Design and Analysis of a Compliant Shoulder Mechanism for Assistive Exoskeletons



DMS4 Master Thesis Group 4

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Abstract:

In this thesis the design and analysis of a passive exoskeleton to assist the elderly and workers for overhead tasks are presented. The scope of the exoskeleton is to compensate for the gravitational forces. The exoskeleton consists of a spherical shoulder mechanism and a passive variable stiffness mechanism. Both mechanisms are described and further ideas are presented. Numerical, analytical and experimental analyses of the variable stiffness mechanism are carried out and compared. Furthermore, topology optimisation is performed to reduce weight. The exoskeleton focuses on assistance in the sagittal plane, but due to the properties of the modules, also motion in different planes is supported. The final design compensates for 50% of the gravitational torque of the arm with the elbow stretched.

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Preface

The report is written by DMS4 Group 4 from Aalborg University in the period February 1st, 2021 to June 3, 2021, with supervising by Shaoping Bai.

We would like to thank Shaoping Bai for knowledgeable supervising, for providing help and information for the project. We also want to thank Zhongyi Li and the workshop in Fibiger-stræde 14 for their help in manufacturing of the exoskeleton and providing equipement that was used in the experiment.

Reading guide

This report uses the reference style APA. Source references will appear like the following: [Author, Year]. The bibliography can be found at the end of the report followed by the appendices. If a larger section is based on a specific reference, this reference will be stated explicitly at the beginning. The reference is displayed directly if specific statements, data or figures are used for an external source. If no source is given, the material is generated by the authors.

The numbering of figures, tables and equations is done by first stating the number of the chapter they appear in, a full stop and the incremented number of elements of the same class. For instance, the third element of a class in the fifth chapter is denoted as 5.3. Each table and figure have a short description next to the numbering underneath the element. Equations are numbered in the same scheme. The number is enclosed in brackets, that is (1.1) and is given at the right next to or closely under the equation.

The appendices appear after the bibliography with a capital letter instead of a chapter number. Sections and subsections are numbered with the capital letter of the appendix they belong to and the number of the section or subsection. A.1.1, A.1.2 and A.2 are examples.

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Chapter 1

Introduction

With the population of the world getting older and the working period in people's lives becoming more and more extended, there is an increasing need for motion aids. Exoskeletons can provide a motion aid for upper and lower body movements. An exoskeleton is a wearable system that provides physical assistance to its user through assistive torques and/or structural support [Maurice et al., 2020]. Wearable exoskeletons can be used in manufacturing, everyday living assistance, military and recreational applications. Some of the existing exoskeletons can be seen in Figure 1.1.



Figure 1.1: Examples of commercialised exoskeletons. In Figure 1.1(a) is the Angle Lges exoskeleton [Choi, 2021], in 1.1(b) the Backx exoskeleton [Choi, 2018] and in 1.1(c) is the ExoHeaver exoskeleton [Yatsun & Jatsun, 2018]

Most research has been carried out regarding the lower body movements and the reason for this is that the upper body movement is more complex, as a high level of range of motion is needed [Bai et al., 2018]. The upper body exoskeleton can assist the user especially when physical labour includes overhead tasks or tasks where the arms have to be raised for a long time.

Exoskeletons can be classified as active, passive and pseudo-active [Marinov, 2016]. Active exoskeletons consist of one or more actuators that transfer forces to the human body and thereby produce or assists in movement. These exoskeletons are used more for rehabilitation after an injury [Tim Bosch et al., 2016]. Despite their useful usage, the heavyweight, high cost and power supply are of concern. On the other hand, in passive exoskeletons, there is no need for external power. The function of passive exoskeletons is based on compliant elements, like springs, which can store or release energy when the human body is moving. In the remainder of this chapter, first, the range of motion of the shoulder is presented and then different upper limb exoskeletons and shoulder mechanisms are described before the problem formulation and approach of this thesis are presented.

1.1 Range of Motion

This section is based on [Krishnan et al., 2019]. The movements of the upper body are very complicated because of the high range of motion and dexterity of the arm. The basic movements of the shoulder can be seen in Figure 1.2.



Figure 1.2: Representation of human shoulder movements [Krishnan et al., 2019]

The movements in the sagittal plane are called flexion and extension. During flexion, the relative angle of the humerus from the rest position to the fully flexed position varies in the range of $0^{\circ} - 180^{\circ}$. The reversal of this motion is known as extension. If the extension

proceeds beyond the rest position of the humerus, it results in hyperextension.

The movements in the coronal plane are called abduction and adduction. During the abduction, the humerus moves away from the mid-line of the body. The reversal of this motion from a fully abducted position to the mid-line is known as adduction.

The movements in the transverse plane are internal and external rotations, which contribute to the internal and external axial rotation of the humerus. The movements around the vertical axis are called horizontal abduction, horizontal adduction and cross abduction.

There are also movements that are not confined to any cardinal plane. The first being the circumduction, which is the conical movement of the humerus, and the second being the generalised raising and lowering of the humerus, called elevation and depression, respectively.

1.2 Passive Upper Limb Exoskeletons Applications

Upper limb exoskeletons are designed to work together with the movements of the human upper limb. They can be divided into exoskeletons for motion amplification and exoskeletons for medical rehabilitation [Gull et al., 2020]. The first category includes the exoskeletons that can help people by reducing the effort of the user. For example, overhead work is a frequent cause of shoulder work-related musculoskeletal disorders, very common in the automotive and aerospace industries. In order to perform overhead activities, the arms have to be stretched up, which means that the shoulder experiences stress because of the weight of the arms. The second group of exoskeletons are associated with rehabilitation and upper body weakness or paralyses like stroke, spinal cord injury and orthopaedic injuries. This kind of exoskeletons assists patients with various arm and shoulder impairments. In this project, the focus is on the first category and so only the background of this is presented.

1.3 Background of Existing Passive Upper Limb Exoskeletons

In recent years different upper limb exoskeletons are produced with certain advantages and also some disadvantages. With time, the exoskeletons are enhanced in order to combine lightweight, reduced cost and comfort to the user. In this section, some upper limb exoskeletons, whose scope is for motion assistance, are described.

1.3.1 Paexo

The Paexo shoulder exoskeleton helps the workers to do activities, especially those that are overhead [Paexo, 2018]. The mechanism that this exoskeleton uses is mechanical cable pull technology. The weight of this exoskeleton is a significant advantage as it only weighs 1.9kg. Other benefits of the Paexo exoskeleton are the comfortability and the range of motions that it allows to the user. However, the range of assistance and the adjustment required to fit the arm of these exoskeletons are limited [Luque, 2019].



Figure 1.3: The Paexo shoulder exoskeleton [Paexo, 2018]

1.3.2 EksoVest

EksoVest is a passive exoskeleton produced by Eksobionics. The purpose of this exoskeleton is to help with tasks from chest height to above the head [Exobionics, 2018]. EksoVest reduces the strain and fatigue on workers and thereby lowering their overall risk of injury. However, some significant disadvantages of this exoskeleton are the high cost, the bulkiness and the weight. The weight of this exoskeleton is approximately 4.3kg.



Figure 1.4: The EksoVest exoskeleton [Exobionics, 2018]

1.3.3 EVO

EVO exoskeleton is an upgrade to EksoVest exoskeleton, which is also produced by Eksobionics. The specific exoskeleton, compared to the EksoVest, improves the range of motion and reduces the weight and the cost [Exobionics, 2020]. Their goal with the EVO is to help reduce shoulder, neck, and back injuries from overhead work, repetitive tasks, and overexertion.



Figure 1.5: The EVO exoskeleton [Exobionics, 2020]

1.3.4 MATE-XT

One of the products the Comau S.p.A. offers is an exoskeleton called MATE-XT. It provides passive and shoulder angle dependent support for the arm weight of the user [Pacifico & Scano, 2020]. It supports motion in the saggitational plane only and the overall structure is self aligning to the body of the user. The torque can be varied in four steps. Figure 1.6 shows the aperture.



Figure 1.6: Commercialized version of the MATE-XT S.p.A. [2021]

The support for the arm is generated in the shoulder part of the exoskeleton by a spring and an off-centred attachment to a disc [Pacifico & Scano, 2020]. The forces are transferred to the body via a kinematic chain that goes down to the lower back, where the exoskeleton is attached with a belt. The device is worn like a backpack and weighs 3.5kg. To the authors' knowledge, two of its advantages are the easy adjustability to the carriers body and the scalability of the supporting torque. Two of the drawbacks are, firstly that the back is straight which prohibits bending down and secondly the constant torque curve that does not account for the angle of the elbow.

1.3.5 Hyundai Vest Exoskeleton

Hyundai Vest Exoskeleton (H-VEX) is an exoskeleton that is produced by Hyundai Motor Group. The exoskeleton consists of an energy-storage multi-linkage mechanism, dissipating spring-loaded energy and a poly-centric shoulder joint mechanism. This joint mechanism is positioned on the transverse plane for its alignment with the movement of the human shoulder joint [Hyun et al., 2019]. Using the shoulder joint mechanism, during the abduction/adduction movement of the shoulder, the misalignment between the shoulder and the exoskeleton is avoided.



Figure 1.7: The Hyundai Vest Exoskeleton [Hyun et al., 2019]

1.3.6 Comparison of Existing Upper Limb Exoskeletons

In order to be more comprehensible a comparison of the aforementioned exoskeletons is done, which also includes more features of each exoskeleton. In Table 1.1 the weight, the price and the range of motion are compared.

Upper limb	Price	Weight	Range of
exoskeletons	[€]	[kg]	motion
Paexo	5700	1.9	Full
EksoVest	5000	4.3	Limitation on bending of the back
EVO	*	*	Full
MATE-XT	5000	3.5	Limitation on bending of the back
H-VEX	5000	2.5	Limitation on bending of the back

Table 1.1: Comparison of upper limb exoskeletons*Not published yet as EVO exoskeleton is a new release.

It has to be mentioned that EVO is a new release from Eksobionics and is an improvement to EksoVest as the company promises a much lighter and reduced cost compared to the previous model. Furthermore, the Paexo and EVO exoskeletons provide more range of motion when compared to the MATE-XT and Eksovest exoskeletons, where a limitation for bending of the back exist. The range of motion is described more thoroughly after this section. Regarding the cost of the exoskeletons, the Paexo and H-VEX exoskeletons are less expensive than the others.

1.4 Background of Existing Shoulder Joint Mechanisms

Different mechanisms are used in the shoulder to allow the exoskeleton to copy the motion of the complex human shoulder joint. The mechanism used in traditional exoskeletons uses a serial linkage, which consists of 3 revolute joints (3R) to implement the spherical motion of the human shoulder joint [Christensen & Bai, 2017]. However, the mechanism collided with the human body during the abduction motion. To overcome this drawback new designs were developed. Some examples of mechanisms invented by AAU university are the Double Parallelogram Linkage (DPL) [Christensen & Bai, 2017] and the Compact 3-DOF Scissors Shoulder mechanism (SSM) [Castro et al., 2019] which are described in detail in this section. Also, the mechanism used by H-VEX is described. At the end of this section, other existing mechanisms are discussed briefly.

1.4.1 Double Parallelogram Linkage

The double parallelogram linkage consists of four links and two offset angles as seen in Figure 1.8a. The two links L_1 and L_2 are in the first parallelogram and the links L_3 and L_4 are in the second parallelogram. The double parallelogram allows the movement of the internal/external rotation and it needs no actuator. Furthermore, it connects the two revolute joints, which enables the flexion/extension and abduction/adduction movements. The two axes of the revolute joints are aligned with the \mathcal{L}_1 and \mathcal{L}_2 , respectively as it can be seen in Figure 1.8a. These two axes share the same point of rotation, which is called the Remote Center (RC). In [Christensen & Bai, 2017], for the motion of the two revolute joints, two active actuators are used is it can be seen in Figure 1.8b, where the RC can also be seen, that allows the mechanism to rotate about three independent axes.

However, the active actuators are not in the scope of this projects so this mechanism can not be used as it is. A way of replacing these active actuators could be examined and used in this project as the shoulder joint mechanism. The advantages of the specific shoulder joint mechanism is that it is compact, light and reduces the possibilities for the collision with the user [Christensen & Bai, 2017].



Figure 1.8: The double parallelogram linkage in a) and in b) the double parallelogram mechanism with the active actuators for the two revolute joints [Christensen & Bai, 2017]

1.4.2 Compact 3-DOF Scissors Shoulder Mechanism

The Scissors Shoulder mechanism is a 3-DOF mechanism invented by Aalborg University [Castro et al., 2019]. The specific mechanism has the advantage over the 3R mechanisms as it eliminates the problem of collision to the human body. The three revolute joint axes that the mechanism rotates about can be seen in Figure 1.9. They share a common RC and so the linkages of the mechanism move in a spherical surface. Compared to the DPL mechanism, it increases the compactness and also reduces the possibility of collision with the human body.



Figure 1.9: The 3-DOF Scissors Shoulder Mechanism [Castro et al., 2019]

1.4.3 Four-bar Based Poly-Centric Shoulder Linkage

The four-bar based poly-centric shoulder linkage is used in the H-VEX as described before and can be seen in Figure 1.10. Other exoskeletons such as MATE and Eskovest, as described before, use a redundant DOF around the scapula to overcome the misalignment issue of the exoskeleton and the shoulder arm [Hyun et al., 2019]. In contrast, using the fourbar based poly-centric shoulder linkage, no redundant DOF is needed and so the weight of the exoskeleton is not increased unnecessarily. In the figure below the end point of the poly-centric structure, P_0 , is in contact with the upper arm.



Figure 1.10: Anatomical shoulder structure with the four-bar based poly-centric structure (blue lines) on the transverse plane [Hyun et al., 2019]

1.4.4 Other Existing Shoulder Joint Mechanisms

As mentioned before, conventional exoskeletons are using the serial linkage system with 3 revolute (3R) joints [Christensen & Bai, 2017], [Naidu et al., 2011]. The configuration of this shoulder joint mechanism can be seen in Figure 1.11. The disadvantage of this configuration is that it reduces the movement of the user in the coronal plane as it can collide with the body.



Figure 1.11: The serial linkage with 3 revolute joints [Naidu et al., 2011]

Another shoulder joint mechanism can be seen in Figure 1.12. This shoulder joint mechanism overcomes the issue with the collision of the exoskeleton with the user's body because a circular guide is used in the arm. However, the drawback of this shoulder joint mechanism is the singularity that can be seen in Figure 1.12 below, as there is an alignment of the axes of rotations of the joints 1 and 3, respectively. This means that the exoskeleton can not produce the abduction movement in the transverse plane. Furthermore, the circular guide can increase the weight of the exoskeleton by a significant amount.



Figure 1.12: The shoulder joint mechanism with two revolute joints and one circular guide [Lo & Xie, 2014]

One way of avoiding the singularity of the above shoulder joint mechanism is by introducing a redundant joint as can be seen in Figure 1.13. However, the issue with the weight of the circular guide still exists. Besides the circular guide, the weight of the exoskeleton is also increased as a redundant joint is added. This increases the workspace, but also makes the exoskeleton bulkier.



Figure 1.13: The shoulder joint mechanism with three revolute joints and one circular guide [Lo & Xie, 2014]

1.5 Problem Formulation

Tasks as lifting, carrying or handling objects at work often result in musculoskeletal injuries, such as strains and sprains [Exobionics, 2018]. Exoskeletons can be worn by employees in the workplace to reduce the possibility of an injury. Although the existing passive upper limb exoskeletons described before have significant advantages, their drawbacks are considerable and should be improved.

The purpose of this project is to design a mechanism with one or more compliant elements, which will combine portability, modularity, and compactness. The compliant mechanism has to compensate for the gravitational forces and thereby reduce the risk of injuries.

Most of the exoskeletons presented above focus on flexion-extension movement, which concerns mostly the workers on how to lift heavyweights. However, this is not the only movement that can cause injuries and so also a concept for assistance in other planes should be presented. This means that the designed exoskeleton ideally should assist the user in movements in different planes than the sagittal plane, or should easily be upgradable. In summary, the problem is formulated as:

A passive, compact and lightweight exoskeleton is to be developed, that supports the human arm movement by reducing the load acting on the shoulder muscles. Motion in the sagittal plane is of highest priority, but differing planes also have to be considered. If possible with the Covid situation at hand, a prototype is to be used to validate the design. If this proves impossible, numerical simulations are to be used for verification.

1.6 Problem Approach

The purpose of an exoskeleton is to represent the movements of human musculoskeletal structure and applying forces or torques on the user [Papadopoulos & Patsianis, 2007]. In this way, the upper limb motion can be supported. The human upper limb, as it can be seen in Figure 1.14, consists of the shoulder complex, the elbow complex, the wrist joint and the fingers [Chen et al., 2014]. The shoulder complex includes the clavicle, the scapula and the humerus.



Figure 1.14: Anatomy of the human upper limb [Chen et al., 2014]

The project is based on previous work done at Aalborg University, where prior to the period of the thesis at hand an advantageous mechanism has been developed [Bai & Li, 2019]. This mechanism is a revolute joint of variable stiffness. There are other Variable

Stiffness Mechanisms (VSM), but to the authors best knowledge, their range of motion is either too small to be used as torque providing device for an exoskeleton for the upper limb [Dežman & Gams, 2018] [Wolf et al., 2011] [Jafari et al., 2010] [Wolf & Hirzinger, 2008], or their working principle is similar to the VSM developed at AAU [Furnémont et al., 2015] [Vanderborght et al., 2009] [Van Ham et al., 2007]. This VSM is designed with a compliant joint mechanism and is able to adjust the stiffness to the user's needs by choosing certain design parameters. However, the specific mechanism uses an external power to in- or decrease the torque which makes it a pseudo-passive mechanism. To make it purely passive, it will be reconfigured to be used without external power.

The shoulder joint mechanisms presented in Section 1.4 represent different solutions to the problem of connecting an arm to the torso. All of those mechanisms provide different advantages and disadvantages. Since the focus of this thesis is among others on compactness, the SSM will be used. By providing three rotational DOF it follows the anatomical shoulder closely and although other mechanisms, i.e. the four-bar based poly-centric shoulder linkage used in the H-VEX exoskeleton, show better resemblance of certain motions of the shoulder complex, the SSM is very compact and does not require additional structure to follow the human body. The SSM, if physically realised, does not have singularities or redundant DOF, which makes it attractive for this project.

First, the shoulder will be reduced to one rotational DOF, which can move in the sagittal plane only and thereby capture the flexion-extension movement. The torque produced in this 1-DOF system will be compared with the torque produced by the mechanism, whose design concept will be analysed thoroughly in Chapter 2. After adjusting the parameters of the VSM to obtain a torque profile that matches the shoulder torque profile, the exoskeleton will be designed in detail and tested. Suggestions for additional design ideas are presented. The exoskeleton will be assessed based on the experiments and general improvements will be described.

Chapter 2

Modeling of the Exoskeleton

The following chapter explains the VSM as proposed by [Bai & Li, 2019] and [Li et al., 2020]. Special focus is put on the kinematics and the rotation-torque relation. Furthermore, the selected shoulder joint mechanism, which is the SSM, is described and analysed.

