 0.42 & \text{for } \omega_{out} > 0 \wedge r_{ps}^{-1} > 0
 \end{array} \quad (4.45)$$

Efficiency

Based on the coulomb friction that was identified in the setup, we can say something about the efficiency. The maximum input torque, $\tau_1 = \tau_{in}$, is $2Nm$. The ideal, lossless, output torque, τ_{outi} , is given by:

$$\tau_{outi} = r_{ps}^{-1} \tau_{in} \quad (4.46)$$

The actual output torque, τ_{out} , is equal to:

$$\tau_{out} = \tau_{outi} - \tau_{fout} = r_{ps}^{-1} \tau_{in} - \tau_{fout} \quad (4.47)$$

When delivering energy ($\text{sgn}(r_{ps}^{-1}) = \text{sgn}(\omega_{out})$), the efficiency, η , is given by:

$$\eta(\tau_{in}, \omega_{out}, r_{ps}^{-1}) = \frac{\tau_{out}}{\tau_{outi}} = \frac{r_{ps}^{-1} \tau_{in} - \tau_{fout}}{r_{ps}^{-1} \tau_{in}} \quad (4.48)$$

When recovering energy ($\text{sgn}(r_{ps}^{-1}) = -\text{sgn}(\omega_{out})$), the efficiency, η , is given by:

$$\eta(\tau_{in}, \omega_{out}, r_{ps}^{-1}) = \frac{\tau_{outi}}{\tau_{out}} = \frac{r_{ps}^{-1} \tau_{in}}{r_{ps}^{-1} \tau_{in} - \tau_{fout}} \quad (4.49)$$

A visualisation of the efficiency is given in Fig. 4.18 for negative output velocities and in Fig. 4.19 for positive output velocities.

Figure 4.18: Efficiency for negative output velocities

Figure 4.19: Efficiency for positive output velocities

4.6.2 Reconfiguration mechanism

Fig. 4.20 shows a lumped model of the reconfiguration mechanism.

The transmission ratio was determined experimentally by rotating the output shaft with a constant velocity and measuring the velocity at the input (ω_1) and output (ω_7) while changing the transmission ratio over the full range. Fig. 4.21 shows the experimentally determined transmission ratio with the theoretical relation. The difference is negligible.

Figure 4.20: Schematic illustration of the reconfiguration mechanism.

Figure 4.21: Inverse transmission ratio of the PS-CVT

Bandwidth

The total inertia of the reconfiguration system as seen from the input is equal to $I = 0.075 \text{ kgm}^2$. On this inertia, three torques act: A torque exerted by the gravity compensating spring τ_s , a torque exerted by gravity τ_g and a torque exerted by the reconfiguration actuator τ_m . The total torque that is applied to the inertia, τ_{tot} is equal to:

$$\tau_{tot} = \tau_m + \tau_s + \tau_g \quad (4.50)$$

The torque that is contributed by gravity, τ_g , is equal to:

$$\tau_g = -L \cos(\phi) m g \quad (4.51)$$

where $g = 9.81 \text{ ms}^{-2}$ is the gravitational acceleration constant, $m = 4.5 \text{ kg}$ the

mass of the system, $L = 0.12m$ the length of the arm and ϕ the angle of the mechanism as shown in Fig 4.20.

The torque that is contributed by the gravity compensating spring, τ_s , equal to:

$$\tau_s = (F_{s0} + (L_s - L_{s0})K)L\sin(\phi + \phi_2) \quad (4.52)$$

$$\text{where} \quad (4.53)$$

$$L_s = \sqrt{(L\sin(\phi) - P_y)^2 + (L\cos(\phi) - P_x)^2} \quad (4.54)$$

$$\text{and} \quad (4.55)$$

$$\phi_2 = \text{atan}\left(-\frac{L\sin(\phi) - P_y}{L\cos(\phi) - P_x}\right) \quad (4.56)$$

L_s is the length of the spring, ϕ_2 the orientation of the spring, $F_{s0} = -22N$ the minimum force that is required to elongate the spring, $L_{s0} = 0.106m$ the rest length of the spring, $K = 1800Nm^{-1}$ the stiffness of the spring and $P_x = -0.065m$, $P_y = 0.045m$ the location of the point P relative to the hinge point as shown in Fig. 4.20.

Assuming no friction losses, the reconfiguration actuator can deliver up to $\tau_m = 8.659Nm$.

Fig. 4.22 shows the maximum torque that can be applied to accelerate the inertia of the reconfiguration mechanism for all ϕ .

Figure 4.22: Max magnitude of the torque that can be applied to the reconfiguration mechanism

This shows that at any angle ϕ , we can at least apply $\pm 8.23Nm$ to accelerate the reconfiguration mechanism.

The general concept of bandwidth cannot be applied but we can do something pragmatic as follows.

Say ϕ_1 is a sinusoid:

$$\phi_1 = A \sin(\omega t) \quad (4.57)$$

then the acceleration is also a sinusoid:

$$\ddot{\phi}_1 = -A\omega^2 \sin(\omega t) \quad (4.58)$$

with amplitude $A\omega^2$.

Our maximum torque is $8.23Nm$ results in a maximum acceleration of:

$$a_{max} = \frac{\tau_{max}}{I} = 109.73 \text{rad/s}^2 \quad (4.59)$$

So

$$A\omega^2 \leq 109.73 \text{rad/s}^2 \quad (4.60)$$

We know that A lies within the range of the mechanism, $0 \leq A \leq \frac{0.79}{2}$. Assuming we don't want to restrict the range, we get:

$$\omega \leq \sqrt{109.73/0.395} \text{rad/s}^{-1} \approx 2.65 \text{Hz} \quad (4.61)$$

which gives us a feeling for the “bandwidth” of the system.

4.7 Discussion and Exploration

In this section we discuss two parameters that can be changed in the design of the DH-CVT to achieve different results: the maximum allowable output torque and the ratio of the fixed-ratio transmissions.

4.7.1 Scaling of the system with output torque

For the discussed realisation of the DH-CVT, dimensions of the friction spheres are chosen at 60mm radius for practical reasons. However, the spheres could be scaled in order to achieve a different maximum allowable output torque. By combining equation 4.18 and 4.24, a relation can be obtained between the minimum required sphere radius and the maximum allowable output torque. As the sphere radius (R) scales to the third power with the maximum input and output torque, an increase of a factor ten of the sphere radius, results in an increase of a factor thousand of the output torque (figure 4.23). In perspective, upscaling the considered system to increase output torque can be achieved rather easily for high torque applications. However, downscaling the system to reduce dimensions would dramatically reduce the maximum output torque, making the considered variable transmission impractical for small robotic systems.

Figure 4.23: Scaling of the required friction wheel radius with output torque

4.7.2 High torque vs high velocity power transmission in the spheres

The maximum output torque, τ_7 , of the Dual-Hemi CVT is limited by the maximum allowable throughput torque at the friction spheres, τ_3 and the kinematic relations of the system, Eq. 4.13...4.17 as explained in Section 4.4.2. By increasing the fixed ratio transmissions (r_{fr}), the throughput torque of the spheres can be reduced (Eq. 4.14). When reducing the throughput torque of the spheres, the required preload on and the radius of the spheres can be reduced (Eq. 4.21).

Reducing the radius of the spheres reduces their inertia and enables reducing the size and weight of the reconfiguration mechanism - yielding a higher bandwidth (Section 4.6.2) - and of the complete system. On the other hand, the effect of the inertia on the system scales quadratically with r_{fr} .

Reducing the preload reduces the load on the bearings and thus frictional forces at the bearings. In addition to this, the increased rotational velocity may further reduce the losses in the bearings ((Schaeffler, 2013)). On the other hand, the impact of the frictional forces on the system scales quadratically with r_{fr} .

Summing up, increasing the ratio of the fixed-ratio transmission r_{fr} is expected to reduce the size of the system and have a favourable effect on the bandwidth of the reconfiguration mechanism. Whether or not it has a favourable effect on the total inertia and friction of the system requires closer examination.

4.8 Conclusion

In this work, we presented the Dual-Hemi CVT: a novel CVT designed that is aimed to have properties that we believe are required to apply continuously

variable transmissions to their full potential in robotics. We discussed the most important design details and a characterization of the system. We showed that the system has all the aforementioned required properties.

The ratio range includes positive, negative and an infinite ratio, the system is fully back-drivable under all conditions and the reconfiguration action is completely decoupled from the transmission action.

In Section. 4.6.1 we showed that the efficiency is between 90 and 100% for a large part of the operation range of the CVT. The efficiency quickly drops to zero when the inverse transmission ratio, r_{ps}^{-1} or the input torque τ_{in} approaches zero. The significance of the work required for reconfiguration is application-dependent and will be studied in future work.

