Design and Development of a Force-Regulated Hexapod Leg

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MASTER THESIS

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Abstract

In this thesis a concept for a new leg construction, for a force regulated Stewart Platform called Hexapod, was created. The design process for this concept was split into three major parts: the actuator, joints, and the frameless motor, which make up the dynamic elements, the legs, that move the Hexapod.

The design of these three parts was approached in a similar fashion, through four phases.

The research phase, a planning, which included establishing requirements, estimates and assumptions, performing calculations and comparing and choosing components.

The implementation phase, where through sketches and 3D software, the chosen components were employed and interfaces, which connect and mount them, were created.

The verification phase, which often occurred parallel to the previous phase, revolved around analysis of the applied components and their constructed links, upon which further iterations of the implantation phase followed.

Finally, the review phase, a stage in the project where through peer examination the design iterations were assessed, and necessary changes proposed.

Once all three parts had gone through several iterations of these phases and a satisfactory result was achieved, they were then put together creating the concept of the Hexapod leg. It was then compared against the initial scope of the project creating guidelines for future developments and testing of alternative solutions.

Sammanfattning

I denna rapport utvecklades ett koncept för konstruktionen av ett ben till en kraft reglerad Stewart Plattform, kallad Hexapod. Designkonceptet för detta var uppdelat i tre huvudsakliga delar: en aktuator (ställdon), leder och en ramlös motor, vilka utgör de dynamiska elementen, benen, i Hexapoden.

Metodiken för dessa tre delar utfördes på liknande sätt, genom fyra faser.

Undersökningsfasen, ett planeringsteg som inkluderar etablering av krav, uppskattningar och antagande, utförande av uträkningar samt jämförande och val av komponenter.

Realiseringsfasen, där genom skisser och 3D program blev de valda komponenterna implementerade samt de kroppar som fäster och kopplar ihop dessa skapade.

Verifieringsfasen, som ofta sker jämlöpande med den föregående fasen, handlar om analysering av de tillämpade komponenternas utformade länkar, vilket möjliggör ytterligare iterationer och implantationer av den tidigare fasen.

Slutligen, reflektionsfasen, ett skede i projektet där genom sakkunnigas granskning bedömdes designiterationerna och nödvändiga ändringar identifierades.

När alla tre huvudsakliga delar genomgått flera iterationer av dessa faser och ett godtagbart resultat uppnåtts, sammankopplades de för att skapa ett ben i Hexapoden. Därefter jämfördes benet mot den inledande omfattningen av projektet, vilket skapade nya riktlinjer för hur framtida utveckling och testning av möjliga lösningar bör utföras.

Acknowledgments

I want to start by thanking my Dad, Janusz Szutkowski, and my Mom, Marzena Langer. My Dad, for motivating me, not only to finish this thesis but throughout my entire studying career since primary school. Without Him this work would not have been made and I wouldn't be where I am today.

My Mom, for tirelessly working everyday to keep me healthy even when the sicknesses prove tough and complicated, prioritizing me over her own life.

I want to thank my Grandpa, my Aunts and my Mom's colleagues for supporting me medically through the difficult times caused by COVID-19 during the times of writing.

Finally, I want to thank my supervisor, Klas Nilsson, at Cognibotics AB, for showing understanding and patience during these challenging times.

Thank you all for making this project possible.