2.1 VSM Modeling

The following description is a short description of the VSM as described in [Bai & Li, 2019] with the purpose of clarifying the following chapters. This section is based on [Bai & Li, 2019]. The torque that the VSM can provide can be utilized to compensate the gravitational forces.

The VSM is based on a 4-bar linkage, where link-2 (l_2) is replaced by a linear spring element. The length of the fourth link is reduced to zero. Link-1 (l_1) is taken to be the input link and link-3 (l_3) is assumed to be connected to the output. Figure 2.1 shows the basis of mechanism and the final configuration, which is used in the VSM.



Figure 2.1: The initial four-bar linkage (left), which is the basis for the VSM. For the VSM the fourth link has zero length and the bar-2 is a linear elastic element (center). On the right is the final configuration of the VSM.

The torque T of the three-bar linkage is given by the multiplication of the Jacobian of the second link and the force produced by the spring:

$$T = JF \tag{2.1}$$

where F is the tension force within the compliant coupler link-2.

With link-4 having no length, the length of bar two is expressed as

$$l_2 = \sqrt{l_1^2 + l_3^2 - 2l_1 l_3 \cos\theta} \tag{2.2}$$

where θ is the angle between the input and output links. For the VSM, the length l_2 is defining the Jacobian:

$$J = \frac{dl_2}{d\theta} \tag{2.3}$$

The cable force *F* is composed of the force F_0 due to pretension and the force due to elongation of the second link, δl_2 .

$$F = k\delta l_2 + F_0 \tag{2.4}$$

where *k* is the stiffness of the spring.

By combining the equations and splitting them up into two terms, one influenced by the stiffness of the elastic element and the other by the pretension, the following torque expression is obtained:

$$T = k\delta l_2 J + NF_0 J \tag{2.5}$$

However, the final configuration of the VSM, which is shown in Figure 2.1 on the right, is composed of an inner (dashed light grey arc) and an outer (dashed dark grey arc) disc with freely rotating pulleys (blue circles), connected by a cable. This is not exactly the same configuration as the original three-bar linkage. Therefore the torque of the final configuration changes. The torque provided from the final configuration of VSM is given as:

$$T = J_1 F = J_1 k \delta l + J_1 F_0 \tag{2.6}$$

where δl is the elongation of the spring, J_1 the Jacobian of the mechanism. They are defined as:

$$\delta l = cN\delta l_2 \tag{2.7}$$

$$J_1 = cNJ \tag{2.8}$$

where c is a geometrical variable and N is the configuration number and is given by N = 1, 2, 3. The configuration number is the number of pulleys on the inner disc, that is connected to the outer disc. For N = 1 the configuration is shown in Figure 2.1 whereas Figure 2.2 displays the arrangement for N = 2. For each connected pulley on the inner disc, two pulleys of the outer disc have to be connected.



Figure 2.2: VSM configuration for N=2

From Equation (2.6) it can be concluded that the torque of the configuration, besides the length of the links and the spring properties, is also depended on the configuration number N and the geometrical variable c. The value for the geometric value is taken equal to 2, the reason for this is explained in Appendix A.

Figure 2.3 shows a torque curve of a VSM with arbitrary values for the design parameters.



Figure 2.3: Torque over relative angle θ for k = 4, $F_0 = 0$, N = 1

2.2 AAU's Scissors Shoulder Mechanism

In a PhD thesis presented at Aalborg University, a novel shoulder joint mechanism has been suggested. To allow for all rotatory degrees of freedom of a human shoulder, the links of the mechanism lay on a sphere, which has its centre in the centre of rotation of the shoulder [Castro et al., 2019]. To eliminate singularities that result from locking of motions in certain configurations, the SSM is composed of a row of two parallelograms as shown in Figure 2.4. This avoids the singularities and enables the same range of motion as provided by the human shoulder. As it can be seen in the figure below, the SSM consists of six linkages - four short and two long ones. The small linkages, which have half of the length of the longer ones, are connected with each other and the bigger one via a revolute joint. The two bigger linkages, which are located in the middle of the SSM, are also connected with a revolute joint in their centre point.



Figure 2.4: The SSM prototype [Castro et al., 2019]

2.2.1 Kinematics of the SSM

In order to describe the motion of the SSM, the kinematics of it has to be analysed. This can be done through forward kinematics or inverse kinematics. In the first analysis, the position of the end-effector is found with the angle of the joints given. In contrast, using inverse kinematics, the angles of the joints are found based on the location of the end-effector. For simplicity, a single rhombus is used to determine the kinematics, which can be seen in Figure 2.5. In this one rhombus, each linkage describes a circle arc on a spherical surface that can rotate. Furthermore, the three angles that the rhombus can rotate around can be seen in the figure below. The kinematics of the SSM is already derived by [Castro et al., 2019] and will be described briefly.



Figure 2.5: The one rhombus mechanism [Castro et al., 2019]

Forward Kinematics

For the forward kinematics, the inter-linkage angles φ_i and the Euler angles θ_i are used to find the rotation matrix R_e . The rotation matrix refers to the transformation of the end-effector coordinates to the ones of the global frame. For all the frames, the remote centre that was shown in Figure 1.9, is chosen. In Figure 2.6, where the different angles are shown, it can also be seen that there is an extra link VI, which represents the end-effector.



Figure 2.6: Inter-linkage joint angles φ_i and the Euler angles θ_i in the SSM [Castro et al., 2019]

From the figure above the equations that can be derived are:

$$\theta_1 = \varphi_1 + \frac{\varphi_1}{2} \tag{2.9}$$

$$\theta_3 = \varphi_6 - \frac{\varphi_2}{2} \tag{2.10}$$

Another equation can be derived using the spherical law of cosines:

$$\cos\theta_2 = \cos^2\alpha + \sin^2\alpha\cos(\pi - \varphi_2) \Rightarrow \theta_2 = \arccos(\cos^2\alpha - \sin^2\alpha\cos\varphi_2) \quad (2.11)$$

Equation (2.11) describes the relation between the end-effector pitch angle θ_2 and the internal angle φ_2 . This relation is represented in Figure 2.7 for different values of the curvature angle α .



Figure 2.7: The relation between the end-effector pitch angle θ_2 and the internal angle ϕ_2 for different values of the curvature angle α

From the figure above it can be seen that when the curvature angle α is increased, the end-effector pitch angle θ_2 is increased. Furthermore, for $\varphi_2 = 0$, the end-effector pitch angle θ_2 is twice the curvature angle α .

In order to relate the end-effector with the global reference frame the rotation matrices $Rz(\theta_1)$, $Rx(\theta_2)$ and $Rz(\theta_3)$ are used. These matrices express the 'route' from the end effector to the point A. The three rotations matrices are found as:

$$Rz(\theta_1) = \begin{bmatrix} c\theta_1 & -s\theta_1 & 0\\ s\theta_1 & c\theta_1 & 0\\ 0 & 0 & 1 \end{bmatrix}$$
(2.12)

$$Rx(\theta_2) = \begin{bmatrix} 1 & 0 & 0 \\ 0 & c\theta_2 & -s\theta_2 \\ 0 & s\theta_2 & c\theta_2 \end{bmatrix}$$
(2.13)

$$Rz(\theta_3) = \begin{bmatrix} c\theta_3 & -s\theta_3 & 0\\ s\theta_3 & c\theta_3 & 0\\ 0 & 0 & 1 \end{bmatrix}$$
(2.14)

With the three rotation matrices, and since all the links have as a common origin the remote centre as shown in Figure 1.9, the rotation matrix R_e is found as:

$$R_{e} = R_{z}(\theta_{1})R_{x}(\theta_{2})R_{z}(\theta_{3}) = \begin{bmatrix} c\theta_{1}c\theta_{3} - s\theta_{1}c\theta_{2}s\theta_{3} & -c\theta_{1}s\theta_{3} - s\theta_{1}c\theta_{2}c\theta_{3} & s\theta_{1}s\theta_{2} \\ s\theta_{1}c\theta_{3} + c\theta_{1}c\theta_{2}s\theta_{3} & -s\theta_{1}s\theta_{3} - c\theta_{1}c\theta_{2}c\theta_{3} & -c\theta_{1}s\theta_{2} \\ s\theta_{2}s\theta_{3} & s\theta_{2}c\theta_{3} & c\theta_{2} \end{bmatrix}$$

$$(2.15)$$

where $s\theta_i$ and $c\theta_i$ correspond to the sine and cosine functions of the angle θ .

Assuming a sphere of 60mm radius three cases of the position of the end-effector with respect to an initial point, which is located in the position z = 60mm, x = 0mm, z' = 0mm, can be seen in Figure 2.8. The Case 1 is for $\theta_1 = 0^\circ$, $\theta_2 = 45^\circ$, $\theta_3 = 0^\circ$, Case 2 is for $\theta_1 = 90^\circ$, $\theta_2 = 135^\circ$, $\theta_3 = 0^\circ$ and Case 3 for $\theta_1 = 45^\circ$, $\theta_2 = 90^\circ$, $\theta_3 = 0^\circ$.



Figure 2.8: End effector position for three different cases depending on the pitch angles θ_1 , θ_2 , θ_3

2.2.2 Design Limitations

A limitation of the SSM prototype is the collision of the bearings as can be seen in Figure 2.9. This happens when the SSM are fully stretched or fully folded.



Figure 2.9: Possible collision of the SSM because of possible bearings in the joints [Castro et al., 2019]

Assuming an angle θ_2 between the collision point axis M and the axis A of the base joint, an angle β from point C to the collision point axis M and using again the spherical law of cosines for the spherical triangle AMC then:

$$\cos \alpha = \cos \theta'_2 \cos \beta + \sin \theta'_2 \sin \beta \cos A \tag{2.16}$$

where A is the angle between the arcs β and θ'_2 but it was chosen not to shown in the figure for better clarity. Since the two edges are perpendicular to each other then the angle cos A is equal to zero and so the Equation (2.16) can be written as:

$$\theta'_2 = \arccos(\cos \alpha / \cos \beta)$$
 (2.17)

From the above equation it can be seen that the maximum angle, that the SSM can be stretched, depends on the angle of the links and also on the dimensions of the bearings. For that reason, the avoidance of big bearings can ensure that the SSM can be stretched further.

Singularities of the SSM

In order to see the possible singularities in the SSM the Yoshikawa's manipulability, w, has to be derived. The manipulability can give information on how far a position in the workplace

is from a singularity [Vahrenkamp et al., 2012]. It is defined relative to the determinant of the Jacobian *J* as:

$$w = \sqrt{\det\left(JJ^T\right)} \tag{2.18}$$

The Jacobian matrix is found from the equation that relates the angular velocities ω_e with the mechanism's joint velocities $\dot{\theta}$ and is equal to:

$$J = \begin{bmatrix} 0 & c\theta_1 & s\theta_1 s\theta_2 \\ 0 & s\theta_1 & -c\theta_1 s\theta_2 \\ 1 & 0 & c\theta_2 \end{bmatrix}$$
(2.19)

More details on how the Jacobian is derived can be found in Appendix B.

Substituting Equation (2.19) into Equation (2.18), the manipulability is equal to:

$$w = |s\theta_2| \tag{2.20}$$

From the above equation it can be seen that the manipulability and so the possible singularities depend on the pitch angle θ_2 . For the singularities to appear, the manipulability has to be equal to zero. This happens in two cases:

$$w = \begin{cases} 0, \theta_2 = 0^\circ \Leftrightarrow \phi_2 = 180^\circ \\ 0, \theta_2 = 180^\circ \Leftrightarrow \phi_2 = 0^\circ \end{cases}$$
(2.21)

From the above equations it can be concluded that the first singularity happens when the SSM is totally closed ($\theta_2 = 0^\circ$), which automatically means that the joint angle $\phi_2 = 180^\circ$. The second singularity occurs when the SSM is fully stretched ($\theta_2 = 180^\circ$), which results in the joint angle $\phi_2 = 0^\circ$. From Figure 2.7 it can be seen that this situation occurs when $\alpha = 90^\circ$. The two relations can also be verified from equation (2.11).

According to [Castro et al., 2019], the first singularity happens at 90° of shoulder external rotation and the second one at 90° of internal rotation. Both singularities should not concern the user as the first one is unusual in daily activity and the second one cannot exist because it would mean the penetration of the torso.

However, to avoid or come close to these two singularities a relation was derived in [Castro et al., 2019], in which the θ_{2max} is dependent on the number of rhombus n and the linkage curvature angle α as:

$$\theta_{2max} = 2\alpha n < 180^{\circ} \tag{2.22}$$

As mentioned before, the holes of the links are surrounded by material in order to include the bearings, bolts etc. This means that the possible singularities described before will not happen as $\theta 2$ or φ_2 can not reach the 0° and 180°. Combining Equation (2.17) and Equation (2.22) the limits of θ_2 can be derived as:

$$\theta_{2max} = 2n\theta_2' \tag{2.23}$$

$$\theta_{2min} = 2n\beta \tag{2.24}$$

Having these two equations, stability of the mechanism is achieved since the singularities are avoided as it ensures that the pitch angle θ_2 can not reach the singularity angles 0° and 180° .

2.2.3 SSM provided by AAU

The new prototype that has been received from AAU can be seen in the Figure 2.10. The device as it can be observed has bearings with small radius and can assure a significant amount of the maximum stretching of the SSM. The bearings that are used in the SSM can be found in Appendix C.



Figure 2.10: The provided SSM from AAU

As mentioned before there is material around each joint axis and so the links do not behave like line entities. This means that the maximum angle the SSM can be stretched to and how much they can be folded, both depend on the radius of the bearings and the curvature angle of the links, as shown from Equation (2.23) and Equation (2.24). With Equation (2.23), Equation (2.24) and Equation (2.17) the maximum and minimum pitch angles for θ_2 can be found. The results of the pitch angles and the specifications of the new SSM prototype can be seen in Table 2.1.

Linkage curvature angle α	38°
Intruisive angle β	3°
θ_{2max}	151.5°
θ_{2min}	12°

Table 2.1: Specifications of the provided SSM and the maximum and minimum pitch angles

From the above table the limits of θ_2 can be seen. It can be concluded that the SSM allows 78° of external shoulder rotation and 61.5° of internal shoulder rotation.
Chapter 3

Assistance in the Sagittal Plane

In the following chapter, a model for the motion of a human arm in the sagittal plane is presented. Based on this kinematic model optimisation is used to adjust the parameters of a VSM to obtain a best possible support for the important ranges of motion.

3.1 Motion of the Human Arm

To model the human arm, three sections are considered: the upper arm, the forearm and the hand. Those sections are connected by joints of various degrees of freedom. The shoulder joint or glenohumeral joint is a ball and socket joint. Following Reference [Zhang et al., 2011] it is kinematically represented by a spherical joint, resembled by three revolute joints, which have zero distance between each other and whiches axes of rotation are perpendicular to each other. The elbow can conduct two rotations, one that rotates the forearm around itself and one perpendicular to both the forearm and the upper arm. Thus it can be represented by two revolute joints with zero distance from each other. The wrist joint is included as a ball and socket joint with two rotational degrees of freedom [Soames et al., 1994]. In total, the model of the arm has seven degrees of freedom. To determine the position of the joints and the centres of mass of the different segments in space, the rotation matrices for the different coordinate systems of the joints are introduced in the following. The general coordinate system of the reference frame with index m is thereby oriented such that the x-axis is pointing sideways away from the body, the y-axis is pointing towards the front and the y-axis is pointing upward. The origin of the reference frame is located in the centre of the glenohumeral joint. In the following Figure 3.1 the arrangement of the joints is depicted along with the introduced coordinate systems and the positive definition of the rotation angles.



Figure 3.1: Configuration of the joints of the human arm following [Zhang et al., 2011]; (a) shows the configuration of the joints and (b) displays the introduced coordinate systems and definition of rotation angles

To determine the position of the centres of mass, which will be used to estimate the required torque, the homogeneous transformation matrices **H** are set up. Those consist essentially of the rotation matrix **R**, which gives the rotation from one coordinate system to another and the vector **d** pointing from the old coordinate system to the centre of the next one. The homogeneous matrix to relate a position from the i^{th} to the $(i - 1)^{th}$ system is composed as follows:

$$\boldsymbol{H}_{i-1}^{i} = \begin{bmatrix} \boldsymbol{R} & \boldsymbol{d} \\ \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{1} \end{bmatrix} = \begin{bmatrix} R_{11} & R_{12} & R_{13} & \boldsymbol{d}_{1} \\ R_{21} & R_{22} & R_{23} & \boldsymbol{d}_{2} \\ R_{31} & R_{32} & R_{33} & \boldsymbol{d}_{3} \\ \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{1} \end{bmatrix}$$
(3.1)

The distance between the coordinate systems three and four and the coordinate systems five and six can be taken from Table 3.1.

Sagmant	Length	Centre of Mass*	Weight**
Segment	[mm]	[%]	[%]
Upper Arm	281.7	57.72	2.71
Forearm	268.9	45.74	1.62
Hand	86.2	79.00	0.61

Table 3.1: Dimensions for the male arm [de Leva, 1996]
* Distance from the joint in percent of the segment length
** in percent of the body weight

With the distances given the homogeneous matrices are set up and can be seen in Appendix D. The indexing is done in such a way that the lower index refers to the original coordinate system and the upper one is the coordinate system that a point is related with. To obtain a translation of a point from an arbitrary coordinate system n to the base frame, the homogeneous matrices are multiplied.

$$H_m^n = H_m^1 H_1^2 H_2^3 \dots H_{n-1}^n \tag{3.2}$$

Thus the position of the arm can now be described in the base coordinate system and if the position of a point in one of the mentioned coordinate systems is known, it is also known in relation to the body. Figure 3.2 shows the arm in about the position to grab a glass from a table. The shoulder and elbow are both in flexion and the hand is held in hyperextension.



Figure 3.2: Arm model with $\theta_1 = -45^\circ$, $\theta_4 = 45^\circ$ and $\theta_6 = 30^\circ$; all other angles are zero. The shoulder, elbow and wrist are marked with a red circle. The green circles indicate the position of the centre of mass of the segments according to [de Leva, 1996]. The shoulder joint is positioned in the origin of the m-system.

For the estimation of the required torque to keep the arm elevated in a certain position the lever arms of the centres of mass are of special interest. Those are calculated from the xand y-coordinates of the centres of mass of the different segments by use of the Pythagoras theorem. The torque that is required to keep a segment elevated is the vector product of its centre of mass and the individual force vector. The direction and magnitude of the torque of the whole arm is then given by the vector sum of the torque vectors of the individual segments.

3.2 Motion of a Human Arm in the Sagittal Plane

To develop an exoskeleton for the whole arm with seven actuators is out of the scope of this work. Therefore the general model from the previous section is reduced to only two actuators: θ_1 and θ_4 . For simplification, those are renamed as follows:

$$\theta_1 = \gamma \tag{3.3}$$

$$\theta_4 = \alpha \tag{3.4}$$

In this reduced model, the hand is not considered to move but stays aligned with the forearm. In that case, pronation and supination have no influence on the position of the centre of gravity in space, which is why they are not considered and the corresponding actuator 5 is fixed at $\theta_5 = 0^\circ$. Hence, the first degree of motion that is considered, is flexion of the shoulder with the elbow angle α acting within the sagittal plane. This makes the overall shoulder torque a function of two inputs and defines its direction to be perpendicular to the sagittal plane. Figure 3.3 depicts the simplified arm model.