When looking at the friction properties of the system in Section. 4.6.1, we see that the stiction is significantly higher than the coulomb friction. In combination with the relatively low bandwidth discussed in Section 4.6.2, this may cause problems in terms of controlling the output torque.

In the discussion in Section. 4.7, we have shown that this specific design of the CVT will have significantly more beneficial properties when scaling up while it will have significantly less beneficial properties when scaling down.

Using a different (higher) transmission for the fixed ratio transmissions is expected to reduce the size of the system and the inertia of the reconfiguration mechanism, resulting in an increased bandwidth. It may or may not have a net favourable effect on the friction and inertia of the system.

In chapters 5 and 6, we discuss the application of the DH-CVT in controlled-passive actuation experiments. It would also be interesting to compare the properties of the Dual-Hemi CVT design with other CVT designs such as described in (Peshkin et al., 2001) and (Everarts et al., 2015).

CHAPTER 5

INERTIA-DRIVEN CONTROLLED PASSIVE ACTUATION

5.1 Introduction

For traversing unstructured environments, legged robots could be applied effectively. However, until now, a too high energy consumption prevents practical applicability ((Hardarson, 1997; Seeni et al., 2008)).

As we have shown in chapter 2, significant losses occur in electric actuators when used in legged locomotion. For this reason, we aim to reduce the energy flow through, and as a result the losses in electric actuators.

To minimise the energy flow through electric actuators, the required energy for moving the legs could be provided from a different energy source. Since moving a leg is a periodic motion, there is a periodic energy flow and, therefore, this energy source can be a buffer. This buffer can either be a spring, mass or equivalent in an other domain. Since energy will be dissipated in the system, the system needs to be extended with an energy source to replenish the energy that is lost.

However, a storage element alone is not enough: in order to control the state of the leg, the energy flow needs to be controlled. This can be done with a transmission: by changing the transmission ratio the energy flow can be controlled. Preferably, this is done in a continuous manner. A transmission

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where the transmission ratio can be changed in a continuous manner is called a CVT (Continuously Variable Transmission) and when it has a transmission range that includes positive ratios, negative ratios and a zero ratio, it is called an IVT (Infinitely Variable Transmission). From now on, we only use the term CVT, as it encapsulates the entire class. The range of attainable transmission ratios is treated as a property of a CVT.

In this work, we present a method to manipulate a load mass with a force actuator in the form of a CVT with a flywheel, as an alternative to delivering the actuation power with an electric actuator. The flywheel provides the energy for actuation and by changing the transmission ratio of the CVT the energy flow between the flywheel and the load mass can be controlled. This alternative method is named inertia-driven Controlled Passive Actuation, or inertia-driven CPA.

To show how the state of a system can be controlled using inertia-driven CPA, we first derive the differential equation that describes the dynamics of the system. Feedback linearisation is applied to simplify the system to a moving mass. On this simplified system, feedback control can be applied to achieve control goals. For this work, we apply velocity and position control in both simulations and experiments to show that inertia-driven CPA can be used to achieve these control goals.

Section 5.2 positions inertia-driven CPA in the field. Section 5.3 treats the theory of actuating a mass with another mass. In Section 5.4, the simulations are introduced and the simulation results presented. In Section 5.5, the experiments are discussed and experimental results presented. Section 5.6 concludes the chapter.

5.2 Related work

Using energy storage elements and CVTs in drive-trains is not a new idea. To start, many authors already suggested combining springs with electric actuators (see, for example: (Cherouvim & Papadopoulos, 2010; Poulakakis et al., 2005; Pratt & Williamson, 1995; Folkertsma et al., 2012)). Variable Stiffness Actuators (for example (Groothuis et al., 2012; Fumagalli et al., 2012; Jafari et al., 2011, 2010; Kim & Song, 2010)) take compliant actuation one step further by achieving tuneable compliant actuation with the goal of energy efficient actuation. Some include a CVT in the design to achieve this. In the work of Stramigioli, van Oort and Dertien (Stramigioli et al., 2008), a method is presented to recover, and re-use, negative work by using a spring in combination with a CVT. In Stramigioli and van Dijk (Stramigioli & Dijk, 2008), the same method is advocated to mechanically implement energy-conservative non-linear feedback to create stable limit cycles. Both aim at dynamically stable locomotion with oscillatory behaviour. These methods are applying storage elements

- springs - in actuation, not to achieve motion control but rather impedance control. In our work, we want to achieve motion control. Also, we apply a flywheel instead of a spring.

The use of flywheels in drivetrains is well known for acting as a low-pass filter on the velocity, thus smoothing out irregularities in the energy supply. Also the use of an inertial storage element in combination with a CVT in the drive train is well known. In the automobile industry, fly-wheels in combination with CVT's have been applied to use combustion engines in their more efficient range of operation while at the same time maintaining the car's capabilities to accelerate and decelerate (referred to as driveability) (see, for example, (Serrarens et al., 2003), (Serrarens et al., 1999)). In addition, CVT's have been applied in combination with an electric actuator to enable using the electric actuator in a more efficient range of operation (see, for example, the work on Cobots (Peshkin et al., 2001), (Faulring et al., 2004), (Moore et al., 1999), the work on MOSTS (Kim et al., 1999), (Kim et al., 2002) and (Widner & Hamel, 1998a), (Widner & Hamel, 1998b)). In these methods, the CVT (and flywheel) are used to assist the original energy source, being either a combustion engine or an electric motor. In our work, we use the flywheel as primary energy source.

5.3 Actuating a mass with another mass

The goal of our CPA experiment is to control the state, $[x_l, p_l]$, of a load mass that is affected by gravity using the energy that is present in a storage element, in this case another mass.

Changing the state of a mass changes the kinetic energy, E_k :

$$E_k = \frac{1}{2m}p^2 \quad (5.1)$$

where m is the mass and p the momentum, as well as the potential energy, E_p , (when gravity affects the mass)

$$E_p = mgx \quad (5.2)$$

where x is the height, m the mass and g the gravitational acceleration.

So, essentially, we want to transfer energy from the “source” mass to the “load” mass and the other way around in a controlled way. This controlled transfer of energy is achieved with a variable transmission, a CVT.

5.3.1 System dynamics

To achieve a control solution that enables the controlled transfer of energy from the “source” mass to the “load” mass, we start with deriving the differential equation that describes the dynamics of the system.

Figure 5.1: A bond-graph model of a load mass, m_ℓ , that is affected by gravity and actuated with an inertia-driven CPA. The elements that are part of the CPA are enclosed in a rectangle: the source mass, m_s and the variable transmission, r . The control input to the system is the transmission ratio, r . The source mass, m_s , is not affected by gravity.

We have a model of the inertia-driven CPA system as shown in Fig. 5.1.

In Fig. 5.1, r is the transmission ratio of the CVT, m_ℓ is the mass of the “load” mass, m_s is the mass of the “source” mass and g is the gravitational acceleration.

Note that the “source mass” has differential causality in the bond-graph model. This suggests that there are some unmodeled dynamics in the system: the finite flexibility of the components of the transmission. In this work, we assume that the transmission is rigid and that any unmodeled modes of vibration have a sufficiently high frequency to make this assumption valid.

With $\dot{x}_s$ the velocity of the “source mass” and $\dot{x}_\ell$ the velocity of the “load” mass, we obtain:

$$\ddot{x}_\ell = -\frac{1}{rr_p} \frac{m_s}{m_\ell} \ddot{x}_s - g \quad (5.3)$$

$$\ddot{x}_s = \frac{1}{r_p} \left(\frac{\dot{1}}{r} \right) \dot{x}_\ell + \frac{1}{r_p} \frac{1}{r} \ddot{x}_\ell \quad (5.4)$$

Which can be combined into:

$$\ddot{x}_\ell = -\frac{1}{r_p^2} \frac{1}{r} \frac{m_s}{m_\ell} \left(\left(\frac{\dot{1}}{r} \right) \dot{x}_\ell + \frac{1}{r} \ddot{x}_\ell \right) - g \quad (5.5)$$

We rewrite this as:

$$\ddot{x}_\ell(r, \dot{r}, \dot{x}_\ell) = \frac{-\frac{1}{r_p^2} \frac{1}{r} \frac{m_s}{m_\ell} \left(\frac{\dot{r}}{r}\right) \dot{x}_\ell - g}{1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell}} = \frac{\frac{1}{r_p^2} \frac{1}{r^3} \frac{m_s}{m_\ell} \dot{r} \dot{x}_\ell - g}{1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell}} \quad (5.6)$$

5.3.2 Controlled energy transfer

Now that we have the differential equation that describes the dynamics of the system, we will explain how the controlled transfer of energy can be done in such a way that control goals are achieved.