Lund, August 2022

Adam Langer

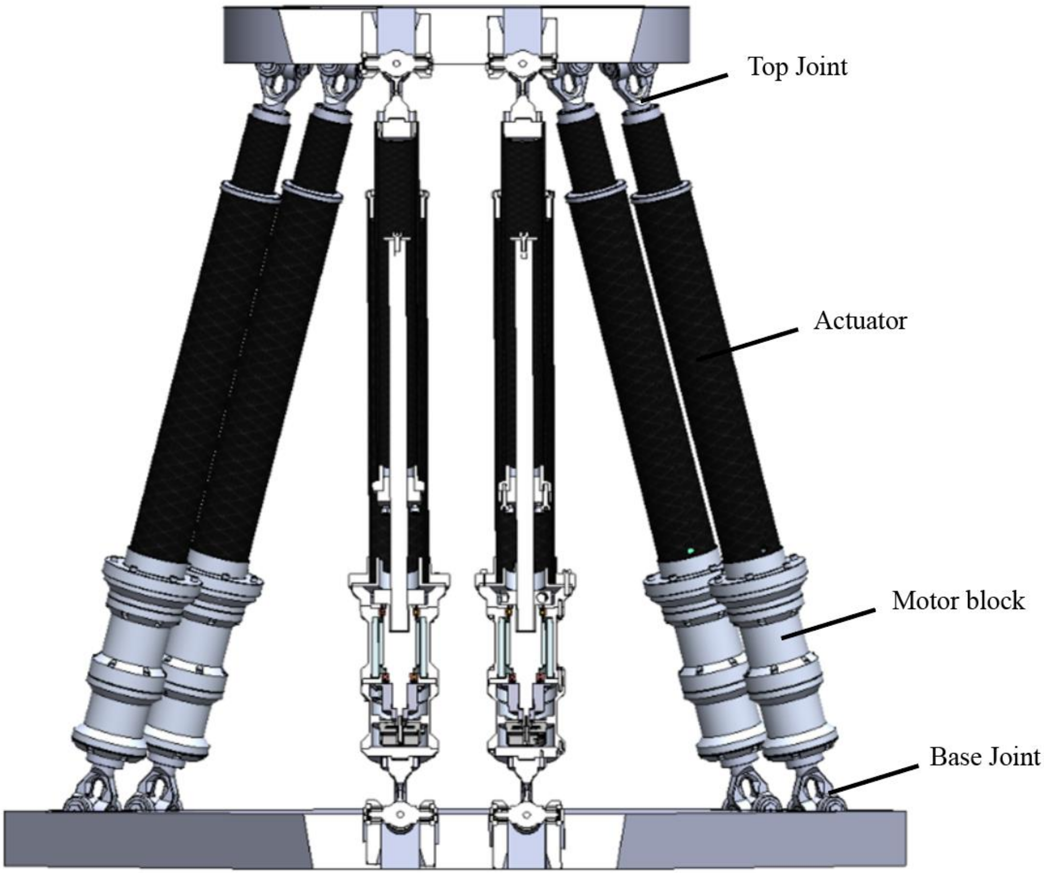

Figure 1: The Hexapod with placeholder plates. The Hexapod Leg is made up of the shown parts.

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1 Introduction

This section introduces the project and the companies revolving around it, as well as its contribution to the industry.

1.1 Background

1.1.1 Hexapod and Stewart Platform

Hexapod, as a term refers to something with six feet (from the Greek *hexápous*, “six-footed”, where *hexa* refers to six and *pous* means “foot”), is commonly used to describing insects but also mobile robots with six legs. Like how we in the current spoken language denote insects as six legged rather than six footed, and the mobile robot is defined by its legs rather than its feet, the word pod has had its meaning shifted from foot to leg. We can see similarities with camera stands called tripods, where there isn’t much of a foot present. With this in mind, the Hexapod refers to a stand, with six mechanically moving legs enabling positioning at various heights and orientations.

A Stewart platform (which was invented in the 1950s) is a Hexapod with its feet fastened to a base plate, legs jointed at both ends, also referred to as actuators as their length is linearly adjustable. These actuators are symmetrically orientated in a circular pattern, and at the top mounted to the so-called platform. The term Stewart platform originated from the need of powerful flight simulations during and after the second world war (Stewart, 1966). Its other applications are primarily in the control of telescopes in astronomy, and micro-positioning of optical equipment. The six servocontrolled actuators, give a wide range of positioning and orientations (x, y, z and rotation around those axes), with high precision and stiffness, but with a limited workspace. Due to the actuators working in parallel, the Hexapod is also a so called PKM (Parallel-Kinematic Machine) with its abundance of issues and possibilities (Merlet, 2006).

1.1.2 **Industry**

The demand for quick rearrangement of production lines of various goods in manufacturing has drastically increased over the last decades (Erdem, 2020). Currently the standard for most processes and industrial tasks is to use fixtures, which as the name implies are static (which are reconfigured manually) and fixate the workpiece. However, with the rising interest of reconfigurability, several companies have sought after the ability to program and control fixtures. The Hexapod like products currently on the market are very heavy or very expensive and have rather primitive positioning systems. They usually lack force regulation which is a requirement when converting a classic steel fixture to a servo-controlled robot fixture. A force regulated version of the Stewart platforms was adapted by Prodtex AB. Cognibotics AB intends to continue this development.