Figure 3.3: Graphical explanation of γ and α

The torque that is necessary to counter the gravitational forces for a sequence of n links in 2D with point masses is given by

$$T_{shoulder} = g \sum_{i=1}^{n} l_{CMi} m_i$$
(3.5)

where l_{CMi} is the lever arm of the mass of the *i*th segment, m_i is the mass of the segment and g is the gravitational acceleration. The length l_{CMi} is

$$l_{CMi} = \sum_{j=1}^{i-1} l_j \sin \sum_{k=1}^{j} \alpha_k + c_i l_i \sin \sum_{l=1}^{i} \alpha_l$$
(3.6)

with α_i being the angle between a link and its precursor. Applied to the n = 3 mechanism at hand with no angle between the hand and the forearm the three lever arms become

$$l_{CM1} = c_1 l_1 \sin \gamma$$

$$l_{CM2} = l_1 \sin \gamma + c_2 l_2 \sin (\gamma + \alpha)$$

$$l_{CM3} = l_1 \sin \gamma + l_2 \sin (\gamma + \alpha) + c_3 l_3 \sin (\gamma + \alpha + 0)$$
(3.7)

The weight of the segments is given from the person's weight and Table 3.1. With the mentioned considerations, the torque that the shoulder has to create can be calculated. Figure 3.4 shows the resulting curves.



Figure 3.4: Torque that is needed to counter the torque produced by the mass forces over the shoulder angle γ for different elbow angels α

It is noted, that there is no payload taken into account. The main reason is, that a passive exoskeleton is not capable of reacting to changes in the required torque by itself. Hence,

if the payload is considered wrongly to be too large, the positive effect of the support of the exoskeleton may be turned into a negative one, when the user has to constantly apply a downward torque instead of an upward torque to keep the arm at a constant level. This especially applies to the torque curves of $\alpha > 0^\circ$ due to the sign change for large γ . The VSM as considered in this thesis cannot sense this and will apply torque in the same direction as the arm. Thus the human muscles are dealing with heavier loads than without the exoskeleton. Contrary to this higher loading of the body, even too small assistance reduces the risk of injuries [Maurice et al., 2020].

3.3 Optimisation of the VSM-Parameters

To obtain a behaviour of the VSM that matches the desired behaviour as close as possible, numerical optimisation is used. MATLAB provides the command *fmincon*, which finds the minimum of a specified function with equality and inequality constraints. It uses an interior-point algorithm with a set of start values x_0 and linear and nonlinear constraints [MathWorks, 2021b]. The following overview is based on [MathWorks, 2021a]. The original optimisation problem

$$\begin{array}{ll}
\min_{x} & f(x) \\
\text{subject to} & h(x) = 0 \\
& g(x) \le 0
\end{array}$$
(3.8)

is being transformed for ease of solving. It becomes:

$$\min_{x,s} \quad f_{\mu}(x,s) = \min_{x,s} f(x) - \mu \sum_{i} ln(s_{i})$$

subject to
$$h(x) = 0$$

$$g(x) + s = 0$$

$$s \ge 0$$
(3.9)

By the introduction of s_i the constraints of the problem are transformed to equality constraints, which makes the problem easier to solve. μ is a factor that approaches zero for large iteration numbers and forces the design variable to approach the minimum. By default, the *fmincon* algorithm tries to do a Newton step first. If this fails, e.g. because the problem is locally not convex, a conjugate gradient method is used as a backup.

In the following the constraints are presented.

Constraints

• Dimension of l_3

To restrict l_3 from getting larger than 60mm and smaller than the diameter of a pulley and l_1 the following constraint is established. A distance of 1mm is enforced between the two pulleys and 3mm distance between the outer edge of the mechanism is defined.

$$l_1 + 2R \le l_3 \le 60mm - R - 3mm \tag{3.10}$$

• Dimensions of *l*₁

The length l_1 has to be restricted in a similar way to prevent it from becoming too small. The relation between l_1 and l_3 is defined in Equation (3.10).

$$R + 3mm \le l_1 \tag{3.11}$$

• Constraint for spring stiffness The stiffness of the spring has to be larger than ON-www and has no w

The stiffness of the spring has to be larger than 0Nmm and has no upper boundary.

$$0 \le k \le \infty \tag{3.12}$$

• Constraint for the pretension F_0

The cable that connects the input and output shaft cannot handle negative pretensions, therefore F_0 has to be positive at all points. Material limits are not taken into account at this point, so there is no upper boundary to the pretension.

$$0 \le F_0 \le \infty \tag{3.13}$$

• Size constraint for the pulleys

The size of the pulleys must not get too large, hence the radius *R* is restricted:

$$6.5mm \le R \le 15mm \tag{3.14}$$

· Length constraint of the spring

The overall length of the spring is restricted to fit between the shoulder joint and the elbow. This space is further reduced, because of the dimensions of the VSM. The final design of the VSM cannot be determined yet, but an estimate is

$$d = 2(l_3 + R) + 60mm \tag{3.15}$$

where an extra 60mm are added to account for design choices that might occur later. The maximum spring elongation Δl_c occurs when the angle between l_1 and l_3 is 180°, which results in

$$\Delta l_c = 2 \cdot 2l_1 \tag{3.16}$$

Combined with the spring elongation due to the pretension and the length of the upper arm as a maximum value from Table 3.1 the constraint is:

$$\frac{F_0}{k} + 4l_1 \le 281.7 - \frac{d}{2} \tag{3.17}$$

Objective Function

Ideally, the VSM shows the same torque behaviour as the shoulder in reverse direction. With the models for VSM and the shoulder torque in place, the basis for the objective function is taken to be the difference between the required torque, which is the shoulder torque, and the provided torque from the VSM. To have an effective torque of zero when the arm is at $\gamma = 0^{\circ}$, the angle θ of the VSM has to be periods of 180° . Since the VSM has to provide energy, the stored potential energy has to be higher when the arm is hanging down than when the arm is raised. Therefore if $\gamma = 0^{\circ}$, the internal angle of the VSM has to be $\theta = 180^{\circ}$. To obtain a more intuitive formulation of the objective function, it is made use of the point symmetry of the torque curve of the VSM and θ is related to γ

$$\theta = 180^{\circ} - \gamma \tag{3.18}$$

This results in a torque that is positive over the domain of the optimisation for the VSM and the shoulder model, although the torque resulting from the mass forces of the arm is of opposite sign than the torque of the VSM. In Equation (3.20) the deviation is hence a subtraction.

To also minimize the required prior elongation of the spring, the ratio between F_0 and the spring stiffness k is added as the exponent of an e-function. Thereby a large penalty is introduced if this ratio is large and configurations with small starting elongations l_0 are favoured. The error is squared to account for varying signs and cumulated over all elbow angles α and shoulder angles γ :

$$f(x) = \sum_{\gamma=0^{\circ}}^{180^{\circ}} \sum_{\alpha=0^{\circ}}^{150^{\circ}} \left(w \cdot (T_{shoulder} - T_{VSM}) \right)^2 + e^{\frac{F_0}{k}}$$
(3.19)

w is a weight function that is introduced to emphasize the influence of certain angles α and γ that are considered more important than others.

Weight Function

The initial purpose of the exoskeleton is to ease overhead working tasks. For those the elbow angle α is mostly between 10° and 90°. The higher angles to the end of the considered range are deemed less important and the maximum w.r.t. α is constructed at around 50°. To leverage the importance of the shoulder angle γ that are around 110°, a function that is quatratic w.r.t. γ is included. The specific parameters are found with a MATLAB optimisation. The weight function is

$$w(\gamma, \alpha) = \left(p_1 e^{p_2(\alpha+3)+p_3} + p_4 e^{p_5(\alpha+3)+p_6} + p_7 e^{p_8(\alpha+3)+p_9} + p_{10} e^{-\alpha-3}\right) \\ \cdot \left(-0.0002 \left(\gamma - 110^\circ\right)^2 + 3\right)$$
(3.20)

Parameter	Value	Parameter	Value	Parameter	Value	Parameter	Value
p_1	579.08	p_4	-0.04	<i>p</i> ₇	763.92	p_{10}	3.97
p_2	-0.05	p_5	0.23	p_8	-0.03		
<i>p</i> ₃	-1.72	p_6	7.06	<i>p</i> 9	0.32		

Table 3.2: Parameters $p_{1...10}$ of the weight function.

Table 3.2 contains the values of the parameters of the weight function. In Equation (3.20) the function is offset of 3° to the negative α -direction. This is to avoid a shift in the gradient at $\alpha = 0$ in Equation (3.20). Figure 3.5 shows the graphs of the main terms of the weight function. The complete weight function in the region of interest is depicted in Figure 3.6.



Figure 3.5: Graphs of the components of the weight function, 3.5a shows the weight function at $\alpha = 0^{\circ}$, 3.5b shows the graph at $\gamma = 0^{\circ}$



Figure 3.6: Weight function w in the area of interest

3.4 Result of the optimisation

Table 3.3 lists the results of the optimisation, which inlcudes the lengths l_1 and l_3 , the radius R, the stiffness of the spring k and the pretension F_0 .

l_1	l_3	R	k	F_0
[mm]	[mm]	[mm]	[N/mm]	[N]
18.0880	31.0880	6.5000	6.3350	9.9342

Table 3.3: Optimisation results; for the model those values are rounded to one decimal number



model and the VSM for various Torque Curves of the Shoulder

Figure 3.7: Torque curves of the shoulder model, the VSM model and 50% of the VSM model for α = $0^{\circ}, 45^{\circ}, 90^{\circ}, 135^{\circ}$; The curves have been adjusted to be in the same quadrant although they are of opposing signs.

Figure 3.7 displays the curves of the shoulder model and the VSM model. It also includes the graph of half the VSM torque. The signs of the curves have been adjusted to show them all next to each other although the torque due to gravity is of opposite direction that the torque of the mechanism. For the VSM to be effective, the angle θ has to be 180° when the arm is hanging down, that is at $\gamma = 0^{\circ}$, so that the energy, that is stored in the spring, is able to support the lifting process.

As can be seen from Figure 3.7, for a straight arm and shoulder angles up to around 60 degrees the VSM matches the required shoulder model very well. For higher shoulder angles γ with straight elbow the VSM torque shows a constant difference to the shoulder torque, but it follows the shape of the curve quite well. For elbow angles of $\alpha = 45^{\circ}$ the VSM matches the shoulder torque very well for shoulder angles between 80° and 170° . This is unsurprising, as the objective function was intentionally designed to fit this area best. The cost of the good fit for $\alpha = 45^{\circ}$ is, that with increasing elbow angles the VSM does not match the shoulder torque very well. It shows some drawback for $\alpha = 90^{\circ}$ and a significant difference to the torque with $\alpha = 135^{\circ}$. Those drawbacks are decreased significantly, if only half the shoulder torque is compensated. This provides the advantage, that the user does not need to put in extra effort to pull down the mechanism. This also reduces the expected weight and size of the design. Hence, in this thesis 50% support are implemented.

Chapter 4

Conceptual Design and Numerical Validation of the VSM

In this chapter, the preliminary design of the VSM and the assembly of the exoskeleton is described. Furthermore, a numerical analysis is carried out in order to validate the behaviour of the designed VSM. The numerical simulation is performed in the software MSC ADAMS. The analysis includes the maximum torque that the VSM can provide and also the angle that this corresponds with. Results regarding the deformation of the spring are described. In the end, a comparison of the numerical analysis with the analytical analysis described in Chapter 3 is done.

4.1 Preliminary Design

The initial design of the exoskeleton is based on the dimensions that are obtained from the optimisation and the selection of the extension spring. The exoskeleton is divided into several different assemblies which are explained in detail in this section.

4.1.1 VSM Assembly

Outer Frame

The dimensions of the *Outer Frame* are determined by the l_3 value obtained from the optimisation result and the dimensions of the selected spring. The *Outer Frame* has attachments for the pulleys, spring and the cuff. The design of the frame is done with intuition, by taking the forces from the spring and the pulleys into consideration. The *Outer Frame* is depicted in Figure 4.1.



Figure 4.1: Outer Frame

Inner disc

The dimensions of the *Inner Disc* are determined by the l_1 value obtained from the optimisation. The *Inner Disc* has attachments for the inner pulley and will be used to connect the exoskeleton to the scissor mechanism. There are also attachments, where it is connected with the housing assembly that is described later. The design of the *Inner Disc* can be seen in Figure 4.2.



Figure 4.2: Inner Disc

4.1.2 Scissors Assembly

SSM

The SSM is an important component of the exoskeleton as it replicates the range of motion of a shoulder joint and connects the VSM mechanism to the backplate. It is also responsible for transferring the torque generated by the VSM to the backplate. The design of the SSM can be seen in Figure 4.3.



Figure 4.3: SSM

Slot Link

The main purpose of the *Slot Link* is to keep the inner disc in a fixed position with respect to the SSM when the torque is applied. The slot provided in the link enables the sliding of the exoskeleton along with the movement of the end effector of the SSM. The *SlotLink* is depicted in Figure 4.4.



Figure 4.4: Slot Link

Connection Slot

The *Connection Slot* is used to connect the VSM Assembly to the Scissors Assembly. It is responsible for transferring torque produced by the VSM to the *Slot Link*. The design of the *Connection Slot* can be seen in Figure 4.5.



Figure 4.5: Connection Slot

4.1.3 Housing Assembly

The Housing Assembly is divided into the Inner Housing and the Outer Housing assemblies. The purpose of these is to hold the *Inner Disc* in position with respect to the *Outer Frame*. Two sleeve bearings, that are depicted with the black colours, are used to ensure compactness and smooth rotation of these two components.



Figure 4.6: (a) Outer Housing assembly (b) Inner Housing assembly

4.1.4 Exoskeleton Assembly

In Figure 4.7 the assembly of the parts described above can be seen. However, this is not the whole exoskeleton design as this also includes the spring, the backplate and the cuffs to connect it with the arm of the user.



Figure 4.7: Exoskeleton assembly

4.2 MSC ADAMS Approach

MSC ADAMS software is a modelling and simulating environment for analysing the behaviour of mechanical assemblies [Mscsoftware, 2010]. The software is used to examine kinematic motions that are induced by the action of applied forces on the system. The major advantage of Adams includes the ability to transfer loads and motion information from Adams to FEM software, e.g. ANSYS for stress analysis. ADAMS uses the system of Euler-Lagrange equation of motion. In order to understand how the software solves the equation numerically the two dimensional Euler-Lagrange equation is described below.

Assuming that the y-axis is in the vertical direction, the kinetic energy T and the potential energy V of a rigid body in two dimensions are given as:

$$T = \frac{1}{2}(m\dot{x}^2 + m\dot{y}^2 + I\dot{\theta}^2)$$
(4.1)

$$V = mgy \tag{4.2}$$

where *m* is the mass of the body, *I* is the rotational mass, (x,y) is the location of the centre of mass in the fixed rectangular coordinate system, θ is the orientation of the body around the x-axis and *g* is the gravitational acceleration.

The difference between the kinetic and potential L = T - V energy is called the Lagrangian of the dynamical system and for more than one bodies is formulated as:

$$L = \sum_{j=1}^{N} (T_j - V_j)$$
(4.3)

where N is the number of bodies in the system.

The Euler-Lagrange equation for a multi-body system, which describes its motion is formulated as:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}}\right) - \frac{\partial L}{\partial q} + \Phi q^T \lambda = Q_{ex}$$
(4.4)

where q is the column matrix of generalized coordinates, λ represents the Langrange multipliers and Q_{ex} is the vector of generalized external forces acting along the coordinates q. Φ_q is the Jacobian matrix of the constraint equations, which can be written as:

$$\Phi_q = \frac{\partial \Phi}{\partial q} \tag{4.5}$$

An example of a simple pendulum and how ADAMS software is using the Lagrange equation can be seen in Appendix E.

4.3 Design of the VSM

For the design of the VSM the dimensions, that used are those that found in Chapter 3 and more specific in Table 3.7. The diameter for the cable is chosen 2mm. The VSM that is used in MSC ADAMS can be seen in Figure 4.8. The start of the cable is attached to a fixed part and the end is attached to a spring. The number of pulleys is six, those are fixed to the outer (dark grey) and inner (light grey) parts. The outer part is allowed to rotate around the z-axis and the inner part is fixed to the ground.



Figure 4.8: The VSM model in MSC ADAMS

In order to initiate a motion of the mechanism a preload has to be applied to the spring. The value of the preload, which corresponds to the maximum spring elongation, has to be calculated. In Figure 4.9 the VSM can be seen, although only the two disks are displayed for simplicity - the inner disk (light grey) and the outer disk (dark grey). The blue circles represent the pulleys, where also the cable that connects them can be observed. l_1 is the distance from the center of the inner disk to the center of the pulley in the inner disk. The distance from the center of the two upper pulleys to the center of the pulley in the inner disk is the length l_2 . The distance between the center of the two upper pulleys and the center of the inner disk is the length l_3 . The angle between the lengths l_1 and l_3 is represented by the angle θ . In Figure 4.9a, the angle θ is equal to 180°, which happens when the arm is in the initial position stretched down and in Figure 4.9b the angle θ is equal to 0°, which occurs when the arm is elevated 180° from hanging down.



Figure 4.9: a) VSM sketch when $\theta = 180^{\circ}$ and b) VSM when $\theta = 0^{\circ}$

The maximum force F_{max} of the spring which corresponds to the maximum spring elongation l_{2max} is equal to:

$$F_{max} = k l_{2max} \tag{4.6}$$

where k is the stiffness of the spring.

The maximum spring elongation l_{2max} is equal to the prior elongation $l_{2,0}$ of the spring, which happens when $\theta = 0^{\circ}$, plus the change in the elongation of the spring Δ_{lc} when the angle θ is equal to 0° and 180° :

$$l_{2max} = l_{2o} + \Delta_{lc} \tag{4.7}$$

where $l_{2,0} = \frac{F_0}{k}$.

The change in the elongation of the spring Δ_{lc} can be derived geometrically from Figure 4.9 as:

$$\Delta_{lc} = 2(l_3 + l_1) - 2(l_3 - l_1) = 4l_1 \tag{4.8}$$

Using the optimisation results from Table 3.3 and substituting in the above equations it can be found that the maximum elongation $\Delta_{lc} = 72mm$ and the corresponding force $F_{max} = 239N$.

4.4 **Results of the Numerical Analysis**

From the numerical analysis of the VSM, the important results are the torque profile, the change in length of the spring, the cable tension and the forces that are applied in the pulleys.



The results of the analytical and numerical analysis of the VSM can be seen in Figure 4.10.

Figure 4.10: Comparison of analytical and numerical results of the VSM model for the provided torque

From the figure above it can be seen that both the analytical and the numerical analyses give similar results. The peaks of the resulting curves are listed in Table 4.1. The difference between the peak loads is 102Nmm or 2% of the analytical result. The reason for this deviation could be, the diameter of the pulley in ADAMS can only be set in certain standard increments which result in differently dimensioned pulleys causing a discrepancy of the resultant force in them. Also, ADAMS requires mass to calculate the Lagrange equation. The mass used was close to zero, whereas in the analytical results the mass of the model is not taken into account. Regarding the angle that the maximum torque is provided at, the difference of the two analyses is at around 1.7° . This corresponds to a deviation of 1.717% compared to the peak angle of the analytical model.