As part of the solution, feedback linearisation is applied in a similar way to what is done in computed torque control. This is extended with feedback control to achieve velocity and position control.

Feedback linearisation

Since the system dynamics are non-linear, we propose the use of feedback linearisation. To shape the system to resemble a classical control problem, the dynamics of the system are preferably described by Eq. 5.7.

$$\ddot{x}_\ell = \frac{F_a}{m_\ell} \quad (5.7)$$

where F_a is the actuation force.

Recall from Eq. 5.6 that the system dynamics are described by:

$$\ddot{x}_\ell(r, \dot{r}, \dot{x}_\ell) = \frac{\frac{1}{r_p^2} \frac{1}{r^3} \frac{m_s}{m_\ell} \dot{r} \dot{x}_\ell - g}{1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell}} \quad (5.8)$$

By multiplying with $\frac{m_\ell F_a}{F_a m_\ell}$ This can be rewritten to:

$$\ddot{x}_\ell(r, \dot{r}, \dot{x}_\ell) = \frac{\frac{1}{r_p^2} \frac{1}{r^3} m_s \dot{r} \dot{x}_\ell - m_\ell g}{F_a \left(1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell}\right)} \frac{F_a}{m_\ell} \quad (5.9)$$

We would like Eq. 5.9 to be equal to Eq. 5.7, hence:

$$\frac{\frac{1}{r_p^2} \frac{1}{r^3} m_s \dot{r} \dot{x}_\ell - m_\ell g}{F_a \left(1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell}\right)} = 1 \quad (5.10)$$

If we replace the actuation force, F_a with the desired actuation force F_{ad} , this can be rewritten to Eq. 5.11.

$$\begin{aligned}
\dot{r} &= \frac{F_{ad} \left(1 + \frac{1}{r^2 r_p^2} \frac{m_s}{m_\ell} \right) + m_\ell g}{\frac{1}{r^3 r_p^2} m_s \dot{x}_\ell} \\
&= \frac{F_{ad} \left(r^2 r_p^2 + \frac{m_s}{m_\ell} \right) + r^2 r_p^2 m_\ell g}{m_s \frac{\dot{x}_\ell}{r}} \\
&= \frac{F_{ad} \left(r^2 r_p^2 + \frac{m_s}{m_\ell} \right) + r^2 r_p^2 m_\ell g}{m_s r_p \dot{x}_s}
\end{aligned} \tag{5.11}$$

By varying r according to Eq. 5.11, such that $F_a = F_{ad}$, the system dynamics as given by Eq. 5.6 can be written as:

$$\ddot{x}_\ell = \frac{F_a}{m_\ell} \tag{5.12}$$

and the system can be represented as shown in Fig. 5.2. $\ddot{x}_\ell(F_a)$ is linear such that controller design is straight forward.

Figure 5.2: A bond-graph model of the simplified CPA system. The model shows the load mass, m_ℓ with a source of effort. This control input is now the desired actuation force, F_{ad} .

Closing the loop

Now that we have applied feedback linearisation, conventional control theory can be applied to design feed-back control loops. In this part we briefly discuss the design of the controllers that will later be used in the simulations and experiments. First, velocity control and second, position control.

Velocity (momentum) control For velocity control we can use the following control law:

$$F_{ad} = K_d (\dot{x}_{\ell d} - \dot{x}_\ell) \tag{5.13}$$

where K_d is the controller gain and $\dot{x}_{\ell d}$ is the desired velocity of the “load” mass. The closed-loop dynamics become a first-order system that converges to the setpoint $\dot{x}_{\ell d}$:

$$\ddot{x}_\ell = \frac{K_d}{m_\ell} (\dot{x}_{\ell d} - \dot{x}_\ell) \quad (5.14)$$

If we take $V_\ell(s) = \mathcal{L}(\dot{x}_\ell(t))$ and $V_{\ell d} = \mathcal{L}(\dot{x}_{\ell d}(t))$ and transform to the Laplace domain, we get:

$$V_\ell = \frac{1}{1 + \frac{M_\ell}{K_d}s} V_{\ell d} \quad (5.15)$$

For which the steady-state solution is:

$$\dot{x}_\ell = \dot{x}_{\ell d} \quad (5.16)$$

and the time constant, τ :

$$\tau = \frac{K_d}{m_\ell} \quad (5.17)$$

Position control We can also achieve a damped second order behaviour for position control:

$$F_{ad} = K_p(x_{\ell d} - x_\ell) + K_d(\dot{x}_{\ell d} - \dot{x}_\ell) \quad (5.18)$$

Where K_p is the proportional and K_d the differential controller gain and $x_{\ell d}$ is the desired position of the “load” mass. The closed-loop dynamics become a second-order system that converges to the setpoint $x_{\ell d}$. Taking $X_\ell(s) = \mathcal{L}(x_\ell(t))$ and $X_{\ell d} = \mathcal{L}(x_{\ell d}(t))$ and transforming to the Laplace domain, we get:

$$y = \frac{\frac{K_p}{m_\ell} + \frac{K_d}{m_\ell}s}{s^2 + \frac{K_d}{m_\ell}s + \frac{K_p}{m_\ell}} y_d = \frac{\omega_n^2 + 2\omega_n^2\zeta s}{s^2 + 2\omega_n^2\zeta s + \omega_n^2} y_d \quad (5.19)$$

With bandwidth:

$$\omega_n = \sqrt{\frac{K_p}{m_\ell}} \quad (5.20)$$

and damping coefficient:

$$\zeta = \frac{K_d}{2\sqrt{m_\ell K_p}} \quad (5.21)$$

5.4 Simulations

In this section, the controllers as discussed in Section 5.3.2 are applied to the dynamic model as shown in Fig. 5.1. This results in the controlled model as shown in Fig. 5.3.

Figure 5.3: A bond-graph model of a load mass, m_ℓ , that is affected by gravity and actuated with a inertia-driven CPA through a pulley, r_p . The elements that are part of the CPA are enclosed in a rectangle: the source mass, m_s and the variable transmission, r . The control input to the system is the transmission ratio, r . The source mass, m_s , is not affected by gravity.

Simulations are done for force actuation and velocity and position control. For force actuation and velocity control, we look at the response to a step

motion profile. For the position control, we look at the response to a pulse motion profile (First a step up and then a step down).

The parameters used in the simulations are shown in Table 5.1. where E_{ks0} is the kinetic energy present in the “source” mass at $t = 0$.

Parameter	Value	Unit
I_s	0.24	kgm^2
E_{ks0}	3	J
m_l	0.5	kg
g	9.81	ms^{-2}
r_p	0.02	
r	$-1.5 \dots 1.5$	
Velocity control		
- K_d	1.58	
Position control		
- K_p	5	
- K_d	5	

Table 5.1: Model parameters

5.4.1 Simulation results

Now, the simulation results are presented. First, the results for force actuation are shown, second the results for velocity control and third, the results for position control. For all simulations, we show how the system performs at following the setpoint, how r is changed to achieve this and where the energy is flowing in the system during the process.

Force actuation

Fig. 5.4 shows the results for force actuation. Fig. 5.4a shows that the actual force exactly follows the setpoint. Fig. 5.4b shows that r needs to be increased continuously to maintain the force. Why this happens is shown in Fig. 5.4c. As the kinetic energy in the source mass, E_{ks} , is converted to energy in the load - in this case a mass in a potential field - E_{Pl} , the velocity of the source mass, $\dot{x}_s$, decreases. Eq. 5.11 shows that decreasing $\dot{x}_s$ results in an increase of $\dot{r}$ and, as a result, exponential growth of r .

Velocity control

Fig. 5.5 shows the simulation results with velocity control. Fig. 5.5a shows that the velocity of the load mass, $\dot{x}_l$, follows the setpoint with a first-order response. In Fig. 5.5b, we see that the rapid change in velocity requires a rapid

change in r . Fig. 5.5c shows that when the velocity of the load mass becomes positive, kinetic energy from the source mass, E_{ks} , is converted into potential energy of the load mass, E_{pl} . As this happens, the velocity of the source mass, $\dot{x}_s$, decreases and r needs to be increased continuously for the system to provide the force to overcome gravity and maintain the velocity of the load mass.