	Max Torque [Nmm]	Angle [°]
Analytical	1803 36	00
solution	4095.50	-99
Numerical	4701 15	07.3
solution	4791.15	-97.5

Table 4.1: Results of the two analysis for the maximum torque and angle

The cable tension with respect to the angle of rotation θ of the numerical analysis is also compared with the one of the analytical analysis. The results of the comparison can be seen in Figure 4.11.



Figure 4.11: Comparison of analytical and numerical results of VSM model for the cable tension

From the comparison of the two analyses it can be seen, that the results regarding the cable tension are similar. Since in the numerical analysis the simulation is done from the fully loaded spring, it is important to confirm that the result at $\theta = 0$ matches the analytical solution. The results can be seen in Table 4.2, where the relative percentage difference of the two analyses is very small and equal to 0.17%.

	Angle [°]	Cable tension [N]	
Analytical	0	0.0342	
solution	0	9.9542	
Numerical	0	0.0160	
solution	0	9.9109	

Table 4.2: Results of the two analysis for the tension of the cable when the angle is 0°

In Figure 4.12 it can be seen how the spring deforms over the angle of rotation θ . From the comparison of the two analyses it can be concluded that results are alike.



Figure 4.12: Comparison of analytical and numerical results of VSM model for the spring deformation

In Table 4.3, the results for the spring deformation can be seen when the angle θ is equal to 0°. The relative percentage difference of the two analyses is equal to 6.79%.

	Angle [°]	Spring deformation[mm]	
Analytical	0	3 1362	
solution	0	5.1302	
Numerical	0	2.03	
solution	0	2.93	

Table 4.3: Result of the two analysis for the change in the spring length when the angle is 0°

From the figures above it can be concluded that the numerical analysis of the VSM model gave similar results as the analytical model that is explained in Chapter 3. The export of the forces from the MSC ADAMS software is analysed next.

4.4.1 Export of the Forces

In order to do the stress analysis of the VSM, the forces are exported from ADAMS and used in ANSYS Workbench. The maximum forces that the VSM experiences occur when the $\theta = 180$ as in this position, the spring provides the maximum force. In Figure 4.13, it can be seen the forces that applied in the pulleys and the two anchor. These forces are used

in Section 5.2 for the analysis of the structure.



Figure 4.13: Forces exported from MSC ADAMS

It is difficult to find the spring with the same property as obtained in the optimisation result in Chapter 3. Therefore, a spring was selected with similar properties, which can be seen in Table 4.4.

Motorial	Initial length L _o	Max length	Max Force	Preload	Stiffness	
Material	[mm]	[mm]	[N]	[N]	[N/mm]	
Piano wire	127	204.98	301.89	27.18	3.52	

Table 4.4: Spring properties that used in the VSM

The torque provided from the VSM is found and compared with the analytical results. Both results use the properties of the available spring. Furthermore when $\theta = 180^{\circ}$, the elongation of the spring is 72mm and so the maximum force of the spring in this position is $F_{max} = 280.62N$. This means that the maximum force of the spring is greater than the one that was found in the optimisation, where the stiffness of the spring was equal to 3.1675N/mm and, including the pretension F_0 , the maximum force of the spring was found to be 239N for $\theta = 180^{\circ}$. However, as mentioned before, it is not easy to find the exact stiffness and the pretension that the optimisation part gave. The comparison of the two analyses can be seen in Figure 4.14.



Figure 4.14: Comparison of analytical and numerical results of VSM model for the provided torque using the new spring properties. The maximum deformation of the spring is 72*mm*.

From the comparison of the results, it can be seen that both analyses give similar results and so the forces from ADAMS can be exported to do the stress analysis. The magnitude of the forces can be seen in Table 4.5. Only the magnitude of the forces is included as the direction of the forces can be seen in Figure 4.13.

Force	<i>F</i> ₁	<i>F</i> ₂	F ₃	F_4	F_5	F ₆	F ₇	<i>F</i> ₈
Magnitude [N]	280.62	545.38	561.24	524.27	228.31	352.57	376.22	280.62

Table 4.5: Magnitude of the exported forces from MSC ADAMS for the deformation of the spring at 72mm

The maximum elongation of the provided spring is 77.98mm. This means that, besides the results of the optimisation for a maximum deformation of the spring of 72mm, the spring can provide more force when the deformation of the spring increases. This can provide the user with increased assistance if needed. It was chosen for safety reasons to examine the forces acting on the VSM at a spring elongation of 77.98mm, which is the maximum. The magnitude of the forces for the maximum elongation of the spring can be seen in Table 4.6.

Force	<i>F</i> ₁	<i>F</i> ₂	F ₃	F_4	<i>F</i> ₅	F ₆	F ₇	<i>F</i> ₈
Magnitude [N]	301.87	584.12	603.34	562.33	245.77	370.73	387.82	301.67

 Table 4.6: Magnitude of the exported forces from MSC ADAMS for the maximum spring elongation of

 77.98mm

Spring deformation	Maximum Torque	Pretension Fo
[mm]	[Nmm]	[N]
72	5907.94	27.18
74	6114.92	34.22
76	6323.83	41.26

Table 4.7: Maximum torque that the spring can provide in the three different levels 72*mm*, 74*mm*, 76*mm* and the pretension of the spring.

Chapter 5

Design and Construction of the Exoskeleton

In this chapter, the design loads calculated from the numerical analysis from MSC ADAMS are used to perform a structural analysis of the structure. Using the results the design is further optimised to reduce the weight of the structure. For this purpose topology optimisation in ANSYS is used.

5.1 Material Selection

To select materials that can be used in the exoskeleton, the stresses are crucial. If possible, light materials should be used to reduce the weight of the structure. However, those tend to increase the cost of the structure. Some options of the material that can realistically be used are presented below.

It is possible to build the components of the exoskeleton from composites. However, this class of materials is not elaborated on, because it seems unfitting in such an early stage of development because the production of composite parts is very cost-intensive.

Polymers

Polymers can reduce the cost and also the weight of the structure in a significant way. Some of the polymers that can be used are Acrylonitrile Butadiene Styrene (ABS) and Polylactic acid (PLA). Both materials have similar cost and strength, but the PLA is more preferable for prototyping because of its ease of printing, aesthetically appealing parts, and high accuracy [Feeney, 2019]. Both materials can be used for 3D printing. The mechanical properties for the two materials can be seen in Figure 5.1. However, the mechanical properties vary drastically depending on how it is manufactured [Farah et al., 2016]. It was decided to make the pulleys, the inner and the outer housings from PLA as in this way the final weight and costs are reduced.

Matarial	Density	Young's modulus	Tensile strength	
	[kg/m^3]	[GPa]	[MPa]	
ABS	1000-1400	2-2.6	37-110	
PLA	1300	3.5	50	

Table 5.1: Mechanical properties of PLA and ABS [Ltd, 2021]

Metals

When high strength and stiffness are required metals alloys are necessary. Steel is the commonly used metal because of its good mechanical properties and the large variety of strength. However, if choosing a steel with higher strength also the cost are increased. Aluminium is a good choice because of its low density. It can replace steel and provides a lighter structure without compromising strength. Compared to steel, aluminium can deform and bend more easily. Other commonly used lightweight alloys are titanium or magnesium alloys. Due to their high cost, those are disregarded here. It is decided to use high strength Aluminium 7075-T6 for the *Slot Link*. For ease of machining and welding, it is decided to use Aluminium 5083-O for the outer frame and the inner disk. To keep the friction low and allow the *Connection Slot* to slide inside the *Slot Link*, it is decided to make the *Connection Slot* from brass. The material used for the SSM provided by AAU is EN-1.4404.

5.2 Finite Element Analysis

In this section the structural analysis of the Scissors Assembly and VSM Assembly will be carried out. The software used for these analysis is Ansys Workbench. The analyses conducted are linear static structural simulations.

Material Properties

Ansys requires material parameters to conduct the analysis. The property of EN-1.4404, Aluminium 7075-T6 and Aluminium 5083-O, that are used for the analyses, are mentioned in Table 5.2.

Matarial	Density	Young's Modulus	Poisson's	Yeild Strength
Wateria	[kg/m^3]	[GPa]	Ratio	[MPa]
EN-1.4404	8000	200	0.3	220
Aluminium 7075-T6	2810	71.7	0.33	503
Aluminium 5083-O	2660	71	0.33	145

Table 5.2: Material property data is taken from [MATWEB, 2021]

Safety Factor

It is always important to consider a proper Factor of safety (FOS). As a general recommendation for commercial aircrafts, which is a structure where weight and safety are both of crucial importance, a FOS of 1.2 - 1.5 is recommended [Norton, 2010]. Since there is no specific FOS for exoskeletons, a safety factor of 1.3 is chosen. This value is valid because it is within the range of an aircraft, but the risk for the users of the exoskeleton is less than in the aviation industry. The von Mises failure criterion is used for the analysis. For Aluminium 7075-T6 the maximum allowable stress including the FOS is 386MPa, for Aluminium 5083-O the stress is 111MPa and for EN-1.4404 the maximum allowable stress is 169MPa.

5.2.1 Structural Analysis of Scissors Mechanism

The Scissors mechanism is designed to transfer the torque from the VSM to the backplate. The model is simulated in a way that approximately represents real-life loading and boundary conditions.

Loading and Boundary condition

The maximum torque calculated for the maximum spring deformation at 77.98mm using the analytical solution is found to be 6532Nmm. Three different loading condition are considered where the position of *Connection Slot* is changed to 8°, 59° and 82°, these configura-

tions are referred to as Open-Scissors, Middle-Scissors and Closed-Scissors respectively in the following section. The angle ω that defines these configurations can be seen in Figure 5.1.

For boundary condition, a moment is applied on the outer face of the *Connection Slot* to simulate the torque generated by the VSM. The left end of the SSM, which is connected to the backplate, is simulated by applying cylindrical boundary condition which is fixed in x, y and z rotation as shown in Figure 5.1.



Figure 5.1: Moment and cylindrical boundary conditions that are applied to the SSM. The angle ω can also be seen.

Element and Contact Definitions

The elements selected for these studies are the default options of Ansys. However, a hexdominant mesh is selected so that the majority of elements used in the simulation are SOLID 186 and SOLID 187 which are 3-D 10 node and 20 node quadratic elements. They are very well suited for simulating 3-D structures with irregular shapes, which are produced because of complicated CAD models.

To simulate the joints between the links of the SSM, the MPC 184 element is used. It is a multipoint constraint element that is used to apply kinematic constraints between two bodies having rotation or sliding movement. To simulate the contact between the *Slot Link* and the *Conncetion Slot*, CONTA 174 elements are used, which is a 3-D eight-node contact element. The contact definition is selected as frictionless as it is more computationally efficient and gives conservative results.

Stress Result for Scissors Mechanism

Due to contacts and fixed boundary condition used in the simulation, there are stress concentrations developed in these areas. According to the *Saint-Venant's* principle, the effect of local disturbances to the uniform stress field remains local. The results away from these disturbances are not affected. To get the correct stress values, new sections are created for each link of the SSM, which are 4mm away from the point of stress concentration as shown in Figure 5.2. For the *Slot Link*, the stresses are probed away from stress concentration caused by the contact faces of the *Connection Slot* to the *Slot Link*. The convergence plots for all the results below are shown in Appendix F.



Figure 5.2: SSM showing different section in the links that are selected to probe for maximum stress

Open-Scissors Configuration

The stress distribution for the Open-Scissors configuration can be seen in Figure 5.3. In Table 5.3, the maximum equivalent stresses for the 6 links and the *Slot Link* are observed.



Figure 5.3: Stress distribution for the Open-Scissors configuration

Scissor Link	Maximum Stress (MPa)
1	121.66
2	116.49
3	560.22
4	517.75
5	464.1
6	516.26
Slot Link	226.12

Table 5.3: Maximum stress for the 6 links and the Slot Link for the Open-Scissors configuration

Middle-Scissors Configuration

The results of the stress analysis for the Middle-Scissors configuration can be seen in Figure 5.4 and also the maximum equivalent stresses for the 6 links and the *Slot Link* in Table 5.4.



Figure 5.4: Stress distribution for the Middle-Scissors configuration

Scissor Link	Maximum Stress (MPa)		
1	43		
2	43		
3	480.94		
4	498.12		
5	498.01		
6	490.29		
Slot Link	272.86		

Table 5.4: Maximum equivalent stresses for the 6 links and the Slot Link for the Middle-Scissors configuration

Closed-Scissors Configuration

For the Closed-Scissors configuration the stress distribution can be seen in Figure 5.5 with the maximum equivalent stresses in Table 5.5.



Figure 5.5: Stress distribution for the Closed-Scissors configuration

Scissor Link	Maximum Stress (MPa)
1	150
2	138
3	227.19
4	279.27
5	270.17
6	252.63
Slot Link	161.89

Table 5.5: Maximum stress for the 6 links and the Slot Link for the Closed-Scissors configuration

Conclusion

From the results above it is seen that the stresses in the *Slot Link* is below the maximum allowable stress. However, the stresses in the links of the SSM are above the yield strength of the material used. Therefore, either another SSM with stronger material and different dimensions that can handle the stresses can be produced or a weaker spring can be used. Since in this project the time has been limited, the option of a weaker spring has been chosen. The properties of the new spring are shown in Table 5.6. The maximum torque produced by this spring is found to be 1912*Nmm* using the analytical model. The maximum equivalent stresses in the Scissors mechanism for this load are given in Table 5.7.

Material	Initial length L _o	Max length	Max Force	Preload	Stiffness
	[mm]	[mm]	[N]	[N]	[N/mm]
Piano wire	114	204.72	83.63	12.70	0.84

Table 5.6: Spring properties of the weaker spring

	Scissors-Open	Scissors-Middle	Scissors-Close
Link	Maximum Stress (MPa)	Maximum Stress (MPa)	Maximum Stress (MPa)
1	35	12	43
2	33	13	41
3	163.95	140.82	67
4	151.58	145.79	82
5	135.77	145.92	79.29
6	151.18	143.34	73.57

 Table 5.7: Maximum stresses in each link of the Scissors using a weaker spring

From the results for the six links for the three configurations of the SSM, as expected the stresses are reduced. Since the material that the provided SSM is made off is EN-1.4404 and the maximum allowable stress is 169MPa it can be concluded, that the SSM with the weaker spring can withstand the stress.

5.2.2 Structural Analysis of VSM Assembly

In this section the structural analysis of the VSM Assembly is described. The structural analysis includes the the *Outer Frame* and the *Inner Disc*. It has to be noted that for the structural analysis of the VSM Assembly the load results for assistance with half the shoulder torque is used and not the one for the weaker spring. The reason for not using the weaker spring, as for the analysis of the SSM, is, that if the VSM can hold the stresses for the original spring then only the SSM has to be changed.

Analysis of Outer Frame

The model is simplified by removing the pins and the pulleys. All the loads used in the simulation are taken from MSC ADAMS in Chapter 4. A fixed boundary condition is applied at the side where the Inner Housing Assembly is in contact with the *Outer Frame*. A bearing

type load is applied on the faces of the holes where the pins are used and a force load is applied on the hole where the spring is attached to the VSM. The loading and boundary conditions are represented in Figure 5.6.



Figure 5.6: Boundary conditions used for the analysis of the Outer Frame

The elements used for this analysis is the default option selected in Ansys which is SOLID 187 elements. A mesh convergence study is carried out to verify the convergence of stresses. This is done iteratively by changing the mesh size in the Ansys convergence criterion for the von Mises stresses. The maximum allowable stress change between each iteration was set to 5%, the results are shown in Table 5.8.

Ittoration	Number of	Maximum Stress	Change
	Elements	(Mpa)	(%)
1	13041	36.68	
2	42216	53.673	38.68
3	101577	53.682	0.0015

Table 5.8: Convergence study results for the outer frame

The stress distribution of the VSM is shown in Figure 5.7. The maximum stress is found to be 53.68*MPa*.



Figure 5.7: Stress distribution of the Outer frame

Analysis of Inner Disc

The load acting on the pulley attached to the *Inner Disc* is found to be 603.34N in Chapter 4. For simplification the pulley load is replaced by a bearing load and applied to the model. The back end of the *Inner Disc* is fixed as it is attached to the *Connection Slot*. The boundary conditions can be seen in Figure 5.8.



Figure 5.8: Boundary conditions for the Inner Disc

The elements used for this analysis is the default option selected in Ansys which is SOLID 187 elements. A mesh convergence study is carried to verify the convergence of stresses. The results are shown in Table 5.9.

The maximum stress for the *Inner Disc* is found to be 29.37*MPa*. The stress distribution is seen in Figure 5.9.

Ittoration	Number of	Maximum Stress	Change
Itteration	Elements	(MPa)	(%)
1	3823	25.923	
2	9665	27.52	5.97
3	45161	29.37	3.05

Table 5.9: Convergence study results for the Inner Disc



Figure 5.9: The stress distribution of the Inner Disc

Conclusion

The structural analysis of the VSM Assembly shows that the stresses are very low in both, the *Outer Frame* and the *Inner Disc* as compared to the yield strength of the material. Therefore, to reduce the weight of the component it is decided to carry out topology optimisation on the *Outer Frame* as this part contributes significantly to the weight of the whole assembly.

5.3 Topology Optimisation

The following section is based on [ANSYS, 2021] unless marked differently. Topology optimisation computes an optimal design of a structure, where design objectives and constraints are specified. It can also be implemented to specific regions that the user desires to change. In this project, the topology optimisation tool in Ansys is used. To conduct the topology optimisation there are three ways:

- Level Set Based Topology
- Lattice Optimisation
· Density Based

The first method is based on the boundaries of the shapes. The design variable for this method are the boundaries of the structure and they are adjusted to optimise the structure. The lattice optimization method computes an optimal variable density lattice distribution in the geometry. It is a useful method for optimisation of 3D printed parts. The third method is based on the density of each element. In this project the density based method is used and so only this is elaborated on.

5.3.1 Density Based Method

In order to increase the stiffness of the structure, the strain energy has to be reduced. This is done by assigning a pseudo-density to each element. An approach for the topology optimization problem is the Solid Isotropic Material with Penalization (SIMP) approach. In this approach the design variables ρ_i , where *i* are the elements, are the element densities that are assigned. The value of the density can be varied from 0, which means that the element is removed, to 1, where the element is preserved. Generally, the original problem consists of 0 or 1 density elements. However, this problem can not be solved because of the large search-space containing N!/((N - M)!M!) possibilities, where N is the number of material elements and M the number of all the elements [Nobel-Jørgensen, 2016]. It can be easily concluded that when the number of material elements is increased, the time to solve the problem rises exponentially. The SIMP approach allows using intermediate values between 0 and 1, which reduces the search space.

General Topology Optimisation Problem

In general, the topology optimisation problem is seeking to minimise or maximise the objective function. The objective function can be the compliance, the natural frequencies etc. The mathematical optimisation problem can be written as [Kohnke, 1999]:

Maximize or **Minimize** : f

Subjected to :
$$0 < \rho_i \le 1$$
 $(i = 1, 2, ..., M)$
 $g_{jl} \le g_j \le g_{jr}$ $(i = 1, 2, ..., N)$

where M is the number of elements, g_j is the computed j_{th} constraint value, g_{jl} and g_{jr} are the lower and upper bounds of the j_{th} constraint and N is the number of the constraints.

Topology Optimisation for Compliance Minimisation

For this project, the aim of the topology optimisation that is carried out is to minimise the compliance of the structure i.e. to maximize the stiffness. The general mathematical optimization can be rewritten now as [Lund et al., 2019]:

Minimize : C(U)

Subjected to :
$$0 < \rho_i \le 1$$
 $(i = 1, 2, ..., M)$
 $K = UF$
 $V^* = aV$
 $E_i = \rho_i^p E^*$

where C(U) is the compliance of the structure, K is the global stiffness matrix, U is the global displacement vector, F is the global force matrix, V^* is the material volume, V is the design domain volume, a is the volume fraction of available material, E_i is the Young modulus of each element, p is a penalty factor and E^* is the Young modulus for $\rho = 1$.

The volume fraction *a* is the amount of the desired reduction of the volume of the structure. The penalty factor is used to reduce the greyscale elements [Nobel-Jørgensen, 2016], where greyscale elements are the elements that have a density between 0 and 1. Using the penalty factor these elements become closer to dark or white colours and in this way the final design is more clear and easier to be manufactured. Typically the value for this penalty factor is 3 [Sigmund, 2001].

The compliant problem can be solved either by using Optimality Criteria (OC) methods or gradient-based algorithms such as Sequential Linear Programming (SLP) and Method of Moving Asymptotes (MMA). The Optimality Criteria is an iterative solver, where the design variables on each point are updated based on the optimality conditions. In the points where the strain energy is high, material is added. The default solver in ANSYS is the Sequential Convex Programming (SCP), which is an extension of MMA. MMA is a nonlinear programming algorithm where the solution is approximated by solving a sequence of convex and separable sub-problems. It is an inexpensive method since the sub-problems are more easily solved [ANSYS, 2021].

5.3.2 Topology Optimisation of Outer frame

From the previous section, it is seen that the stresses in *Outer Frame* are very low as compared to the allowable maximum stress. The outer frame also contributes to the significant percentage of weight to the exoskeleton therefore it is most efficient to modify the design and reduce the weight of the part using topology optimisation. In this section, the initial parameters required for setting up the analysis is discussed.

Simulation Setup

The loading and boundary conditions that are used are the same as in Section 5.2.2. The results of the topology optimisation are sensitive to the mesh and the element type used in the model. The analysis setting of the simulation was set up according to the recommendation in Article [Sotola et al., 2021]. A very mesh size of 1mm with SOLID 186 elements is used in the analysis to get good results. The default options for the penalty factor of 3, convergence criteria of 0.1% and solver setting of Sequential Solver Programming is used.

Optimisation Objective

The optimisation objective function selected for the study is minimising the compliance. It is used because compliance being a global variable has a more convex solution and the solution will result in a global optimum instead of a local optima [ANSYS, 2021].

Response Constraint

The response constraint is set to mass as the objective of the study is to reduce the weight of the structure. Ansys has an option to adjust the value of the percentage of mass to retain. For this study, the mass constraint has been set to 30% retention. This ensures reduction of a significant portion of mass without sacrificing more of the structural integrity and also the computational time of the study.

Optimisation Region

The optimisation region selection gives an option to discard certain parts or faces on the model which does not need to be optimised. The sections in the *Outer Frame* which are being loaded directly by spring tension and the section where the cuff is attached are decided to be excluded from the optimisation study. This is done to avoid complex geometries and



the removal of material from these sections. The optimised section are represented in blue and the excluded regions are represented in red as seen in Figure 5.10.

Figure 5.10: The sections that are optimised (blue colour) and the sections that remain the same (red colour) in the *Outer Frame*

5.3.3 Optimisation Results

The topology optimisation is carried out by selecting all the parameters mentioned above and setting the maximum iteration to 50. In Figure 5.11 the grey region represents the material that is to be retained, the grey region represents the material that is needed to be removed partially and the red region represents the material that is to be removed completely. The excluded regions are kept intact. It is observed that most of the material is removed from the bottom and the side part of the *Outer Frame*. This implies that the material in these parts does not contribute to the stiffness of the *Outer Frame*. However, the optimised part from Ansys is still very complex to manufacture. It is decided to use the optimised result as the basis for designing a lighter *Outer Frame* that is also easy to manufacture.



Figure 5.11: Optimisation result representing retention of 30% material of the Outer Frame

5.3.4 Optimised Design

The Optimised design is created in Solidworks with the focus on making the component lightweight, structurally stable and easy to manufacture. The design is made with intuition by taking the result from topology optimisation as the foundation. As the component has large dimensions and complex geometry, it is decided to split the part into different subparts so as to make manufacturing easy. The updated *Outer Frame* can be seen in Figure 5.12. The mass of the part was reduced significantly to about 70% from 0.46612 kg to 0.14266 kg.



Figure 5.12: Optimised design of Outer Frame

Stress result for optimised design

The boundary conditions applied are the same as discussed in subsection 5.2.2. The stress distribution for the optimised design can be seen in Figure 5.13. It is observed that the overall stress has increased slightly than the initial design, however, the maximum stress is still below the allowable limit of 111 MPa. The convergence of the von Mises stresses is done iteratively by changing the mesh size in Ansys and the results are shown in Table 5.10.

Iteration	Number of	Maximum Stress	Change
	Elements	(MPa)	(%)
1	5492	62.58	
2	19855	83.8	29
3	32250	83.86	0.007

Table 5.10: Convergence study of maximum von Mises stresses in the optimised Outer Frame



Figure 5.13: Von Mises stress distribution of the optimised Outer Frame

All the engineering drawings for the designed parts can be found in Appendix G.

Chapter 6

Additional Design Ideas

The following chapter presents a collection of ideas and possibilities to implement in order to include a second DOF in the assistance of the exoskeleton. This DOF represent the abduction-adduction movement. The first idea describes the possibility of having two VSM in the exoskeleton connected with one cable. The other idea is to have only one VSM but it can be shifted and adjusted to the necessary movement. In the end, a backplate mechanism is presented to ensure that the mechanism is put in the right position and can provide the maximum torque.

6.1 Addition of a Second Degree of Freedom

The preliminary design as presented gives assistance for motion in the sagittal plane only. In this section, the addition of one more assisted plane is explored.

Simply adding a second torque providing device is not a sufficient solution, because the sagittal and the coronal plane are not independent of each other. In other words, if the arm is lifted in one plane, kept in an elevated position and moved to the other plane, the first torque providing device has to go back to its fully loaded condition while the second device has to become active. Additionally, the SSM does not behave like a rigid link, but compresses after a certain range of pure rotation around the fixed point at the backplate. The behaviour of the SSM also depends on the positioning of the joint at the backplate and the joint connected to the VSM, for motion in the sagittal plane. In the following, two different approaches are presented, which work around this problem in different ways.

6.1.1 Two VSM One Cable

In order to work around the problems with two independent torque providing mechanisms, one solution might be to connect the two compliant elements. The VSM as described in Chapter 2 is a desirable mechanism, as its torque varies and can be adjusted not only to the motion in the sagittal plane but also in the coronal plane. Thus in the following, an exoskeleton with two VSMs, one for the sagittal and one for the coronal plane, is suggested. Another advantage of using the same mechanism twice with different parameters is that it can be easily connected. In the present case, such a connection would be to guide the cable from one VSM to the second one. In this manner the cable forces of both VSMs are equal and the number of compliant elements can be reduced to one.

The advantage of such a configuration is, that from a position where the arm is hanging down with all angles $\theta_{1...7} = 0^{\circ}$ from Figure 3.1, both, motion in sagittal and coronal plane are supported from the beginning. However, difficulties arise when a mixed motion of some angle between the two planes is supposed to be supported. Because the cable tension is equal for both VSMs, the torque is influenced by the dimensions, that is l_1 and l_3 . Since the combined torque of both VSMs needs to be composed differently for varying angles between the two planes, the dimensions l_1 and l_3 need to adjust themselves automatically to perfectly balance the weight of the arm. This is very difficult to implement purely passively, but is possible to implement in a pseudo-passive exoskeleton. Figure 6.1a shows the x_m -, y_m - and z_m -component of the torque vector produced by the weight of the arm if the arm is being rotated from the sagittal to the coronal plane at an elevation angle of 90°. Figure 6.1b shows the absolute torque as described in Section 3.4 for a VSM that is rotated with the arm as a function of the shoulder angle γ and the length l_1 .



Figure 6.1: (a) Composition of the torque vector for a transition of the elevated arm from the sagittal to the coronal plane, the torque vector is described in the main coordinate system introduced in Section 3.1.
(b) Absolute torque of a VSM for varying l₁ and the shoulder angle γ; the fixed dimensions are the ones obtained with the optimisation described in Section 3.3.

(c) Absolute torque of a VSM with the dimensions as obtained in Section 3.4 for selected shoulder angles γ as a function of l_1 .

As expected the z_m -component is 0Nmm for the entire range and the shape of the nonzero components follows a sine and cosine curve, respectively. As it can be seen in Figure 6.1a there is no linear relation between the x- or y-component of the required torque and the variation of the torque produced by a change of l_1 . This is especially clear when a look at Figure 6.1c is taken, which documents the change in absolute produced torque over $l_1 = 0mm...18mm$ for selected shoulder angles γ .

Adjustment of *l*₁

To produce the required torque behaviour, the governing differential equation has to be solved for l_1 with a certain torque input. This is not a trivial task, because l_1 influences the torque only indirectly by defining the length l_2 . Therefore the governing differential equation of the VSM torque is solved numerically for l_1 by comparing the required torque with the full range of possible torque values for a certain angle γ . The resulting curves for l_1 over ξ , which describes the angle between the sagittal plane and the plane of motion, is presented in Figure 6.2.



Figure 6.2: Curves for the length l_1 over ξ for selected elevation angles

Note that the above Figure 6.2 is only valid for the x-component of the torque. The y-component shows the same behaviour but is flipped so it starts with zero length. Further elaboration on the required length l_1 as well as the consequences for the active component can be found in Appendix H.

Connection between the two VSM

With the torque of each of the two individual VSM now being able to adjust to the requirement, the problem of connecting the two mechanisms physically remains. The most compact solution is to use the SSM as described in Section 2.2. To be able to apply a force on the arm, the second VSM, which counters the y-component of the torque of the weight of the arm, has to transfer torque to the SSM. This can be done with the same mechanism as used in the design of the prototype for support in the sagittal plane, that is the *Slot Link* presented in this report. To prevent the two *Slot Link* from interfering with each other, one has to be moved to a greater radius.

Another problem that needs to be solved, is that if the cable is guided directly from one VSM to the other, the SSM is contracted. This happens, because the rope changes direction at the beginning and the end of the SSM to power the corresponding VSM. Thereby, a force is acting on the SSM that is not countered and leads to contraction. For the user of the exoskeleton, it makes this uncomfortable to use and will try to twist the arm outside, corresponding to a torque that acts in positive θ_3 angle as defined in Section 3.1. To counter the contracting force in the horizontal plane of the SSM, the cable can make a loop in the vertical direction around two joints. This will produce a force to elongate the SSM. In order to determine the ratio of the two forces that prevents motion, a one-quarter of the SSM is investigated. Figure 6.3 shows the mechanical system as it will be used in the following. The system is supported by a fixed support A and a floating support B. Both links have the same length *a* and the same angle $\frac{\theta}{2}$ to the horizontal.



Figure 6.3: (a) Quarter of the SSM with boundary conditions and forces of the cable (b) Free body diagram of one link, the cut is made in the joint

From Figure 6.3a, it can be concluded that

$$F_{Ay} = F_B = \frac{1}{2}F_y \tag{6.1}$$

$$F_{Ax} = \frac{1}{2}F_x \tag{6.2}$$

The equilibrium of moments of the first link as displayed in Figure 6.3b around the fixed support A is

$$\sum M_A = 0 = a \left(F_y \cos \frac{\varphi_2}{2} - F_B \cos \frac{\varphi_2}{2} - \frac{F_x}{2} \sin \frac{\varphi_2}{2} \right)$$
(6.3)

and it follows with Equation (6.1), that

$$\frac{F_y}{F_x} = \tan\frac{\varphi_2}{2} \tag{6.4}$$

The range of φ_2 is determined from Equation (2.11) and Table 2.1 and ranges from $\varphi_2 = 5.4^{\circ}...167.6^{\circ}$. Figure 6.4 displays the range, which the ratio of F_y to F_x has to have over the range of φ_2 .



Figure 6.4: Ratio of F_y/F_x so that the SSM is stabilised

As it is seen, stabilisation of the SSM over the whole range is impossible to be achieved, because equation (6.4) approaches infinity for $\varphi_2 \rightarrow 180^\circ$. However, a ratio of $1 \le \frac{F_y}{F_x} \le 2$ is obtainable with a vertical loop around two joints as shown in Figure 6.5.



Figure 6.5: SSM with continuous cable from one VSM to the other with a vertical loop for stabilisation

The two pulleys that enforce the 180°-change of direction in the vertical take two times the cable force which correlates to F_y . To get a ratio of $\frac{F_y}{F_x} = 1$, the VSMs must be attached horizontally, so that both points of zero torque are lying on the horizontal. Then the first and the last pulleys that are touched by the cable also experience two times the cable force. If this position of the VSMs is combined with an angle $\varphi_2 = 90^\circ$, the SSM is stabilised. If the VSM rotates to its minimum potential energy position, the ratio of $\frac{F_y}{F_x}$ approaches two, which corresponds to $\varphi_2 \approx 127^\circ$. For φ_2 outside the range of 90° ...127° no force ratio can be obtained with this configuration, which stabilises the SSM.

Another unsolved problem is, that the length of the cable has to be accounted for over the full range of motion of the SSM. If a pulley radius of 0mm is assumed, the cable length from one end to the other of the SSM is given by:

$$l_{C-SSM} = 4\alpha \left(\cos \frac{\varphi_2}{2} + \sin \frac{\varphi_2}{2} \right)$$
(6.5)

Figure 6.6 shows the cable length over the angle φ_2 for $\alpha = 35 \text{ mm}$.



Figure 6.6: Length of the connecting cable l_{C-SSM}

The maximum length of the cable is $198 \ mm$ whereas the minimum length is $140 \ mm$. The difference of $58 \ mm$ has to be accounted for without changing the tension of the cable as this would alter the assistance that the exoskeleton is providing.

6.1.2 Position Shifting VSM

Due to the drawbacks of the aforementioned method of two VSMs that are attached to one SSM, another concept is elaborated on here. In the previous attempt to add a range of supported motion, the second plane of motion was added in and the transition between both was explored. Since, if the arm is fully stretched with θ_4 , θ_6 and θ_7 being zero, the torque that is required to elevate the arm is always perpendicular to the plane of motion, assistance

can also be provided by letting the torque providing device travel with the arm. This not only reduces the weight of the exoskeleton but also reduces complexity in the sense of no transition zone has to be dealt with.

To enforce the torque device to provide a torque vector perpendicular to the plane of motion, the SSM must remain in a horizontal line and contract instead of rotating around the joint at the backplate. This behaviour can be achieved by either adding another *Slot Link* as introduced in Section 4.1 or by replacing the joint that is connected to the backplate with two joints that can move vertically only. To keep the SSM at a constant vertical level, a modification as shown in Figure 6.7 is suggested.



Figure 6.7: A modification of the SSM in order to keep it at a constant vertical level

The vertical gliding support ensures that the SSM remains at a constant vertical level. The two gliding supports at the extensions ensure that the orientation of the mechanism stays the same. Due to the introduced constraints for the range of motion of the SSM, the only motion that is possible is to contract or elongate. By this removal of one degree of freedom of the SSM, the rotation has to result from the other end, where the torque providing device is attached. Hence that one device stays in a position where it can provide assistance.

In order to allow the torque providing device to follow the direction of the torque vector, the attachment to the arm must be able to follow and rotate around the arm. Otherwise, the force cannot be transferred into the direction where support is needed.

The drawbacks of this idea are, that firstly a singularity that was removed in the process of inventing the SSM is introduced again. That is, that a motion in the coronal plane is not only not supported, but prevented. This results from the constraint that the SSM must remain horizontally. Secondly, the *Slot Link* presents an obstacle for the elevation motion. Consequentially the elevation of the arm has to be interrupted to manoeuvre the arm around the *Slot Link* during the elevation phase. One possible solution is to divide the SSM into three instead of two parallelograms, which would remove the need for a long *Slot Link*. However, this restricts the range of motion due to the intrusive angle of the mechanism.

6.2 Scissor based Backplate Mechanism

The backplate is connecting the human back to the SSM that is described in the previous section. It has to transfer the loads from the VSM to the back of the user. Because every person is built different, the connection between the backplate and the SSM needs to account for varying distances from the shoulder joint to the spine. One possibility of enabling such an adjustment with a relatively large range of motion is to use curved long holes that guide two joints of a scissor mechanism as shown in Figure 6.8.



Figure 6.8: Concept of the scissor based backplate mechanism with variables and points as they are used in the following; capital letters signify joints, lowercase letters give dimensions and greek letters represents the angles.