Position control

Fig. 5.6 shows the simulation results for position control. In Fig. 5.6a we see that the position of the load mass, x_ℓ , tracks the setpoint with critically damped second-order behaviour as designed. Fig. 5.6b shows that r is non-zero when the energy content of the load mass is changing and returns to zero when the energy content of the load mass is constant. Fig. 5.6c shows that, when the load mass is lifted to its goal position, kinetic energy from the source mass is converted into potential energy of the load mass. When the load mass is dropped to its initial state, the energy is converted the other way around and recovered completely.

5.4.2 Discussion of the simulation results

The simulation results show that inertia-driven CPA can be applied as a source of effort for actuation when using an ideal CVT. That is, a CVT with no limit to $\dot{r}$ and no internal dynamics (mass, compliance, friction).

5.5 Experiments

To test the theory of inertia-driven CPA in practise, the controllers discussed in Section 5.3.2 are applied to a test setup. In this setup, a CVT with a transmission range of $-1.5 \dots 1.5$ is created by combining a NuVinci CVT (Fallbrook Technology, 2015), with a transmission range of $0.5 \dots 1.8$, with a set of fixed transmissions. This method of extending the range of a CVT is also described as a “parallel IVT” in (Mangialardi & Mantriota, 1999). A top-view photo of the setup is shown in Fig. 5.7.

A boundary condition for Inertia-driven CPA to work is that the energy in the “source” mass, E_{ks} must be greater than 0. If this condition is not met, no energy is available for actuation. A small dc-motor that is connected to the “source” mass can be used to meet this condition. This can be done in an energy efficient way, but for the purpose of these experiments, we have chosen for the following straight-forward solution.

The “source” motor is used to both create a virtual mass by means of control and automatically compensate for changes in the system energy that are not

caused by the load (but by inertia and friction in the CVT) in the following way.

Assuming the total energy of the system is:

$$E_{total} = E_{kl} + E_{pl} + E_{ks} + E_o \quad (5.22)$$

where E_{kl} is the kinetic energy of the load, E_{pl} is the potential energy of the load, E_{ks} is the kinetic energy of the virtual source mass and E_o is other energy that is not in the source nor load. There is also irreversible power loss:

$$P_{loss} = -\dot{E}_{total} \quad (5.23)$$

Such that the lost energy is equal to:

$$E_{loss} = -E_{total} + C \quad (5.24)$$

We can combine Eq. 5.22 and eq. 5.24 into:

$$C = E_{loss} + E_{kl} + E_{pl} + E_{ks} + E_o \quad (5.25)$$

where C actually is the initial kinetic energy in the virtual source, E_{ks0} .

Without compensation, the kinetic energy in the virtual source mass, E_{ks} is equal to:

$$E_{ks} = E_{ks0} - (E_{loss} + E_{kl} + E_{pl} + E_o) \quad (5.26)$$

such that it gradually decreases as energy is irreversibly lost.

To compensate for the changes in the system energy that are not caused by the load, we control the kinetic energy in the virtual source mass, E_{ks} , to be:

$$E_{ks} = E_{ks0} - (E_{loss} + E_{kl} + E_{pl} + E_o) + E_{loss} + E_o \quad (5.27)$$

$$= E_{ks0} - (E_{kl} + E_{pl}) \quad (5.28)$$

$$= E_{ks0} - \left(\frac{m_\ell}{2} \dot{x}_\ell^2 + m_\ell g x_\ell \right) \quad (5.29)$$

This is achieved by controlling the velocity of the source mass by Eq. 5.30.

$$\dot{x}_s = \sqrt{\frac{2}{m_s} \left(E_{ks0} - \left(\frac{m_\ell}{2} \dot{x}_\ell^2 + m_\ell g x_\ell \right) \right)} \quad (5.30)$$

Where E_{ks0} is the kinetic energy present in the “source” mass at $t = 0$.

With the setup, experiments were done with position and velocity control on the load mass. For the position control experiments, we look at the response to a pulse motion profile, for velocity control we look at the response to a motion profile that consists of first a pulse up and then a pulse down.

Except for the control gains, the same parameters are used as in the experiments. The parameters are shown in Table 5.2.

In the experiments, we look at:

Parameter	Value	Unit
I_s	0.24	kgm^2
E_{ks0}	3	J
m_l	0.5	kg
g	9.81	ms^{-2}
r_p	0.02	
r	$-1.5 \dots 1.5$	
Velocity control		
- K_d	0.3	
Position control		
- K_p	0.08	
- K_d	0.3	

Table 5.2: Experiment parameters

1. The control goal
2. The transmissions ratio, r
3. The energy flows in the system

5.5.1 Experimental results

First, the results for velocity control and, then the results for position control are treated. For all experiments, we show how the system performs at following the setpoint, how r is desired to change to achieve this goal, how r actually changes (r is calculated based on velocity measurements) and where energy is flowing in the system in the process.

Velocity control

Fig. 5.8 shows the experimental results with velocity control.

Fig. 5.8a shows that the load approximately follows the setpoint with a low bandwidth and a time delay. A time delay is also present between the desired and actual transmission ratio as shown in Fig. 5.8b. Clearly, the control of r is not optimal, and hence the motion control performance not good. Fig. 5.8c shows that when the load mass is lifted, kinetic energy from the source mass is converted into (primarily) potential energy. When the load mass is lowered again, energy is converted from potential energy in the load mass to kinetic energy in the source mass.

Position control

Fig. 5.9 shows the experimental results with position control.

In Fig. 5.9a we see that the load follows the setpoint with a somewhat low bandwidth and maybe also a time delay. The transmission ratio, r , as shown in Fig. 5.9b shows a delay in the response between the desired and actual transmission ratio. Fig. 5.9c shows that when the load mass is lifted, kinetic energy from the source mass is converted into (primarily) potential energy. When the load mass is lowered again, energy is converted from potential energy in the load mass to kinetic energy in the source mass.

5.5.2 Discussion of the experimental results

The experimental results confirm that inertia-driven CPA can be used to actuate a load. Comparison of the experimental results with the simulations results shows a significantly lower performance. The most significant difference between the experiment and the simulation is the non-ideal CVT. The CVT used in the experiments has significant internal dynamics (friction, mass) and a limited rate at which r can be changed.

The rate at which r can be changed is limited by the design of the NiVinci CVT. There is a significant amount of friction in the “switching” mechanism which limits its ability to “shift” fast. Higher rates of change for r can either be achieved with a more powerful “switching” motor or a with different CVT design.

5.6 Conclusions

In this chapter, we have suggested and shown that an (inertial) load can be actuated using the energy that is present in another rotating inertia. The energy flow is controlled using a CVT. Feedback linearisation was applied to simplify the controlled system to a mass with a force actuator to which conventional feedback controlled was applied. Simulations and experiments have shown the controlled system can be used to achieve velocity and position control. Since the energy for actuating the load is supplied by the source mass, electric actuators only need to supply the energy to change the transmission ratio and to compensate for the energy lost in friction.

Comparing the simulation results with the experimental results shows a significantly lower performance in the experiments. This is expected to be caused by non-idealities in the CVT. The CVT used in the experiments has significant internal dynamics (friction, mass) and a limited rate at which r can be changed.

Future work includes an investigation of how the internal dynamics of the CVT and limited rate at which the transmission ratio can be changed affect the performance of CPA. Also, the effect on the energy supplied by the “source” motor will be evaluated.

(a) The actuation force, F_a . (b) The transmission ratio, r . (c) The kinetic energy in the load mass, E_{kl} , the potential energy in the load mass, E_{pl} and the kinetic energy in the source mass, E_{ks} .

Figure 5.4: Simulation results where the control law as given in Eq. 5.11 is used to actuate a force according to a setpoint. Fig. 5.4a shows that the setpoint is followed. Fig. 5.4b shows how r is changed to achieve this. Fig. 5.4c shows how energy is exchanged in the system to achieve this.

(a) The velocity of the load mass, $\dot{x}_\ell$, setpoint and feedback.

(b) The transmission ratio, r .

(c) The kinetic energy in the load mass, E_{kl} , the potential energy in the load mass, E_{pl} and the kinetic energy in the source mass, E_{ks} .

Figure 5.5: Simulation results where the control law as given in Eq. 5.13 is used control the velocity of the load mass to follow a setpoint. Fig. 5.5a shows how the setpoint is followed. Fig. 5.5b shows how r is changed to achieve this. Fig. 5.5c shows how energy is exchanged in the system to achieve this.

(a) The position of the load mass, x_ℓ . (b) The transmission ratio, r . (c) The kinetic energy in the load mass, E_{kl} , the potential energy in the load mass, E_{pl} and the kinetic energy in the source mass, E_{ks} .