In Figure 6.8 the position of the points A and B are the input variables, that define the position of point F. Point C is a connection of the links AE and BD that is purely rotational. The distances from A to C and from B to C is l_1 , the lengths of the edges of the parallelogram CEFD is l_2 . The shape of the long holes can be described by any function, here a circular shape is used. The arcs have an equal radius r. With the angle between the x-axis and the vector pointing from the origin to point A α_1 this vector $\overrightarrow{p_A}$ is defined as:

$$\overrightarrow{p_A} = r \left\{ \begin{array}{c} -\cos(\alpha_1) \\ \sin(\alpha_1) \end{array} \right\}$$
(6.6)

In a similar manner the vector from the origin to the point $B \overrightarrow{p_B}$ is

$$\overrightarrow{p}_B = r \begin{cases} 1 - \cos \alpha_2 \\ -\sin \alpha_2 \end{cases}$$
(6.7)

The connecting vector of those two points will be called $\overrightarrow{p_{AB}}$ and is given by:

$$\overrightarrow{p_{AB}} = \overrightarrow{p_B} - \overrightarrow{p_A} \tag{6.8}$$

Together with the vector from point A to point C, $\overrightarrow{p_{AC}}$, and the vector from B to C, $\overrightarrow{p_{BC}}$, $\overrightarrow{p_{AB}}$ creates an isosceles triangle. The angle between $\overrightarrow{p_{AB}}$ and $\overrightarrow{p_{BC}}$ is

$$\gamma = 2 \arcsin\left(\frac{\left|\overrightarrow{p_{AB}}\right|}{2l_1}\right) \tag{6.9}$$

To obtain the position of *C*, a vector is needed that is perpendicular to $\overrightarrow{p_{AB}}$ to construct a right-angle triangle. With such a triangle the known length l_1 and the known angle γ can be used to calculate the vector from the origin to *C*, $\overrightarrow{p_C}$:

$$\overrightarrow{p_{C}} = \overrightarrow{p_{A}} + 0.5 \overrightarrow{p_{AB}} + \frac{1}{\left|\overrightarrow{p_{AB}}\right|} \left\{ \begin{array}{c} -p_{AB\ 2} \\ p_{AB\ 1} \end{array} \right\} l_{1} \cos\left(\frac{\gamma}{2}\right)$$
(6.10)

Here the notation $\overrightarrow{p_{AB}}_{i}$ describes the i^{th} element of the vector $\overrightarrow{p_{AB}}$. Because the links of the parallelogram *CEFD* are of equal length and the angle between the links at *C* is known, the position of the endpoint *F* is calculated as vector $\overrightarrow{p_F}$ from the origin to *F*:

$$\overrightarrow{p_F} = \overrightarrow{p_C} + \frac{1}{\left|\overrightarrow{p_{AB}}\right|} \left\{ \begin{array}{c} -p_{AB\ 2} \\ p_{AB\ 1} \end{array} \right\} l_2 \cos\left(\frac{\gamma}{2}\right)$$
(6.11)

Because the links AE and BD are rigid and only connected in C but not interrupted, the corresponding position vectors are found to be

$$\overrightarrow{p_D} = \overrightarrow{p_C} + l_2 \frac{\overrightarrow{p_{BC}}}{\left|\overrightarrow{p_{BC}}\right|} = \overrightarrow{p_C} + \frac{l_2}{l_1} \overrightarrow{p_{BC}}$$
(6.12)

$$\overrightarrow{p_E} = \overrightarrow{p_C} + l_2 \frac{\overrightarrow{p_{AC}}}{|\overrightarrow{p_{AC}}|} = \overrightarrow{p_C} + \frac{l_2}{l_1} \overrightarrow{p_{AC}}$$
(6.13)

The vectors $\overrightarrow{p_{AC}}$ and $\overrightarrow{p_{BC}}$ that have been used have the length l_1 and are obtained from $\overrightarrow{p_A}$, $\overrightarrow{p_B}$ and $\overrightarrow{p_C}$.

$$\overrightarrow{p_{AC}} = \overrightarrow{p_C} - \overrightarrow{p_A} \tag{6.14}$$

$$\overrightarrow{p_{BC}} = \overrightarrow{p_C} - \overrightarrow{p_B} \tag{6.15}$$

With the geometry of this idea fully described, a MATLAB script has been used to plot a variety of the points that *F* can take. Table 6.1 lists the variables that have been used for Figure 6.8 and Figure 6.9. In Figure 6.9, the black arcs signify the curved long holes, in which the points *A* and *B* slide. The blue areas are points, that *F* can take. In Figure 6.9a both long holes reach from 0° to 90°. The result is a range of motion that is more circular than stretched in a diagonal direction. If the second angle α_2 is increased to range from 0° to 180° as depicted in 6.9b, an extension to the shape from 6.9a in the direction of the origin is added. In a real-life application, it would not be possible to have zero distance between $\overrightarrow{p_{AB}}$ and *F*, but *F* can be made to touch the x-axis by extending α_1 into the negative region of its definition.

The influence of the different variables of the system on the shape of the possible motion is not obtained intuitively. As can be seen in Figure 6.9c, the range of motion might include areas that cannot be reached, although they are surrounded by reachable points. For the task at hand, an extended diagonal of the shape $z_m = mx_m$ where *m* is an arbitrary positive number and z_m and x_m are axes in the bodies reference coordinate system as introduced in Section 3.1, might be a desirable shape of the range of motion of *F*. However, to optimise the variables of the system towards such a shape is not a trivial task, as an objective function that accounts for holes in the range of motion is not easily found. It is also questionable if a circular shape of the long holes is the best possible solution since any arbitrary function, even discontinuous ones, are possible. Figure 6.9d shows an educated guess of what might be a good solution for the requirement of the extended diagonal of the shape z_m .

Variable	Value		
l_1	30 <i>mm</i>		
l_2	40mm		
r	20 <i>mm</i>		

Table 6.1: Input variables for the figures of this section



Figure 6.9: Range of motion for various ranges of α_1 and α_2 with the remaining variables as listed in Table 6.1; the angles are incremented with 1° (a) $\alpha_1 = 0^\circ ...90^\circ$ and $\alpha_2 = 0^\circ ...90^\circ$ (b) $\alpha_1 = 0^\circ ...90^\circ$ and $\alpha_2 = 0^\circ ...180^\circ$

(c) $\alpha_1 = 0^\circ ... 180^\circ$ and $\alpha_2 = 0^\circ ... 180^\circ$ (d) $\alpha_1 = 45^\circ ... 120^\circ$ and $\alpha_2 = 120^\circ ... 180^\circ$

This scissor based backplate mechanism provides an approach to account for different distances of the shoulder joint to the spine. It can be adapted for the curvature of the human back by transferring the system to a sphere with a smaller than the indefinite radius. It has the advantage, that with only two screws a relatively wide range of motion is covered, but needs further investigation regarding the shape of the long holes, the dimensions of the different variables and the remaining stiffness of the backplate and the system itself. Since the main objective of the project is the design of a passive exoskeleton that provides assistance with the motion of the shoulder, this approach is not followed.

Chapter 7

Preliminary Test of the Exoskeleton

In this chapter the test of the exoskeleton is described. First the range of motion and the limitations of the exoskeleton are examined. After this, the torque that the included VSM can provide is tested and is compared with the analytical and numerical analyses. The conclusions of the test of the exoskeleton are discussed, which also includes possible improvements. In the end of the chapter, a discussion is made regarding the design process with focus on the problem definition.

7.1 Testing of the Exoskeleton

In order to test the exoskeleton, the Range of Motion (RoM) needs to be examined. To do so, the SSM is attached to a backplate. The backplate used is an already existing one from the AAU laboratory. It is mounted to a wheelchair, so the testing person has to sit and position itself relative to the shoulder mechanism instead of the shoulder mechanism being attached to the body.

In order to validate the results of the analytical and numerical analyses, the torque of the exoskeleton is examined. Due to lack of time and the corona situation, the exoskeleton was examined only by the authors, so the results might not be representative.

7.2 Range of Motion

To examine the range of motion the exoskeleton is attached to the arm in the *Outer Frame* and a backplate in one end of the SSM. The backplate is attached to a wheelchair to be stable so there was no need to connect the user and the backplate by straps. Furthermore, it has to

be mentioned that a rubber band is used instead of a spring as the VSM that has been used for the evaluation of the range of motion is the one made from PLA. The initial configuration can be seen in Figure 7.1.



Figure 7.1: The attachment of the exoskeleton with the backplate and the user.

However, it can be concluded that the range of motion for the SSM that was obtained has drawbacks. Because of its small size, the available workspace of the user was reduced. Furthermore, the *Slot Link* in the sagittal plane did not follow the expected movement as it can be seen in Figure 7.2, where the *Slot Link* should follow the SSM and the VSM. This means that when the user tried to pull down the arm there was a collision between the arm and the *Slot Link*. This also happens because of the small radius of the SSM.



Figure 7.2: The drawback of the collision of the slotlink and the user

It was decided to use a bigger SSM provided by the AAU Lab to confirm, that the problem with the small size of the provided SSM reduced the workspace of the user. In order to distinguish between the provided SSM and the one used for the experiment, this new SSM will be referred to as the experimental SSM. The experimental SSM attached to the user and the backplate can be seen in Figure 7.3.



Figure 7.3: The attachment of the exoskeleton with the backplate and the user using the experimental SSM

Using the experimental SSM the assumption regarding the size of the SSM affecting the RoM was confirmed. The RoM is increased by the experimental SSM and allows the user to do most of the movements. One drawback was detected when the user tried the horizontal abduction as there was a limitation on the movement. The reason for this is because the number of links used for this SSM is increased. According to Chapter 2 and more specific the pitch angle θ_2 a limitation is expected when the SSM is closed. Three positions of the shoulder with the exoskeleton can be seen in Figure 7.4.





(c)

Figure 7.4: Three motions of the shoulder with the exoskeleton, (a) flexion, (b) horizontal adduction and (c) horizontal abduction

7.2.1 Measurement of the RoM

In order to measure the RoM of the exoskeleton, the three main planes are examined. These planes are described in Chapter 1 and are the sagittal, the coronal and the transverse planes. The widely accepted method of measuring the range of motion of joints is the usage of goniometers, but it has been shown that the smartphone application "Clinometer" from Plaincode Software Solutions also gives acceptable measurements [Werner et al., 2013]. The measurements are done using a protractor for the transverse plane and the Clinometer for

Shoulder Motion	Flexion	Hyperextension	Abduction	Horizontal Adduction	Horizontal Abduction
Degrees [°] ± 2	170	34	167	119	34

the sagittal and the coronal plane. Pictures of the measurements can be found in Appendix I. Table 7.1 shows the measured results with the uncertainty of $\pm 5^{\circ}$.

Table 7.1: Measurements for the shoulder motions

According to [Gill et al., 2020] the exoskeleton does not restrict the workspace of the user significantly. Especially for the overhead workers the exoskeleton does not reduce the workspace.

7.3 Measurement of the Provided Torque

To validate the design of the VSM experimentally, torque tests have been conducted. The VSM has been separated from the SSM and attached to a torque measurement cell via an adapter. The torque has been measured with a strain gauge based torque cell from Forsenteck Co. The cell measures the torque based on the strain reaction of a core element. The specific model that has been used is the FTE-20NM and the Indicator FPTD as a reading device. The rated output (R.O.) of the load cell is 1.3930mV/V and the output excitation voltage of the reading device is 5V. With the uncertainties for hysteresis, nonlinearity and nonrepeatability this results in a measuring uncertainty of 0.04Nm [Forsentek Co., 2021].

In lack of an angle encoder, a protractor has been used. It has been clamped to the support structure by the adapter for the VSM. The adapter has made use of the flattened sides of the shaft of the torque cell. The torque cell has been fixed by clamping the screws of its flange to the support structure and did not have noticeable backlash. The VSM has been attached by fitting the *Connection Slot*, as described in Chapter 3.2, to the other side of the adapter. The initial position with unloaded spring has been chosen to be at a comparable angle of $\gamma = 180^{\circ}$, this corresponds to the initial position as would be in combination with the SSM. Figure 7.5 shows the setup.



Figure 7.5: (a) clamping of the protractor, (b) VSM attached in unloaded position with the torque cell in the background in blue

Measurements have been conducted with two setups. The first setup used 3D printed pulleys running on screws. Those results will not be presented here but can be seen in Appendix J, because the high friction between the pulleys and the screws caused a large hysteresis with a maximum torque of around 300% of the predicted value. Instead, the results that are presented have been obtained after replacing the 3D printed pulleys with metal pulleys that run on roller bearings. Those are of the same outer radius, but the radius that the cable runs in is 0.5*mm* smaller in diameter.

Various measurements with different spring stiffnesses have been conducted. The first spring that the VSM has been tested with had a stiffness of 0.84N/mm. The results were used to improve the measurement procedure and can be seen in Appendix K. The measurements, that are presented in Figure 7.6, have been conducted with a spring with a stiffness of 3.52N/mm and a preload of 27.18N. The mechanism has been rotated to a certain angle by one person and two other persons noted the torque value independently of each other. Before saving the value, it has been waited for the reading to settle. Three cycles have been conducted where the angle increment of which readings were taken was 5°. One cycle consists of 180° where the torque was measured during loading and unloading. At 180° two readings have been taken, the first one when coming from 175° and the second one after ensuring a stable position of 180° . After each cycle, the cable has been tightened to eliminate backlash even at small deflections.



Figure 7.6: Average and standard derivation of the measured torque values over γ for the loading and the unloading of the spring as well as the result from the analytical model and the MSC ADAMS simulation

Figure 7.6 shows the average values of the torque readings where all readings have the same weight, so they are interpreted as independent cycles. Each phase is represented by the average value and the standard deviation of the torque readings. At points of the curve where no standard deviation is given, the curve is interpolated and no measurements have been conducted. The highest standard deviation is 0.38Nm and is observed during the loading of the spring at an angle of $\gamma = 125^{\circ}$. A figure with the measured data points can be found in Appendix L.

It can be observed, that from the start of the cycle the measured torque aligns well with the predictions, although it meanders around both, the analytical and the simulated prediction. With γ approaching 90°, the experimental results seem to follow more the analytical model, which gives higher results. At $\gamma = 90^{\circ}$ the experimental results detach from the predicted results and while over the following 35° to 40° the predicted torque decreases, the measured torque remains almost constant with a peak at $\gamma = 55^{\circ}$. From the constant section on, the experimentally obtained torque decreases and shows a similar slope to the predicted curves. At $\gamma = 0^{\circ}$ after 180° deflection, a torque discrepancy of 1.9Nm between the predictions and the readings remains.

At the beginning of the unloading phase at $\gamma = 0^{\circ}$ the measured torque aligns with the predictions. For the first 15° to 20° of the motion, the torque is negative, but shows a similar slope to the predictions and the loading phase until $\gamma = 35^{\circ}$. In this area, the standard deviation shows larger values than in the rest of the unloading curve with the maximum of

0.25Nm at $\gamma = 35^{\circ}$. From there on with increasing γ the slope is more flat-angled than the predictions, although the position of the maximum of the unloading phase aligns with the predicted curves. From the maximum on, the torque diminishes with different slope than before until zero torque is reached at $\gamma = 165^{\circ}$ or 15° deflection from the initial angle.

Conclusions

The begin of the cycle shows very good alignment between the experimental results and the predictions. This is expected as the validity of the analytical model has been shown prior [Li et al., 2020], [Bai & Li, 2019]. The detaching after 90° can be a result of increasing friction. When the spring gets elongated, the cable force increases, thus adding force on the bearings. Because the VSM has been moved by hand in increments, this friction has to be overcome for every increment. It is seen that the difference increases from increment to increment in the range of $\gamma = 90^{\circ}$... $\sim 35^{\circ}$, after which the deviation stays constant. In the unloading phase it can be concluded from looking at the average of the loading and unloading curves in the range of $\gamma = 15^{\circ}...35^{\circ}$, that the same difference acting in opposite direction is causing the hysteresis. This makes the deviation likely to be produced by friction in the bearings. The used bearings are gliding bearings and the gliding surfaces are not perfect. Thus the two surfaces produce more and more friction with increasing cable force, until the maximum amount of friction that can be produced is reached, which leads to the hysteresis that is observed. This theory could be confirmed by further experiments where the temperature in the bearing is measured or where the bearings are exchanged for a roller bearings and the results are compared to the results with the present gliding bearings.

At the start of the unloading phase, the measured torque is negative and free motion is not possible. After overcoming the area where the friction is constricting the motion, the standard deviation is in the largest region between $\gamma = 15^{\circ}...45^{\circ}$. Directly after this region, the slope of the curve changes abruptly, causing a deviation of the average of the loading and unloading from the predictions. Compared to the predicted curves and the loading phase, the slope is now more flat. Because of the high standard deviation of the readings, the slope decoupling, the lower maximum and the fact, that zero torque is reached at a larger than zero deflection, it is assumed, that in the area of $\gamma = 15^{\circ}...45^{\circ}$ the cable is irreversibly elongated, leading to a smaller force and reaching zero torque prior to zero deflection. In Li et al. [2020] a similar effect is observed and explained by nonlinearity of the cable and manufacturing imperfections. It is assumed here, that with a constant cable length, the hysteresis shown should decrease and the average of the loading and the unloading process should follow the predicted torque curve more closely.

7.4 Future Work

In this section improvements and remaining work for the further development of the current design are described. Those are derived from the preliminary testing of the exoskeleton and general reflection on the prototype.

7.4.1 General Improvements

In the following, some ideas to improve the design, as well as detected shortcomings, are listed. Those could not be implemented or improved, respectively, due to the limited amount of time.

- One drawback that has been detected during the experiment for the range of motion concerns the straps that connect the VSM with the user's arm. Since there was only one location assigned on the VSM for connection of the strap, the VSM was not stable. As a result, the whole exoskeleton failed to align properly when the arm was moved. However, when two pairs of straps were used the VSM was more stable and aligned better with the arm.
- During the measurements of the torque provided by the VSM, it was found that there was high friction, which caused a large hysteresis. By switching to pulleys with roller bearings the problem was reduced slightly. To reduce friction further, the design of the housing has to be changed to use roller bearings instead of polymer sleeve bearings. In the literature, the use of double-row roller bearings with large safety factors to reduce internal forces is recommended [Dežman & Gams, 2018]. This approach has not been followed to enable a compact design.
- There are a couple of minor issues with the design of the housing of the prototype, too. The first one is, that the *Outer Housing* is pressed against the pulleys by the screws. This compromises the rotation of the pulleys due to an increase in the friction between the contact surfaces. This problem can be overcome by adding spacers between the *Outer Housing* and the *Outer Frame*. Also for long term usage of the mechanism, the housing should be enclosed to protect it from dust entering, as this again will increase

the friction.

- During the experiment, it proved to be vital for the function, that the joints of the SSM are aligned with the glenohumeral joint of the test person. This calls for a flexible and well adjustable backplate, that is capable of easily customizing the position of the common joint of the backplate and the SSM. A further improvement would be to give the backplate the range of motion to follow the shoulder in the two translational degrees of freedom as well.
- To further improve the user experience, the *Connetion Slot* should contain rollers. This will make it easier for the user to force the axis of rotation of the VSM to follow the arm.
- So far no thought has been given to the weight of the assembly. During the experiment, it has been observed, that the SSM has the tendency to rotate around the common joint with the backplate. Stiffening this rotation not only makes the mechanism less fee-lable but also increases the assisted range of motion as described in Section 6.1.2.
- From the torque measurement it has been concluded, that the cable length is increased irreversibly during the usage of the mechanism. An ultimate reason could not be found as the cable that was used is a plaited PES rope with a tensile strength of 1000N and the maximum applied force is around one-third of the strength. For a quick solution, a steel cable can be used, although this cannot be expected to reduce the hysteresis completely as there are other influences like manufacturing imperfections that can also cause hysteresis [Li et al., 2020].
- The exoskeleton is only examined by the authors of this project. In order for the results to be more representative regarding the range of motion and the comfort of the exoskeleton, the exoskeleton has to be examined by external users.

7.4.2 Updated Scissors Design

Since the SSM is found to be too small, it is decided to update the design and make it larger to fit the user and cover more range of motion. Using the provided SSM design and taking inspiration from the experimental SSM the final design of the SSM is developed and can be seen in Figure 7.7. The final design combines the range of motion of the experimental SSM and also the compactness of the provided SSM. The stress analysis is conducted on the updated design in order to validate that it can withstand the torque provided by the VSM. The dimensions of the SSM are provided in Appendix G.



Figure 7.7: Updated design of the SSM

It is decided to use Aluminium-7075 T6 for this design. The loading and boundary condition of the study is similar to the one used in 5.2.1. A cylindrical fixed support is used on one end of the Scissors and the moment of 6532*Nmm* is applied on the *Connection Slot*. The stress distribution on the updated Scissors mechanism can be seen in Figure 7.8. It is seen that there is a stress concentration where the boundary condition is applied. Ignoring this and probing the sections on the links which are not under the influence of stress concentration, it is found that, the maximum stress is 227*MPa*. This value is still under the safety limit. The result for the stress value in the links and convergence study can be seen in Appendix F.