Figure 5.6: Simulation results where the control law as given in Eq. 5.18 is used control the position of the load mass to follow a setpoint. Fig. 5.6a shows that the setpoint is followed. Fig. 5.6b shows how r is changed to achieve this. Fig. 5.6c shows how energy is exchanged in the system to achieve this.

Figure 5.7: A top view of the setup that is used to conduct the experiments. The figure shows: The CVT complete CVT (red) which is created by combining a NuVinci CVT (green) with a set of fixed transmissions, including a planetary gear (orange). The “switching” motor (blue) that is used to change the transmission ratio. A “source” motor (purple) that is used to create the source mass by means of control and the load (yellow) that is manipulated using the combination of CVT and “source” mass.

(a) The velocity of the load mass, $\dot{x}_\ell$, reference and measured.

(b) The transmission ratio, r .

(c) The kinetic energy in the load mass, E_{kl} , the potential energy in the load mass, E_{pl} and the kinetic energy in the source mass, E_{ks} .

Figure 5.8: Experimental results where the control law as given in Eq. 5.13 is used control the velocity of the load mass to follow a setpoint. Fig. 5.8a shows that the setpoint is followed. Fig. 5.8b shows how r is changed to achieve this. Fig. 5.8c shows how energy is exchanged in the system to achieve this.

(a) The position of the load mass, x_ℓ . (b) The transmission ratio, r . (c) The kinetic energy in the load mass, $E_{k\ell}$, the potential energy in the load mass, $E_{p\ell}$ and the kinetic energy in the source mass, E_{ks} .

Figure 5.9: Experimental results where the control law as given in Eq. 5.18 is used control the position of the load mass to follow a setpoint. Fig. 5.9a shows that the setpoint is followed. Fig. 5.9b shows how r is changed to achieve this. Fig. 5.9c shows how energy is exchanged in the system to achieve this.

CHAPTER 6

SPRING-DRIVEN CONTROLLED PASSIVE ACTUATION

6.1 Introduction

For traversing unstructured environments, legged robots could be applied effectively. However, until now, a too high energy consumption prevents practical applicability ((Hardarson, 1997; Seeni et al., 2008)).

As we have shown in chapter 2, a significant part of the energy consumption is due to the use of electric actuators, which have a low efficiency when applied straightforwardly in applications such as locomotion. For this reason, we aim to reduce the energy flow through, and as a result the losses in electric actuators.

To minimise the energy flow through electric actuators, the required energy for moving the legs could be provided by a different energy source. Since moving a leg is a periodic motion, this source of energy can be a storage element like a spring or mass. However, a storage element alone is not enough: in order to control the state of the leg, the energy flow needs to be controlled. This can be done with a transmission: by changing the transmission ratio, the energy flow can be controlled. Preferably, this is done in a continuous manner. A transmission where the transmission ratio can be changed in a continuous manner is called a CVT (Continuously Variable Transmission) and when it has a transmission range that includes positive ratios, negative ratios and a zero ratio, it is called an IVT (Infinitely Variable Transmission). From now on,

This chapter is based on Dresscher, D., de Vries, T. J. A., & Stramigioli, S. Spring-driven Controlled Passive Actuation, Manuscript submitted for publication.

we only use the term CVT, as it encapsulates the entire class. The range of attainable transmission ratios is treated as a property of a CVT.

In this work, we present a method where we apply a CVT with a storage element in the form of a spring as a force actuator to control the state of an inertia (“the load”) as an alternative to delivering the actuation power with an electric actuator. This alternative method will be named spring-driven Controlled Passive Actuation, or spring-driven CPA. In spring-driven CPA we decouple the energy supply function from the servo behaviour function (see Fig. 6.1); electric motors will still be present in the set-up, however, they will be used only for

1. supplying energy to the energy source in order to compensate for the energy losses, which can be done in an high-efficiency mode;
2. reconfiguration of the CVT; this can be done with little energy.

Hence, we expect that this actuation principle can result in a much lower energy consumption.

Figure 6.1: A diagram showing how CPA separates the energy source function from the servo behaviour function. A DC-motor is used in its high-efficiency region of operation as energy supply while CPA is applied to obtain servo behaviour.

Using experiments, we show that the proposed spring-driven CPA can be used to achieve servo behaviour.

The outline of the paper is as follows: In Section 6.2, we discuss a range of related efforts that involve the use of CVT’s and mechanical storage elements. We continue with the theory of spring-driven CPA in Section 6.3, followed by a description of the experimental setup that is used to put the theory to the test. In Section 6.5 the experimental results are presented and discussed. In Section 6.6 the most important conclusions are summarised.

6.2 Related work

CVT’s have been around for a while and many designs have been proposed (see, for example, (Beachley & Frank, 1979; Ansink, 2008; Peshkin et al., 2001)). In addition, some work on the development of Variable Stiffness Actuators

contains a form of CVT (see, for example, (Groothuis et al., 2012; Jafari et al., 2010, 2011; Kim & Song, 2010)).

Also, using energy storage elements and CVTs in drive-trains is not a new idea. To start, many authors already suggested combining springs with electric actuators (see, for example: (Cherouvim & Papadopoulos, 2010; Poulakakis et al., 2005; Pratt & Williamson, 1995)). Variable Stiffness Actuators (for example (Groothuis et al., 2012; Fumagalli et al., 2012; Jafari et al., 2011, 2010; Kim & Song, 2010)) take compliant actuation one step further by achieving tuneable compliant actuation with the goal of energy efficient actuation. Some include a CVT in the design to achieve this. In the work of Stramigioli, van Oort and Dertien (Stramigioli et al., 2008), a method is presented to recover, and re-use, negative work by using a spring in combination with a CVT. In Stramigioli and van Dijk (Stramigioli & Dijk, 2008), the same method is advocated to mechanically implement energy-conservative non-linear feedback to create stable limit cycles. Both aim at dynamically stable locomotion with oscillatory behaviour. In (Everarts et al., 2012), the first steps are made to apply the concept described in (Stramigioli et al., 2008) to an active ankle prosthesis. These methods are applying storage elements - springs - in actuation, not to achieve motion control but rather impedance control. In our work, we want to achieve motion control.

Here, we consider a spring as the source of energy, but an inertia could also be applied. The use of flywheels in drivetrains is well known for acting as a low-pass filter on the velocity, thus smoothening irregularities in the energy supply. The use of an inertial storage element in combination with a CVT in the drive train is well known. In the automobile industry, the use of fly-wheels in combination with CVT's have been applied to use combustion engines in their more efficient range of operation while at the same time maintaining the car's capabilities to accelerate and decelerate (referred to as driveability) (see, for example, (Serrarens et al., 2003), (Serrarens et al., 1999)). In addition, CVT's have been applied in combination with an electric actuator to enable using the electric actuator in a more efficient range of operation (see, for example, the work on Cobots (Peshkin et al., 2001), (Faulring et al., 2004), (Moore et al., 1999) and the work on MOSTS (Kim et al., 1999), (Kim et al., 2002) and (Widner & Hamel, 1998a), (Widner & Hamel, 1998b)). In these methods, the CVT (and flywheel) are used to assist the original energy source, being either a combustion engine or an electric motor. In our work, we use the storage element as primary (and single) energy source.

6.3 Actuating a mass with a spring

The goal of our CPA experiment is to control the position of a load inertia using the energy that is present in a storage element, in this case a spring.

Changing the state of a mass changes the kinetic energy, E_k :

$$E_k = \frac{1}{2I}p^2 \quad (6.1)$$

Where I is the inertia and p the momentum.

So, essentially, we want to transfer energy from the spring, the source, to the mass, the load, and the other way around in a controlled way. This controlled transfer of energy is achieved with a variable transmission, a CVT.

To achieve a control solution that enables the controlled transfer of energy from the spring to the inertia, we start with deriving the differential equations that describe the dynamics of the system.

We have a model of the spring-driven CPA system as shown in Fig. 6.2.

Figure 6.2: A model of an inertia, I_ℓ , that is actuated with a spring-driven CPA. The elements that are part of the CPA are enclosed in a rectangle: the spring, C_s and the variable transmission, r . The control input to the system is the transmission ratio, r .

in Fig. 6.2, r is the transmission ratio of the CVT, I_ℓ is the inertia of the “load”, C_s is the compliance of the source spring.

With $\dot{x}_s$, we indicate the velocity of the spring and with $\dot{x}_\ell$ the velocity of the mass, we obtain:

$$F_\ell = m_\ell \ddot{x}_\ell = -r F_s = -r \frac{x_s}{C_s} \rightarrow \ddot{x}_\ell = -\frac{r}{m_\ell C_s} x_s \quad (6.2)$$

$$\dot{x}_s = r \dot{x}_\ell \quad (6.3)$$

By partially inverting the dynamics of the system, we can calculate r as:

$$\begin{aligned} r &= -\tau_a \frac{C_s}{x_s} \\ &= -\frac{\tau_a}{\tau_s} \end{aligned} \quad (6.4)$$

where τ_a is now the desired actuation torque and τ_s the spring torque.

By varying r according to Eq. 6.4, we create an abstraction layer such that Eq. 6.2 can be rewritten to:

$$\ddot{x}_\ell = \frac{\tau_a}{m_\ell} \quad (6.5)$$

and the system can be simplified to Fig. 6.3. $\ddot{x}_\ell(\tau_a)$ can be treated as linear such that controller design is straight forward.

Figure 6.3: A model of the abstracted CPA system. The model shows the load inertia, I_ℓ with a source of effort. This control input is now the desired actuation torque, τ_a .

6.4 The experimental setup

An experimental set-up has been realised in order to test the feasibility of this concept. A photo of the setup is shown in Fig. 6.4, a schematical representation is shown in Fig. 6.5. It consists of a spring, C , with a Dual-Hemi CVT, DH-CVT, that drives an inertial load, I_ℓ . The ratio of the CVT can be changed by means of an electric motor. Note that, with respect to the conceptual model in Fig. 6.2, the spring winder (enclosed in the blue dotted rectangle) is added to the scheme. The spring winder is used to add energy to the system if necessary. The control of the device is built in three layers, namely: Ratio control, Torque control and Position control. The Dual-Hemi CVT is discussed in detail in chapter 4. In this section, the other above mentioned components are discussed.

6.4.1 Source spring and load inertia

The source spring is a clock-spring with a stiffness, C , of 0.17 Nmrad^{-1} , the load inertia is a brass disc with an inertia, I_ℓ , of $6.35 \cdot 10^5 \text{ gcm}^2$.

6.4.2 Spring winder

The spring winder allows to add energy to the source spring. It consists of a (back-drivable) DC motor-gearbox combination. To maintain the energy in the

Figure 6.4: A photo of the experimental setup.

spring, the motor is driven with a constant current.

Assuming that the gearbox losses are mainly not velocity dependent (Schempf & Yoerger, 1993), we estimate the relation between the output torque, τ_{mout} , and input current, i_m of the motor-gearbox combination to be linear for positive currents:

$$\tau_{mout} = K_m r_{gb} i_m - \tau_\ell \quad (6.6)$$

Where K_m is the motor constant, r_{gb} the gearbox ratio and τ_ℓ the (linear) gearbox losses. The parameters for the relation are experimentally determined by fixing the output axis of the gearbox to a torque sensor and measuring the output torque for a range of input currents. Using a linear fit, the relation is estimated to be:

$$\tau_{mout} = -0.350 i_m \quad (6.7)$$

Based on this estimate of the springwinder output torque, we estimate the spring torque, τ_s .

6.4.3 Ratio and position control

The ratio and position controllers control the position of the reconfiguration mechanism to obtain a ratio setpoint and the position of the load, respectively. This is done using PID controllers which are tuned based on the system dynamics. The details are not instrumental to the goal of this chapter and hence not discussed in more detail.

Figure 6.5: IPM model of the complete CPA system

6.4.4 Torque control

The torque control layer calculates the ratio setpoint, r_{sp} of the CVT based on Eq. 6.4 with the addition of a friction compensating term τ_{fout} :

$$r_{sp} = \frac{\tau_{sp} + \tau_{fout}}{\tau_s} \quad (6.8)$$

where τ_{sp} is the torque setpoint, τ_{fout} the friction torque which is estimated based on the friction model discussed in Section 4.6.1 and τ_s the spring torque.

6.4.5 Motion profile

The motion profile that is used in the experiments is shown in Fig. 6.6.

6.5 Experimental results

In this section, we discuss the experimental results in order of control layer: first the ratio control, second the torque control, and third the position control. In addition we discuss the energetic behaviour of the system.

Figure 6.6: Motion profile

6.5.1 Ratio control

First, we evaluate the performance of the ratio control layer. To do this, we compare the measured ratio with the ratio setpoint (Fig. 6.5). Fig. 6.7 shows the tracking performance of the transmission ratio. This figure shows that the ratio is tracked reasonably well.

Figure 6.7: Ratio setpoint and actual ratio during the experiment

6.5.2 Torque control

Second, we evaluate the torque control layer. For this, we compare the actual torque with the torque setpoint (Fig. 6.5). The actual torque is estimated based on the acceleration of the output and the inertia of the load, I_ℓ : $\tau_a = I_\ell \ddot{x}_\ell$. Fig. 6.8 shows the tracking performance for the actuation Torque.

Figure 6.8: Torque setpoint and actual torque during the experiment

Fig. 6.8 shows that the friction compensation that is applied is not able to compensate adequately for the influence of friction on the output torque. This may be improved by applying a more advanced friction model such as the LuGre model (Astrom & Canudas-de Wit, 2008), which would be able to capture the hysteresis effect that is present (see chapter 4).

6.5.3 Position control

Third, we evaluate the performance of the position control layer. For this, we compare the position feedback with the position setpoint (Fig. 6.5). Fig. 6.9 shows the tracking performance for the position and Fig. 6.10 for the velocity.

Figure 6.9: Position setpoint and actual position during the experiment

Figure 6.10: Velocity setpoint and actual velocity during the experiment. The black ellipses mark clear stiction behaviour.

Although the load follows the setpoint, there are large positioning errors. This can be explained by the stick-slip behaviour - recognisable by alternating zero and non-zero velocities - that shows in Fig. 6.10 (Astrom, 1995). This is caused by the presence of stiction, of which the effect is most recognisable in the locations that are marked with black ellipses. The stick-slip behaviour causes jerky movements and significantly affects the performance. At this moment, it is unclear whether the stiction is present between in the contact between the spheres or in the planetary gear.

6.5.4 Energy efficiency

An important goal of controlled passive actuation is to create an energy efficient alternative to actuate robots. To evaluate if we achieved this goal, we first compare the energy that is supplied by the spring winder motor, $E_{supplied}$ with the energy that is stored in the spring, E_s , and the load, E_ℓ .

The energy supplied by the springwinder motor, $E_{supplied}$, is calculated as the integral of the product of the estimated spring torque, τ_s , and velocity of the spring winder motor, $\dot{x}_{springwinder}$, over time:

$$E_{supplied} = \int \tau_s \dot{x}_{springwinder} dt \quad (6.9)$$

The potential energy in the spring, E_s , is calculated based on the spring state, x_s , and the spring stiffness, K , as:

$$E_s = \frac{1}{2} K x_s^2 \quad (6.10)$$

The kinetic (co-)energy stored in the load, E_ℓ , is calculated based on the velocity of the load, $\dot{x}_\ell$, and the inertia of the load, I_ℓ :

$$E_\ell = \frac{1}{2} I_\ell \dot{x}_\ell^2 \quad (6.11)$$

Fig. 6.11 shows the energy in the load, E_ℓ , in the spring, E_s , and the energy that is supplied by the spring winder, $E_{supplied}$.

Figure 6.11: Energy energy that is stored in the load (red), stored in the spring (blue) and supplied by the spring winder motor (Green).

Fig. 6.11 shows that at the end of the experiment, only 8J of the approximately 30J supplied by the springwinder is still present in the system. The overall mechanical efficiency in this experiment is about 27%.

To evaluate how effective we are at recovering the energy used in the cyclic motion, we compare the energy exchange between the spring and the CVT, E_{in} , and the load and the CVT, E_{out} . The energy exchange between the spring and the CVT, E_{in} , is calculated with the integral of the estimated spring torque, τ_s and the velocity at the input of the CVT, which is calculated based on the output velocity and the ratio of the CVT:

$$E_{in} = \int \tau_s r_{PS}^{-1} \dot{x}_{output} dt \quad (6.12)$$

The energy exchange between the CVT and the load is calculated as the kinetic energy stored in the load, given in Eq. 6.11.

Fig. 6.12 shows the energy that is exchanged between the CVT and the spring, E_{in} and the energy that is exchanged between the CVT and the load, E_{out} .

Fig. 6.12 shows that, over the course of the experiment, the energy that is recovered with the spring is insignificant compared to the losses. Short

Figure 6.12: Energy exchange between the CVT and the load (red) and the CVT and the spring (blue)

moments where a little energy is recovered to the spring are present but hardly visible. At other times when the energy in the load is reduced, the energy is simply lost in friction in the system.