Figure 7.8: Stress distribution of the updated SSM

7.5 Discussion

The exoskeleton that was developed in this thesis has the task of assisting workers and the elderly in lifting the upper limbs to reduce the danger of injuries or ease movement, respectively. The problem formulation, as stated in Chapter 1, is:

A passive, compact and lightweight exoskeleton is to be developed, that supports the human arm movement by reducing the load acting on the shoulder muscles. Motion in the sagittal plane is of the highest priority, but differing planes also have to be considered. If possible with the covid situation at hand, a prototype is to be used to validate the design. If this proves impossible, numerical simulations are to be used for verification.

The mechanism that has been developed is capable of assisting the human arm not only in the sagittal but also in other planes. This happens if the SSM contracts instead of rotating around the joint of the backplate and is described in Section 6.1.2 as "Position shifting VSM" as a method to increase the assisted range of motion in Chapter 6. This means, that the objective of a broad assisted range of motion has been met. The weight of the SSM in the larger version as used in the experiment is 385g, the VSM weighs 522g and the total assembly comes to 908g. This is without a backplate and straps for it, so a full exoskeleton with two times the suggested assembly, a backplate and straps to wear it is assumed to have a weight of 2.7kg. So it is at the lower end of the analysed exoskeletons. To further reduce the weight, the component with arguably the largest potential is the backplate. One solution is to build it from a composite material as in Peng et al. [2021] or Peng et al. [2020], or to replace it with a new design as seen in various existing exoskeletons in Chapter 1.

Due to the covid situation and the time limit, no experiments in an actual usage scenario have

been conducted. This is to be done when the design improvements have been implemented. The conducted tests have proven, that the range of motion of the original scissors is too small. However, with the design as described in Section 7.4.2 the range of motion is deemed to be sufficient, but more precise measurements have to be conducted. It also has been proven by FEM analysis, that the loads from the torque providing mechanism can be sustained by the updated scissors.

The torque tests as described in Section 7.3 revealed, that the overall design and optimisation process has yielded a functional actuator design. It has been concluded from the experiment, that the torque that is predicted by the analytical model of the mechanism and the MSC ADAMS simulation can be met if certain improvements are implemented. The most important one is the usage of a stiffer cable with less nonlinear effects. Secondly, bearings with lower friction have to be used in order to reduce the hysteresis.

Chapter 8

Conclusion

In this thesis, a design of a compliant shoulder exoskeleton is presented. The recently developed Shoulder Scissors Mechanism is used to reproduce the motion of the shoulder joint and a Variable Stiffness Mechanism is used for gravitational compensation for the human arm.

The upper body kinematics was studied to develop the torque requirement for the VSM. Using optimisation, the dimensions of the VSM were calculated and used to develop the initial design of the mechanism. The loads acting on the VSM were calculated in MSC ADAMS and used in Ansys to conduct a structural analysis of the mechanism.

Topology optimisation was used to reduce the mass of the VSM. A final design was developed which reduced the weight of the original *Outer Frame* by 70% from 466g to 143g. This design was successfully confirmed to withstand the loads experienced by the mechanism. Additional design ideas were also presented to include assistance for the second degree of freedom in the abduction/adduction plane.

The torque generated by the VSM was experimentally measured using a torque sensor and the result was compared with the analytical and numerical solution. It was observed that there was a discrepancy in the readings due to the high friction experienced by the pulleys in the VSM. The Range of Motion of the device was also measured experimentally by connecting the exoskeleton to the backplate. The drawbacks of the mechanism were studied and possible improvements were suggested.

Overall, the novel mechanism provides a large range of motion and is estimated to be lighter than the most of the exoskeletons that have been studied. Its compactness, modularity and adaptable torque behaviour make it interesting for further development.

Bibliography

ANSYS (2021). Mechanical user's guide. ANSYS. Downloaded: 18-5-2021.

- Bai, S. & Li, Z. (2019). A novel revolute joint of variable stiffness with reconfigurability. *Mechanism and Machine Theory*, 133, 720–736.
- Bai, S., Virk, G. S., & Sugar, T. (2018). *Wearable Exoskeleton Systems : Design, Control and Applications*. Control, Robotics and Sensors. Institution of Engineering and Technology.
- Castro, M. N., Rasmussen, J., Andersen, M. S., & Bai, S. (2019). A compact 3-dof shoulder mechanism constructed with scissors linkages for exoskeleton applications. *Mechanism* and Machine Theory, 132, 264–278.
- Chen, Y., Li, G., & Zhu, Y. (2014). Design of a 6-dof upper limb rehabilitation exoskeleton with parallel actuated joints. *Bio-Medical Materials and Engineering*, 24(6), 2527–2535.
- Choi, H. (2018). Suitx. https://www.suitx.com/backX. Accessed: 30-05-2021.
- Choi, H. (2021). Assistance of a person with muscular weakness using a a joint-torqueassisting exoskeletal robot. *Applied Sciences*, 11(7).
- Christensen, S. & Bai, S. (2017). A novel shoulder mechanism with a double parallelogram linkage for upper-body exoskeletons. *Biosystems and Biorobotics*, 16, 51–56.
- de Leva, P. (1996). Adjustment to zatsiorsky-seluyanov's segment inertia parameters. *Journal of Biomechanics*, 29(9), 1223–1230.
- Dežman, M. & Gams, A. (2018). Rotatable cam-based variable-ratio lever compliant actuator for wearable devices. *Mechanism and Machine Theory*, 130, 508–522.
- Exobionics (2018). Exobionics. https://eksobionics.com/past-products/. Accessed: 05-03-2021.

- Exobionics (2020). Evo. https://eksobionics.com/ekso-evo/. Accessed: 05-03-2021.
- Farah, S., Anderson, D. G., & Langer, R. (2016). Physical and mechanical properties of pla, and their functions in widespread applications — a comprehensive review. *Advanced Drug Delivery Reviews*, 107, 367–392. PLA biodegradable polymers.
- Feeney, D. (2019). 3d printing with abs vs pla. https://www.sd3d.com/3d-printing-abs-vs-pla/. Accessed: 16-05-2021.
- Forsentek Co., L. (2021). Static torque measurement device 20nm 10nm 5nm 2nm 1nm measure torque. http://www.forsensor.com/sale-8832707-static-to rque-measurement-device-20nm-10nm-5nm-2nm-1nm-measure-torq ue.html. Accessed: 31-05-2021.
- Furnémont, R., Mathijssen, G., van der Hoeven, T., Brackx, B., Lefeber, D., & Vanderborght, B. (2015). Torsion maccepa: A novel compact compliant actuator designed around the drive axis. In 2015 IEEE International Conference on Robotics and Automation (ICRA) (pp. 232–237).
- Gill, T. K., Shanahan, E. M., Tucker, G. R., Buchbinder, R., & Hill, C. L. (2020). Shoulder range of movement in the general population: age and gender stratified normative data using a community-based cohort. *BMC Musculoskelet Disord*, 21(1).
- Gull, M. A., Bai, S., & Bak, T. (2020). A review on design of upper limb exoskeletons. *Robotics*, 9(1).
- Hyun, D. J., Bae, K., Kim, K., Nam, S., & hyun Lee, D. (2019). A light-weight passive upper arm assistive exoskeleton based on multi-linkage spring-energy dissipation mechanism for overhead tasks. *Robotics and Autonomous Systems*, 122, 103309.
- Jafari, A., Tsagarakis, N. G., Vanderborght, B., & Caldwell, D. G. (2010). A novel actuator with adjustable stiffness (awas). In 2010 IEEE/RSJ International Conference on Intelligent Robots and Systems (pp. 4201–4206).
- Kohnke, P. (1999). Theory reference. Downloaded: 15-05-2021.
- Krishnan, R., N. B., Gutierrez-Farewik, E. M., & Smith, C. (2019). A survey of human shoulder functional kinematic representations. 57, 339–367.

- Li, Z., Bai, S., Madsen, O., Chen, W., & Zhang, J. (2020). Design, modeling and testing of a compact variable stiffness mechanism for exoskeletons. *Mechanism and Machine Theory*, 151, 103905.
- Lo, H. S. & Xie, S. S. (2014). Optimization of a redundant 4r robot for a shoulder exoskeleton. *Robotica*, 32(8), 798–803.
- Ltd, I. B. L. (2021). Plavs abs. https://www.makeitfrom.com/. Accessed: 16-05-2021.
- Lund, E., Olhoff, N., & Du, J. (2019). Topology optimization and its application in static design of continuum structures. Downloaded: 25-05-2021.
- Luque, E. P. (2019). Evaluation of the use of exoskeletons in the range of motion of workers. Downloaded: 13-4-2021.
- Marinov, B. (2016). Introduction to the commercial exoskeletons catalog. https://exoskeletonreport.com/2016/12/introduction-to-the-commercial-exoskeletonscatalog/. Accessed: 20-03-2021.
- MathWorks, I. (2021a). Constrained nonlinear optimization algorithms. https://de.m athworks.com/help/optim/ug/constrained-nonlinear-optimizat ion-algorithms.html#brnpd5f. Accessed: 22-03-2021.
- MathWorks, I. (2021b). fmincon. https://de.mathworks.com/help/optim/ug /fmincon.html. Accessed: 22-03-2021.
- MATWEB (2021). Material property data. http://www.matweb.com/index.aspx. Accessed: 16-05-2021.
- Maurice, P., Camernik, J., Gorjan, D., Schirrmeister, B., Bornmann, J., Tagliapietra, L., Latella, C., Pucci, D., Fritzsche, L., & Ivaldi, S. (2020). Objective and subjective effects of a passive exoskeleton on overhead work. *IEEE Transactions on Neural Systems and Rehabilitation Engineering*, 28(1), 152–164.
- Mscsoftware (2010). Md adams release guide. https://www.mscsoftware.com/. Accessed: 21-03-2021.
- Naidu, M. D., Stopforth, D. R., Bright, P. G., & Davrajh, M. S. (2011). A 7 dof exoskeleton arm:shoulder,elbow,wrist and hand mechanism for assistance to upper limb disabled individuals. *Computer Science*, (pp. 1–6).
Nobel-Jørgensen, M. (2016). Interactive Topology Optimization. PhD thesis.

Norton, R. L. (2010). Machine Design - An Integrated Approach, volume 4. Prentice Hall.

- Pacifico, I. & Scano, A. (2020). An experimental evaluation of the proto-mate: A novel ergonomic upper-limb exoskeleton to reduce workers' physical strain. *IEEE Robotics Automation Magazine*, 27(1), 54–65.
- Paexo (2018). Paexo. https://paexo.com/paexo-shoulder/. Accessed: 05-03-2021.
- Palais, B., Palais, R., & Rodi, S. (2009). A disorienting look at euler's theorem on the axis of a rotation. *The American Mathematical Monthly*, 116(10), 892–909.
- Papadopoulos, E. & Patsianis, G. (2007). Design of an exoskeleton mechanism for the shoulder joint. *IFToMM*.
- Peng, X., Dai, Z., Liu, J., & Wang, Q. (2020). The design and calculation of the exoskeleton backplate based on the composite sandwich structure. *IOP Conference Series: Earth and Environmental Science*, 571, 012118.
- Peng, X., Dai, Z., Liu, J., & Wang, Q. (2021). Design and simulation of sandwich structure of exoskeleton backplatebased on biological inspiration. *Journal of Physics: Conference Series*, 1885.
- Sigmund, O. (2001). A 99 line topology optimization code written in matlab. *Structural and Multidisciplinary Optimization*, 21, 120–127.
- Soames, R., Palastanga, N., & Field, D. (1994). *Anatomy and human Movement structure and function*. Butterworth-Heinemann, second edition edition.
- Sotola, M., Marsalek, P., Rybansky, D., Fusek, M., & Gabriel, D. (2021). Sensitivity analysis of key formulations of topology optimization on an example of cantilever bending beam. *Symmetry*, 13(4).
- S.p.A., C. (2021). Mate-xt fit for workers. https://mate.comau.com/. Accessed: 21-02-2021.
- Tim Bosch, J. v. E., Knitel, K., & de Looze, M. (2016). The effects of a passive exoskeleton on muscle activity, discomfort and endurance time in forward bending work. *Applied ergonomics*, 54, 212–217.

- Vahrenkamp, N., Asfour, T., Metta, G., Sandini, G., & Dillmann, R. (2012). Manipulability analysis. In 2012 12th IEEE-RAS International Conference on Humanoid Robots (Humanoids 2012) (pp. 568–573).
- Van Ham, R., Vanderborght, B., Van Damme, M., Verrelst, B., & Lefeber, D. (2007). Maccepa, the mechanically adjustable compliance and controllable equilibrium position actuator: Design and implementation in a biped robot. *Robotics and Autonomous Systems*, 55(10), 761–768.
- Vanderborght, B., Tsagarakis, N. G., Semini, C., Van Ham, R., & Caldwell, D. G. (2009). Maccepa 2.0: Adjustable compliant actuator with stiffening characteristic for energy efficient hopping. In 2009 IEEE International Conference on Robotics and Automation (pp. 544–549).
- Werner, B. C., Kuenze, C. M., Griffin, J. W., Lyons, M. L., Hart, J. M., & Brockmeier, S. F. (2013). Shoulder range of motion: Validation of an innovative measurement method using a smartphone. *Orthopaedic Journal of Sports Medicine*, 1(4_suppl), 2325967113S00106.
- Wolf, S., Eiberger, O., & Hirzinger, G. (2011). The dlr fsj: Energy based design of a variable stiffness joint. In 2011 IEEE International Conference on Robotics and Automation (pp. 5082–5089).
- Wolf, S. & Hirzinger, G. (2008). A new variable stiffness design: Matching requirements of the next robot generation. In 2008 IEEE International Conference on Robotics and Automation (pp. 1741–1746).
- Yatsun, A. & Jatsun, S. (2018). Investigation of human cargo handling in industrial exoskeleton. In 2018 Global Smart Industry Conference (GloSIC) (pp. 1–5).
- Zhang, Z.-Q., Wong, W.-C., & Wu, J.-K. (2011). Ubiquitous human upper-limb motion estimationusing wearable sensors. *IEEE Transactions on Information Technology in Biomedicine*, 15(4), 513–521.

Appendix A

Derivation of geometric variable c

The geometric variable c describes the influence of geometry of mechanical parts on the cable elongation [Li et al., 2020]. When the cable length L is in position that it can be seen in Figure A.1 then a change in the length is affected by the length l_2 and more specifically changes two times the change in length l_2 .



Figure A.1: Configuration of VSM where $\theta = 0$

However, when there is a arbitrary configuration like in Figure A.2 where $\theta \neq 0$ then the cable length does not change proportionally to the length l_2 . In order to see how much the cable length changes two contact angles are induced, which define the area that the cable is in contact with the pulleys. When $\theta = 0$ then the contact angle is equal to $\pi/2$. But when there is a different configuration then the contact angle changes and it is depended on the position of pulley-2(P_2) with reference the pulley-3(P_3) and pulley-1(P_1). This can be found using the triangle AO_2O_3 and calculating the line that connects the centers of P_2 and P_3 as:

$$\overline{O_3O_2} = \sqrt{\overline{AO_2}^2 + \overline{AO_3}^2} \tag{A.1}$$

where the two edges $\overline{AO_2}$ and $|\overline{AO_3}|$ of the triangle can be found as:

$$\overline{AO_2} = 2R - \sin\theta l_1 \tag{A.2}$$

$$AO_3 = l_3 - \cos\theta l_1 \tag{A.3}$$

where R is the radius of the pulleys.

Replacing the equations (A.2) and (A.3) in equation A.1 then the line $\overline{O_3O_2}$ is found as:

$$\overline{O_3 O_2} = \sqrt{(2R - \sin\theta l_1)^2 + (l_3 - \cos\theta l_1)^2}$$
(A.4)

Using the triangle CO_2O_3 and since $\overline{CB} = R$ the angle θ_2 can be found as:

$$\theta_2 = \arcsin \frac{\overline{CO_2}}{\overline{O_2O_3}} = \arcsin \frac{2R}{\overline{O_2O_3}}$$
(A.5)

To find the angle θ_1 the triangle AO_2O_3 is used:

$$\theta_1 = \arcsin \frac{\overline{AO_2}}{\overline{O_2O_3}} \tag{A.6}$$

The difference of θ_2 and θ_1 defines the angle θ_3 . If θ_3 is added with $\pi/2$ then the angle θ_3 can be found as:

$$\theta_4 = \theta_3 + \frac{\pi}{2} \tag{A.7}$$

In contrast, the angle θ_3 has to be subtracted with $\pi/2$ to find the angle that the cable length is in contact with P_1 .

$$\overline{DB} = \sqrt{\overline{O_2 O_3}^2 - 4R^2} \tag{A.8}$$

The same procedure is followed to find the \overline{EF} , which corresponds to the pulleys P_1 and P_3 . As it can be seen from Figure A.2 the perpendicular distance from the center of P_1 to the center of P_2 is the same as the perpendicular distance for the pulleys P_2 and P_3 :

$$\overline{O_1 G} = \overline{AO_3} \tag{A.9}$$

Regarding the horizontal distance $(\overline{O_2G})$, it can be found as:

$$\overline{O_2 G} = 2R + \sin \theta l_1 \tag{A.10}$$

Using the triangle $O_1 G O_2$ then the edge $\overline{O_2 O_1}$ can be found as:

$$\overline{O_2 O_1} = \sqrt{\overline{GO_1}^2 + \overline{GO_2}^2} \tag{A.11}$$

The distance \overline{EF} can now be found as:

$$\overline{EF} = \sqrt{\overline{O_2 O_1}^2 - 4R^2} \tag{A.12}$$

on is found as:



Figure A.2: An arbitrary configuration of VSM where $\theta \neq 0$.

The cable length L_a in an arbitrary configuration and the cable length in L when *theta* = 0 can be found as:

$$L_a = R\theta_4 + \overline{DB} + \pi R + \overline{EF} + R(\pi/2 - \theta_3)$$
(A.13)

$$L = R\frac{\pi}{2} + \overline{DB} + R\pi + \overline{EF} + R\frac{\pi}{2}$$
(A.14)

The geometric variable is defined as the change in the cable because of the change in length of l_2 :

$$c = \frac{L_a - L}{l_{2a} - l_2}$$
(A.15)

where l_{2a} is the length of link-2 in an arbitrary position.

In Figure A.3 it can be seen how the geometric variable *c* varies as a function of θ . It can be concluded that the value of the geometric variable is close to 2 for the range $0^{\circ} - 180^{\circ}$ of θ .



Figure A.3: The geometric variable c as a function of the angle θ .

Appendix B

Manipulator Jacobian matrix

The angular velocity ω_e of the end-effector of the scissors is given from the equation below:

$$\omega_e = J\dot{\theta} \tag{B.1}$$

where J is the Jacobian matrix and $\dot{\theta}$ are the mechanism's joint velocities.