The energy efficiency is significantly lower than expected. In chapter 4, a characterisation of the DH-CVT is discussed and this shows that the efficiency is close to one for a large part of the range of operation (See Fig. 4.6.1). A possible explanation for the low overall efficiency in these experiments is that, due to the stick-slip effect, the system resides unnecessary long in the region of lower efficiency around $r_{PS}^{-1} = 0$.

6.6 Conclusions and future work

In this work we discussed the concept of Spring-Driven Controlled Passive actuation. In Controlled Passive Actuation we want to achieve servo behaviour by using the energy from a mechanical storage element and controlling the energy flow with a CVT. Doing so, we separate the energy supply function from the servo behaviour function. The ultimate goal is to realise an energy efficient alternative to actuate robots. With Controlled Passive Actuation, we try to achieve this in two ways. Firstly, we want to reduce the losses in the electric actuator by minimising the energy flow. Secondly, we want to effectively recover energy from cyclic motions. Using experiments, we have put the idea to the test.

The experiments show that servo behaviour can be achieved with spring-driven Controlled Passive Actuation and that we are successful in separating the servo behaviour function from the energy source function of an actuator. In

the experiments discussed in this paper, the performance of the servo behaviour is still significantly limited. Based on what we observed in the experiments, we suspect this is primarily caused by stick-slip effects. For future work, it is interesting to investigate where the stiction is located in the system and if it can be reduced. In addition, the force control loop needs improvement, possibly by applying more advanced friction models to estimate the influence of friction.

The goal of Controlled Passive Actuation is to realise an energy efficient alternative to actuate robots. So far, we did not succeed in this. Both the overall efficiency of the system as its capability to recover energy are still insufficient. Given that the energy efficiency of the Dual-Hemi CVT is nearly one in a large part of its range of operation this does not need to be a limiting element. It would be interesting to see how the efficiency improves when more smooth operation is achieved.

In chapter 7, a comparison of the characteristics of spring-driven CPA as discussed in this chapter with the characteristics of inertia-driven CPA as discussed in chapter 5 is discussed. In the former, most of the system is rotating continuously at relatively high velocities while in the latter, most of the system is at almost standstill most of the time. Due to this, friction effects in the CVT will have a different effect on the performance. To evaluate how, we plan to do inertia-driven CPA experiments with the Dual-Hemi CVT.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Introduction

Legged robots can offer solutions in areas where wheeled and tracked robots fail to handle the terrain elegantly. Two remaining obstacles for practical application of legged robots are robustness and energy efficiency. Parties like Boston Dynamics (Boston Dynamics, 2016) have been very successful at improving the robustness over the past years, at the expense of using more energy. If we want energy autonomous robots - robots that can harvest the energy that they need for operation from their environment - we need to harvest more energy or reduce the energy consumption or both. This research was set out to contribute to decreasing the energy consumption of legged robots by reducing the energy consumption in the locomotion system. This way, we wanted to make a step toward energy autonomous legged robots. For this, we formulated two key objectives:

1. Increase insight in the mechanisms of energy loss
2. Contribute toward the improvement of energy efficiency

In light of these goals, this chapter reflects on the insights and contributions obtained in this thesis in section 7.2, the implications for the research field in section 7.3 and suggestions for future research in section 7.4. With section 7.5 we conclude the conclusion.

7.2 Findings

In this section, we piece together the insights and contributions from the work in this thesis, with respect to the key objectives of the study.

7.2.1 Insight in the mechanisms of energy loss

When comparing legged locomotion to wheeled and tracked locomotion, there are two clear differences that affect energy consumption directly. First, there is extra movement as the legs need to make steps - start-stop behaviour - and second, the mass of the robot needs to be actively supported by the legs - this requires continuous torques.

Electric motors are widely applied in legged locomotion. When operating near their nominal velocity and with relatively low torques, electric motors can be very efficient. However, when actuating start-stop behaviour and continuous torques, electric actuators are not very efficient as we have shown in chapter 2. When applying electric motors in robotics in a naive way, the energy losses in the actuators are unnecessarily high: in an example that we discussed in chapter 2, the losses in the electric motor constitute 72% of the total energy consumption (Fig. 2.8). We propose a procedure for carefully selecting a motor for maximum efficiency. By applying this procedure, improvements can be achieved: in the same example, we have shown a reduction in the energy losses to 40% of the total energy consumption (Fig. 2.10). The higher the motor steepness, the lower the losses for providing a given torque. However, the losses remain significant. There is a lot to gain if the efficiency of actuation can be improved even more.

7.2.2 Contributions toward the improvement of energy efficiency

Following the argument in the previous section, it makes sense to contribute toward the improvement of energy efficiency by improving the efficiency of actuation. We have worked towards reduction of the energy losses in two ways, namely:

1. When actuating robots, we typically want to achieve servoing behaviour. By decoupling the energy supply function from the servoing functionality and using the electric motor for the energy supply function only, the motor can operate in a more energy-efficient range of operation. A variable transmission is used to obtain the servoing functionality.
2. By including a mechanical storage element in the actuator, energy can be recycled in the mechanical domain for repetitive motions. By doing

so, the energy flow through the electric motor and thus the energy loss is reduced. A variable transmission can be used to control the energy flow between the load and the mechanical storage element and hence still incorporate the decoupling mentioned above.

Both are included in the actuation principle called Controlled Passive Actuation (CPA in short) which is discussed in chapters 5 and 6 of this thesis. In Controlled Passive Actuation, a mechanical storage element serves as an energy buffer and a Continuously Variable Transmission (CVT in short) is used to create the required actuation force and control the energy flow. Electric actuators are used to change the transmission ratio of the CVT and supply energy to the storage element.

We have discussed a version that uses a flywheel as storage element (chapter 5) and one that uses a spring as storage element (chapter 6). In chapters 3 and 4 we have discussed two novel CVT designs that have been specifically designed to be applied in Controlled Passive Actuation.

Between spring-driven and inertia-driven Controlled Passive Actuation, there are three main differences. The first and maybe most obvious difference between spring-driven and inertia-driven is the way the energy is stored. In spring-driven CPA, the energy is stored as potential energy in a spring. The amount of energy that is stored in a spring is given by the following relation:

$$E = \frac{1}{2}Kx^2 \quad (7.1)$$

where E is the stored energy, K the spring constant and x the state (compression/elongation) of the spring. Storing a non-zero amount of energy in a spring requires a non-zero position. In inertia-driven CPA, the energy is stored as kinetic energy in an inertia. The amount of energy that is stored in an inertia is given by:

$$E = \frac{1}{2I}p^2 \quad (7.2)$$

Where E is the stored energy, I the inertia and p the state (momentum) of the inertia. Storing a non-zero amount of energy in an inertia requires a non-zero momentum and thus a non-zero velocity. This difference has consequences for the losses in the system. Mechanical losses only occur at non-zero velocities. Therefore, storing energy in a flywheel is inherently associated with mechanical losses where this is not the case when storing energy in a spring.

The second difference is in the way the CVT is connected in the concept. In inertia-driven CPA, the CVT is connected such that $r = 0$ results in a constraint at the load and free motion at the source, which is required to maintain the energy in the source inertia (a constraint would result in the complete absorption of the stored energy). In spring-driven CPA, the CVT is connected in the opposite way such that $r^{-1} = 0$ results in free motion at the

load and a constraint at the source, which is required to maintain the energy in the source spring (free motion would result in the instantaneous release of the stored energy). Hence, in spring-driven CPA, the source spring is connected to the opposite side (right side in Fig. 7.1a) of the CVT compared to the source inertia in inertia-driven CPA (left side in Fig. 7.1b). The load is connected to the other side, left for spring-driven CPA and right for inertia-driven CPA.

Figure 7.1: Simplified IPM models of spring-driven and inertia-driven CPA. C_s/I_s are the source spring and inertia, respectively, I_ℓ is the load

The third difference is the way the actuation force is controlled. In spring-driven CPA, the ratio, r , is directly related to the actuation force F_{ad} (see also Eq. 6.4):

$$r^{-1} = \frac{F_{ad}}{Kx_s} \quad (7.3)$$

Therefore r will stay within its (physical) boundaries as long as F_{ad} stays within boundaries. Keeping r within its physical boundaries requires a bounded x_s and F_{ad} . In inertia-driven CPA, the rate of the ratio, $\dot{r}$, is changed to control the actuation force (simplified Eq. 5.11):

$$\dot{r} = \frac{F_{ad} \left(r^2 + \frac{m_s}{m_\ell} \right)}{m_s \dot{x}_s} \quad (7.4)$$

If the integral of this function stays within the boundaries of r , inertia-driven CPA can be applied. If not, the ratio will drift to either its minimum or maximum value, thus crippling the possibilities for actuation. There are two ways to ensure that $\int \dot{r} dt$ stays within the boundaries of r . First, F_{ad} can be such that $\int \dot{r} dt$ stays within the boundaries of r . Second, the velocity of the source inertia, $\dot{x}_s$ and the velocity of the load inertia, $\dot{x}_\ell$ are always related through the transmission ratio, r : $\dot{x}_\ell = r\dot{x}_s$. In addition, $\dot{x}_s$ should never go to zero as this would mean that there is no energy for actuation. Therefore, r will

become zero when $\dot{x}_\ell$ goes to zero. If $\dot{x}_\ell$ always goes to zero before r reaches a boundary, the boundaries of r are not reached.

This difference in control of actuation force has two consequences; First, it is easier to ensure proper operation within the bounds when applying spring-driven CPA than in the inertia-driven case. Second, the rate of the transmission ratio is changed to control the force in inertia-driven CPA, whereas this is ratio itself in the spring-driven case. In case of a comparable transmission, changing $\dot{r}$ instead of r results in a 90° lead and thus a faster response and a higher bandwidth.

When applying Controlled Passive Actuation, achieving good performance and high efficiency has proven to be difficult. By adding a CVT, a complex device with non-idealities that have to be dealt with is added. We have shown that we can separate the energy source function from the servoing behaviour function. We were not (yet) able to obtain good servo performance nor achieve the envisioned efficiency gains.

7.3 Implications for the research field

Based on the findings that were discussed in the previous section, we now relate the work discussed in this thesis to other work in the field.

To achieve robots that can move with high energy-efficiency, we need to look beyond currently available electric actuators. Either the motor steepness of electric motors needs to be increased such that they offer an energy efficient way of actuating robots when used in the traditional sense, or additional facilities need to be included in the drive to let the motor operate in its current high-efficiency region and/or reduce the energy flow through the motor.

The former is outside the scope of this thesis, the latter can for example be achieved with the approach discussed in chapters 5 and 6 of this thesis. A step has been made on this path but the search should continue. Other work in this direction includes the work of (Stramigioli et al., 2008; Stramigioli & Dijk, 2008), (Everarts et al., 2015, 2012) and (Alo et al., 2015).

CVT's can help in achieving lower energy losses in the actuators, but the requirements on the CVT are high. To successfully apply CVT's in robotics, we believe they need to have at least the following properties (see also Chapters 3 and 4):

1. To use the CVT to transfer energy both from the source to the load and vice versa, as well as the option not to transfer energy, the transmission range needs to include positive ratios, negative ratios and a zero or infinite ratio.
2. Building on point 1, the system needs to be back-drivable under all conditions.

3. For the actuation energy not to be delivered by the reconfiguration actuator, changing the transmission ratio needs to be kinematically decoupled from the transmission such that:
 - (a) Changing the transmission ratio should not perform work on other parts of the system.
 - (b) It is possible to change the transmission ratio when the system is loaded (non-zero forces on “Energy source” and “Load”).

In addition, to achieve a high level of energy efficiency, the CVT needs to be designed such that:

4. The transmission has high efficiency.
5. Changing the transmission ratio requires little work.

Furthermore, they need to be able to handle the typically high torques that occur in these applications while at the same time be sufficiently small and light. We have contributed to the exploration of the design space, but there is more work to do. In light of application for the actuation of robots, the work of (Peshkin et al., 2001) and (Everarts et al., 2015) is among the most interesting that we encountered.

There are other approaches that can provide (partial) solutions to help operate electric motors in their more energy efficient range of operation. For specific applications, springs can be added in series or parallel to the drive to reduce the load on the actuator (see for example (Holgate et al., 2008), (Folkertsma et al., 2012), (Brown & Ulsoy, 2011), (Rouse et al., 2013) and (Au & Herr, 2008)). By including a variable stiffness actuator or module, the compliance can be continuously adjusted to best fit the needs of the application (see, for example: (Groothuis et al., 2012), (Fumagalli et al., 2012), (Jafari et al., 2010, 2011) and (Kim & Song, 2010)). Some variable stiffness actuators include a CVT to adjust the effective stiffness. Although reducing the load on the actuator results in a reduced flow through the actuator, the actuator is still operating in its lower efficiency region.

Utilising oscillatory behaviour is recycling energy in the mechanical domain in its purest form (see, for example: (Poulakakis et al., 2005), (Brown & Zeglin, 1998), (Folkertsma et al., 2015; Wanders et al., 2015)). Applying this technique reduces the energy flow through the electric actuators. Utilising oscillations for locomotion typically results in “bouncy” behaviour which can work well for certain applications and less for other.

7.4 Recommendations for future research

Based on this thesis, we have some recommendations for how the work on this topic should be carried on.

First of all, the research in the concept of Controlled Passive Actuation is not finished yet. The work should continue with:

1. The inertia-driven version of controlled passive actuation should be investigated further, including experiments with a well-designed CVT such as the Dual-Hemi CVT. Based on the conceptual differences discussed in section 7.2, we think the concept could have interesting properties.
2. The “stick-slip” effect that is present in the Spring-Driven CPA experiments (chapter 6) needs further investigation. A specific question that needs an answer is: does it originate from the planetary gear or from contact between the spheres? In the case of the former, it can probably be addressed with a different design. If the latter is the case, it could be inherent to traction-based CVT’s.
3. Investigate the effect of the planetary gear on the efficiency.
4. Once well-working setups are achieved for both the inertia-driven and the spring-driven version of Controlled Passive Actuation, the efficiency gains need to be evaluated.

Second, in supplement of the research in the concept of Controlled Passive Actuation, some work needs to be done on closely related paths:

1. Do a thorough survey of different approaches that can help to reduce the energy losses in actuators.
2. Work towards miniaturisation of CVT’s that can handle high torques and have desirable properties (for example, as formulated in chapter 4) for application in robotics.
3. Evaluate and compare the energy-efficiency of alternative actuation technologies such as pneumatic actuation.
4. Investigate if it is possible to design electric motors with a high efficiency in the low-velocity - high-torque region.

Eventually, when the research in this field has progressed to apply energy efficient actuators in robots, an energy autonomous robot could be designed with such actuators and state-of-the-art solar panels. Such a robot could, for example, be applied for robotic dike inspection.

7.5 Conclusion

With a walking dike inspection robot in mind, we started on the path to increase insight in the energy losses in walking robots and to contribute to improving

the energy efficiency. By default, most robots are designed with electric actuators. In this thesis, we have shown that they contribute significantly to the energy losses. We have proposed a procedure to select motors for optimal efficiency. However, to achieve robots that can move with high energy-efficiency and versatility, we need to look beyond electric actuators alone. We have done so in this thesis by investigating the inclusion of a CVT and a mechanical storage element in the drive to reduce these losses: Controlled Passive Actuation. We have shown that the principle works and proposed innovative CVT designs to enhance the implementation. Now it needs to be further developed towards application.

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Douwe Dresscher was born in Smilde, the Netherlands in 1987. He received his B.Sc. in electrical engineering from the Hanze University of Applied sciences in 2008. During his B.Sc. studies he did an internship at Nestle Philippines, Cabuyau factory where he, among other things, worked on a system to control the chlorination of the local water supply. In his final assignment he worked on the development of a system that uses ultrasound waves for positioning and communication.

After obtaining his B.Sc. degree, he obtained his M.Sc. in Systems and Control from the University of Twente in 2011. During his M.Sc. studies he worked on the sensor system and the detection and recognition of litter for the JaClean robot. In his final assignment he worked on control, detection of failures and effective response to failures for the Kuka youBot.

Right after, he continued with his Ph.D. research in the Robotics and Mechatronics group at the University of Twente under supervision of Stefano Stramigioli and Theo J.A. de Vries. His Ph.D. research, reported in this thesis, was part of the Energy-Efficient design and control of mobile Robotic Sensor networks (ROSE) project under the Autonomous Sensor SYStems (ASSYS) program. During his Ph.D. he initiated, organised and participated in the RaM-MakerSpace; an initiative where members of the RaM group meet once a week to build robots. The mission: to combine, into a fun and energising process, the knowledge of enthusiastic roboticists with the existing technology of the RaM group, to generate new robotic solutions that inspire and amaze.

Douwe's research interest, besides the energy efficiency of (legged) robots, include the fields robotics for inspection and maintenance, and robotics for agriculture. For leisure, he loves spending time with his family and friends, reading, playing board games, running, cycling and technical hobbyism (if he can find the time).