In geometry, Euler's rotation theorem states that, in three-dimensional space, any displacement of a rigid body such that a point on the rigid body remains fixed, is equivalent to a single rotation about some axis that runs through the fixed point [Palais et al., 2009]. Assuming the speed of rotation $\dot{\theta}_s$ about the instantaneous axis of rotation \hat{e} then the angular velocities ω_e at any time are given as:

$$\omega_e = \dot{\theta_s} \hat{e} \tag{B.2}$$

Based on the above equation, the angular velocities ω_e can be derived from the skewsymmetric matrix S of the angular-velocities for the particular rotation matrix Re of the mechanism [Castro et al., 2019]. The skew-symmetric matrix S is dervided as:

$$S = \dot{R}_e R_e^T = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix}$$
(B.3)

From the above equation the three following equations can be derived:

$$\omega_x = \dot{r}_{31}\dot{r}_{21} + \dot{r}_{32}\dot{r}_{22} + \dot{r}_{33}\dot{r}_{23} \tag{B.4}$$

$$\omega_y = \dot{r}_{11}\dot{r}_{31} + \dot{r}_{12}\dot{r}_{32} + \dot{r}_{13}\dot{r}_{33} \tag{B.5}$$

$$\omega_z = \dot{r}_{21}\dot{r}_{11} + \dot{r}_{22}\dot{r}_{12} + \dot{r}_{23}\dot{r}_{13} \tag{B.6}$$

Replacing these thee equations in equation (B.2) for $\omega_e = [\omega_x, \omega_y, \omega_z]^T$ then the manipulator Jacobian matrix can be found as:

$$J = \begin{bmatrix} 0 & c\theta_1 & s\theta_1 s\theta_2 \\ 0 & s\theta_1 & -c\theta_1 s\theta_2 \\ 1 & 0 & c\theta_2 \end{bmatrix}$$
(B.7)

Appendix C

Scissors bearings

The bearings that are used in the scissors are IGUS GFM - 0607 - 024. They are put inside the holes of the links in order to reduce the friction of the bolts with the links. IGUS GTM - 0611 - 010 are out above the hole links in order to reduce the friction between the head of the bolts and the links. The material of the two bearings is iglidur and are made by high-performance polymers. Their special composition makes them extremely wear-resistant, robust and self-lubricating.



Figure C.1: The two types of bearings that used in the scissors. a) is the IGUS GFM - 0607 - 024 and b) the IGUS GTM - 0611 - 010 type.

Appendix D

Homogeneous Matrices of the Human Arm

In the following the homogeneous matrices of the model of the human arm as described in Section 3.1 are given. The indexing is done in such a way, that the lower index refers to the original coordinate system and the upper index refers to the coordinate system that a point is transferred to. The sine and cosine functions are abbreviated with s_i and c_i respectively, where *i* is the index of the coordinate system as shown in Figure 3.1b. For the transformation

from the i^{th} to the $(i-1)^{th}$ system are obtained:

$$H_{m}^{1} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ s_{1} & -c_{1} & 0 & 0 \\ c_{1} & s_{1} & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.1)
$$H_{1}^{2} = \begin{bmatrix} -s_{2} & c_{2} & 0 & 0 \\ 0 & 0 & 1 & 0 \\ c_{2} & s_{2} & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.2)
$$H_{2}^{3} = \begin{bmatrix} c_{3} & -s_{3} & 0 & 0 \\ 0 & 0 & -1 & 0 \\ s_{3} & c_{3} & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.3)
$$H_{3}^{4} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ c_{4} & -s_{4} & 0 & 281.7 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.4)
$$H_{4}^{5} = \begin{bmatrix} c_{5} & -s_{5} & 0 & 0 \\ 0 & 0 & 1 & 0 \\ -s_{5} & -c_{5} & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.5)
$$H_{5}^{6} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ s_{6} & c_{6} & 0 & 0 \\ -c_{6} & s_{6} & 0 & 268.9 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.6)
$$H_{7}^{6} = \begin{bmatrix} c_{7} & 0 & s_{7} & 0 \\ -s_{7} & 0 & c7 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(D.7)

Appendix E

Pendulum example in Adams

Assuming a simple pendulum as it can be seen in figure E.1. The pendulum is attached to a fixed frame through a revolute joint that allows the pendulum to rotate. The vector \overrightarrow{A} represent the location of the revolute joint in the fixed reference frame, the vector \overrightarrow{R} represents the location of the center of the mass of the pendulum in the fixed reference frame and the vector \overrightarrow{r} the distance from the center of the mass of the pendulum to the revolute joint.



Figure E.1: Simple pendulum that is attached to a fixed frame through a revolut joint..

ADAMS solver is using the Lagrange equation as it was described in equation (4.4). The

equations that come of the Langrange equation are:

$$\begin{pmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & I \end{pmatrix} \begin{pmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{\theta} \end{pmatrix} + \begin{pmatrix} 0 \\ mg \\ 0 \end{pmatrix} + \Phi_q^T \lambda = Q_{ex}$$
(E.1)

The three equations in equation (E.1) has as results an underdetermined system. The other two equations are coming from the constraint conditions. The restriction on the pendulum can be described by the following equation:

$$\overrightarrow{R} - (\overrightarrow{A} - \overrightarrow{r}) = 0 \tag{E.2}$$

which can be written in matrix notation as:

$$\Phi(x, y, \theta) = \begin{pmatrix} x - l\cos\theta - A_1 \\ y - l\sin\theta - A_2 \end{pmatrix} = 0$$
(E.3)

With the two additional equations from equation (E.3) the system is complete. However, the system contains second order differential equations and ADAMS introduces a new dependent variable and reduces the order by one. After this, these equations are transformed into a system of nonlinear equations by approximating each derivative with a backward differentiation formula.

Appendix F

Convergence Plots

F.1 Convergence plots for Open-Scissors

The convergence is done for the half torque provided (6532Nmm) and also for the weak spring that used where the maximum torque was 1912 Nmm.





SSM Open

Figure F.1: Stress convergence study of Open-Scissors configuration for different links.

F.1.2 Stress for Torque-1912 Nmm



SSM Open-weak spring

Figure F.2: Stress convergence study of Open-Scissors configuration for different links with weak spring.

F.2 Convergence Plots for Middle-Scissors

Again for the Middle-Scissors the convergence is done for the provided torques 6532 Nmm and 1912 Nmm, respectively.

F.2.1 Stress for Torque-6532 Nmm



SSM Middle

Figure F.3: Stress convergence study of Middle-Scissors configuration

F.2.2 Stress for Torque-1912 Nmm



SSM middle-weak spring

Figure F.4: Stress convergence study of Middle-Scissors configuration for different links with weak spring

F.3 Convergence Plots for Close-Scissors

Also for the Close-Scissors the converge study is done for both cases of the provided torques.

F.3.1 Stress for Torque-6532 Nmm



SSM Close

Figure F.5: Stress convergence study of Close-Scissors configuration.

F.3.2 Stress for Torque-1912 Nmm



SSM close-weak spring

Figure F.6: Stress convergence study of Close-Scissors configuration for different links with weak spring

F.4 Convergence Plots for Updated-Scissors

As done before for the provided scissors mechanism because of the stress concentration points the stress analysis is done away from these points. In Figure F.7 it can be seen the links where the stress analysis is done.



Figure F.7: Highlighted links where the stress analysis is done.

The convergence study for the Updated Scissors mechanism can be seen in Figure F.8.



SSM Updated

Figure F.8: Stress convergence study of Updated-Scissors configuration for different links.

Appendix G

Engineering Drawings

G.1 VSM Assembly

G.1.1 VSM side



Figure G.1

G.1.2 VSM Outer Disc



Figure G.2

G.1.3 VSM Inner Disc



Figure G.3

G.1.4 VSM Spring Base



Figure G.4

G.1.5 VSM Cuff attachment



Figure G.5

G.2 SSM Assembly



Figure G.6

Appendix H

Further Considerations for the Two-VSM-One-Cable Concept

Numerical Solving of the governing differential equations for *l*₁

To obtain a behaviour of the length l_1 that produces the required torque behaviour for different elevation angles, a MATLAB script is used. After calculating and storing the values of the required torque as exemplary shown for 90° elevation in Figure 6.1a and the VSM torque in dependency of γ and l_1 as displayed in Figure 6.1b and Figure 6.1c, the difference between the required torque of one component and the available torque values of the VSM is minimized. This is done for each defined elevation angle and each angle $\xi = 0^{\circ}...90^{\circ}$, where ξ describes the angle between the saggital plane and the plane in which the arm is moving. The resulting curves for l_1 are shown in Figure H.1 for the x-component of the torque. The l_1 values for y-component show the same shape, but are flipped, so they begin at zero. Those values give a behaviour of l_1 for each elevation angle, where the torque requirement of the x-component is matched very closely.



Figure H.1: Desirable curves for the length l_1 of a VSM for selected elevation angles; the displayed curves produce a torque that can counter the x-component of the torque produced by the weight of the arm

In the minimization the range of l_1 is widened up to enable a perfect fit of the torque curves. The maximum value of around $l_1 = 29.2mm$ occurs at an elevation angle of ca. 179° . This means that for an optimal countering of the moment, the dimensions obtained with the optimization in Section 3.4 are not feasible anymore. The reason for this difference is, that in the optimization the weight function caused a good fit for an arm with an elbow angle of around $\alpha \approx 45^{\circ}$ in the region of $\gamma \approx 90^{\circ} \pm 30^{\circ}$. For a straight arm with α being zero this resulted in a nearly perfect fit from $\gamma = 0^{\circ}$ to around 45° . For larger γ the difference between the required torque and the provided torque increases as shown in Figure H.2.



Figure H.2: Absolute value of half the torque resulting from the weight of the arm, the torque produced by the VSM and the difference between both for a straight arm with $\alpha = 0^{\circ}$

This difference is removed by elongating l_1 in the way shown in Figure H.1, because an increasing l_1 changes both, the force and the lever arm of the force.

The deviation of the VSM torque and the required shoulder torqe also differs for each elbow angle α , thus an optimal solution for the transition is only obtained, if firstly the length l_1 has a wide range of motion and secondly if the described process of selecting the best suiting length is gone through for each change of angle. The latter consists of relatively simple operations and can be implemented in an controller to calculate the set point. With that given a PID controller can be programmed to adjust l_1 to the required value.

Considerations concerning the selection of the active component

For an implementation of such a mechanism with the maximum dimensions as described in Section 3.4 the size of the mechanism becomes a problem. In an arbitrary position between the two planes, the ideal exoskeleton still provides support for lifting the arm. As can be seen from Figure H.1 this requires being able to shift the pulley on the inner disc to a higher radius. As described in Reference [Bai & Li, 2019], the length of the cable and with it the

force acting in direction of l_2 is depending on the angle of distortion θ . It is calculated by

$$\alpha_{c} = \beta_{c} + \gamma_{c}$$

$$\beta_{c} = \arccos\left(\frac{a - 2l_{1}\sin\theta}{\sqrt{4l_{2}^{2} - 4al_{1}\sin\theta + a^{2}}}\right) - \arccos\left(\frac{a}{\sqrt{4l_{2}^{2} - 4al_{1}\sin\theta + a^{2}}}\right) + \frac{\pi}{2}$$

$$\gamma_{c} = \arccos\left(\frac{a + 2l_{1}\sin\theta}{\sqrt{4l_{2}^{2} + 4al_{1}\sin\theta + a^{2}}}\right) - \arccos\left(\frac{a}{\sqrt{4l_{2}^{2} + 4al_{1}\sin\theta + a^{2}}}\right) + \frac{\pi}{2}$$
(H.1)

where $a = 4R_c + 2d_c$. R_c is the radius that the outside of the cable is touching, d_c is the diameter of the cable. With the lengths of the free cable

$$\left|\overrightarrow{DB}\right| = \sqrt{l_2^2 - al_1 \sin\theta} \tag{H.2}$$

$$\left|\overrightarrow{EG}\right| = \sqrt{l_2^2 + al_1 \sin\theta} \tag{H.3}$$

the cable length becomes:

$$l_c(\theta) = \frac{\alpha a}{2} + \left| \overrightarrow{DB} \right| + \left| \overrightarrow{EG} \right| \tag{H.4}$$

With the cable length determined, the force acting on the pulley on the inner disc in the direction of l_2 is

$$\left|\overrightarrow{F_{l2}}\right| = 2\left(F_0 + k\Delta l_c\right) \tag{H.5}$$

where the cable is pretensioned with F_0 and the spring has a stiffness of k. Δl_c is defined as

$$\Delta l_c = l_c \left(\theta\right) - l_c \left(\theta = 0\right) \tag{H.6}$$

With the system described as vectors, $\overrightarrow{l_3}$ and $\overrightarrow{l_1}$ are defined as follows, the orientation of $\overrightarrow{F_{l_2}}$ is found. (H.6)

$$\overrightarrow{l_3} = \begin{cases} 0\\ l_3 \end{cases}$$
(H.7)

$$\overrightarrow{l_1} = |l_1| \begin{cases} \sin \theta \\ \cos \theta \end{cases}$$
(H.8)

$$\overrightarrow{l_2} = \overrightarrow{l_3} - \overrightarrow{l_1}$$
 (H.9)

$$\overrightarrow{F_{l2}} = |F_{l2}| \frac{-l_2'}{\left|\overrightarrow{l_2}\right|} \tag{H.10}$$

The force vector $\overrightarrow{F_{l1}}$ is then found to be the projection of $\overrightarrow{F_{l2}}$ to the direction of $\overrightarrow{l_1}$.

$$\overrightarrow{F_{l1}} = \left| \overrightarrow{F_{l_2}} \right| \cos \delta \frac{\overrightarrow{l_1}}{\left| \overrightarrow{l_1} \right|} \tag{H.11}$$

where delta is the angle between $\overrightarrow{l_1}$ and $\overrightarrow{l_2}$ that is inside the triangle built from $\overrightarrow{l_3}$, $\overrightarrow{l_1}$ and $\overrightarrow{l_2}$.

$$\delta = 180^{\circ} - \arccos\left(\frac{\overrightarrow{l_1} \circ \overrightarrow{l_2}}{\left|\overrightarrow{l_1}\right| \left|\overrightarrow{l_2}\right|}\right) \tag{H.12}$$

The resulting curves for the angle δ is displayed in Figure H.3.



Figure H.3: Angle included between $\overrightarrow{l_1}$ and $\overrightarrow{l_2}$.

With the angle found, Figure H.1 shows the magnitude of force acting on the pulley on the inner disc for selected elevation angles. It is only the part of the force displayed that points in the direction of $\vec{l_1}$. This force will have to be overcome by a motor, if a pseudo-active exoskeleton is built.



Figure H.4: Curves of F_{l1} and l_1 over the transition process for selected elevation angles. All numbers are absolute numbers.

The right y-axes in Figure H.4 displays the curves for the length of l_1 . As is seen, the behaviour of the force curves, that correspond to the left y-axes, is very nonlinear. The steplike character that is shown at some areas is a result of the numerical solving of the differential equation for l_1 . Based on the above estimations it can be said, that the motor has to provide a force of greater than 464.7N as this is the maximum value obtained for l_3 , R, k and F_0 as in Table H.1. To obtain a required power the speed needs to be known.

Variable	Value
l_3	31 <i>mm</i>
R	6 <i>mm</i>
k	$3.1674 Nmm^{-1}$
F_0	4.9665N

Table H.1: Values that are used throughout the section. k and F_0 are half of the values obtained in Section 3.4 to achieve 50% of support

To obtain a speed, the x-axis is transformed from the transition angle ξ to the transition time t_T , where the time to complete the transition is set to be $t_{Tmax} = 1s$. Then l_1 is numerically differentiated with respect to t_T . To obtain values at the very end of the curve where the highest speeds are expected, a forward differentiation is used that follows the scheme

$$f'(x) = \frac{f(x) - f(x - h)}{h}$$
(H.13)

Figure H.5 shows the required speed of the pulley on the inner disc for selected elevation angles. The highest values occur at the end of the range, where the change is largest. At the begin of the transition process the required speed is fairly uniform. The highest number that is obtained with this numerical process is $0.65ms^{-1}$.



Figure H.5: Required speed of the pulley on the inner disc for selected angles for $t_T = 1s$

The power that is required to move the pulley to the position is the scalar product of the force vector and the speed vector. Since both vectors already have the same orientation, the force F_{l1} is multiplied with the speed. Figure H.6 shows the resulting curves. It is seen, that due to the numerical process the data points are very unsteady. The maximum value obtained is about 7W.



Figure H.6: Power to move the the pulley on the inner disc for selected angles

Appendix I

Measurements of the Range of Motion

In the following pictures of the measurements for the RoM are presented.



Figure I.1: Flexion



Figure I.2: Hyperextension



Figure I.3: Abduction



Figure I.4: Horizontal adduction



Figure I.5: Horizontal abduction
Appendix J

Torque Testing with 3D Printed Pulleys

Before the pulleys have been changed, the torque that the VSM was able to provide has been measured. The results can be seen in Figure x. It can be seen that the friction has been larger than the provided torque. Hence the results are not investigated further and the spring is discarded.



Figure J.1: Results for the torque over the internal angle θ of the VSM with all measured points, the average values for loading and unloading, the corresponding standard deviation, the analytical model and the mean of the average values of loading and unloading

Appendix K

First measurement with weak spring

The conducted measurements are of two kinds. Due to the lack of an encoder, the VSM has been pushed to a certain angle at which the value given at the reading device has been noted. The data that is presented first was obtained by deflecting the VSM and reading the torque after the reading stabilized. The second data was obtained by attempting to read the torque value at the exact time the specific angle was reached. For both methods an angle increment of 5° has been used and three cycles have been conducted, where one cycle consists of a full motion to 180° deflection, setting the VSM to be stable and moving it back to 0° deflection. The settling procedure at 180° deflection in the middle of a cycle consisted of reading a torque value when coming from 175° , adjusting the deflection angel so that the deflection angle remains at 180° without outer influence and reading this new torque value before reducing the deflection angle again. By this procedure two readings for 180° deflection are obtained, the first one that belongs to the spring loading phase, and a second one that belongs to the spring unloading phase. 37 torque values have been read per cycle per phase and are presented in the following Figure K.1 and Figure K.2. In each figure, the measured data points, their average and the predicted torque curve is shown. The predicted torque curve is obtained with the Adams model described in Section x, as the numerical model is quicker to adapt to the slightly changed pulley geometry.



Figure K.1: Measured torque values after they stabilized over the shoulder angle γ for (a) the loading phase, (b) the unloading phase and (c) both phases with the mean of both and the standard deviations of the measurements



Figure K.2: Measured torque values at the time of reaching the next reading angle over the shoulder angle γ for (a) the loading phase, (b) the unloading phase(c) both phases with the mean of both and the standard deviations of the measurements

When observing the obtained data points closely, it is seen that the overall shapes of the obtained curves are similar. Both kinds of reading show a clear maximum in the centre, the data for the loading phase and the instant reading method even indicate the shift of the maximum to small γ . When the single phases are compared to each other, it is seen that the instant reading method gives a smaller standard deviation, but larger hysteresis.

Appendix L

Additionally Measured Torque Data

Figure x shows the unmodified torque data that was measured with a spring stiffness of 3.52N/mm. This corresponds to 50% compensation of the arm weight according to the model in Section 3.2.



Figure L.1: Measured torque values with 50% compensation