1.1.3 **Project**

Cognibotics has in an earlier project, together with Corebon (carbon fiber-based mechanics) and Saab Aero (customer and creator of use case) developed a lightweight Hexapod with combined positioning/orienting-control and force regulation. Several of the Hexapods would then be used in a series to create a fixture for larger objects like airplane wings. The force regulation allows to monitor and control the forces and torques applied at each point of contact. The project led to a fully developed prototype for fixating airplane hatches (airplane doors for loading of goods, developed by Saab for other airplane manufacturers). From testing and analysis, several areas of hardware improvements have been identified.

Due to various circumstances the development of this product has not been prioritized and not been introduced to the market through fairs or similar PR strategies. Further work was obstructed by the introduction of the global pandemic, covid-19. In later development, Corebon backed out of the project. Cognibotics, on the other hand, wants to reignite the project through development of their own hardware (previously owned by Corebon). Thus, the development of actuators, joints and a frameless motor creates the basis for this thesis.

1.2 Project objectives

The primary goal of the project is to create a concept of a Stewart platform (Hexapod) leg which not only will be lighter and better than the previously developed prototype, but also attractive to potential customers. The project is divided into three major development steps, which are the primary foci. The actuators which are responsible for smooth linear movement, being the primary focus of the costumers. The universal joints, which need to give enough angular movement to satisfy requirements, while not being too heavy or hinder the movement of the robot. The frameless motor, more compact, lighter, and specifically created for its task. Future assignments will involve analysis and testing of the product as well as creating an exposition to showcase the Hexapod for potential investors. The later tasks will be tackled after accomplishing the earlier steps successfully.

1.3 Scope and Limitations

The project is primely limited by the standardized market materials. The actuator parts are designed around components like balls screws, sealing solutions, bearings, bushings etc. Most of the design revolves around matching these parts and developing carbon fiber, aluminum or steel interfaces and joints between them. Similarly, the frameless motor is limited by the availability of servos, encoders and brakes that meet the required torques. Furthermore, it has been requested to develop as many parts as possible in carbon fiber, to make the product light and aesthetically pleasing but introducing difficulties to meet its manufacturing constraints.

The project is initially guided by previously created requirements by Saab Aero. These requirements specify certain height and workspace standards, as well as forces applied and accuracy of movement. These are detailed in chapter 3.1.1.

Lastly the design must meet installation tolerances specified by the part manufactures of the necessary components.

2 Methodology

The project was split into the development of the actuator, joints, and frameless motor. Each part would go through the same four project phases which helped organize the work process. The phases created a basis for how to tackle each new task. However, as design is a very iterative practice, many of the phases had to be repeated throughout the whole project.

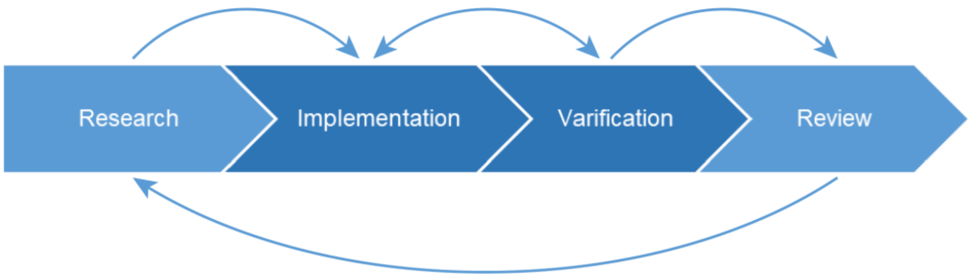

Figure 2: The four project phases.

2.1 Method reference

The method was inspired and loosely based on the double diamond method. The research phase led to a broader understanding and *discovery* of the subjects (divergent thinking), which then gets narrowed down and *defined* by implementing it iteratively (convergent thinking). Once a satisfactory result was reached it is again broadened out and *developed* by investigating safety and serviceability of components and how realistic certain ideas and solutions are in phase 3 (divergent thinking). Finally, when several ideas or solutions were verified, they were brought into review where it was *delivered* into what could be the final product (convergent thinking) (Design-Council, 2019). The major differences are perhaps that phases 2 and 3, implementation and verification, were often worked on in unison. Thus, distinguishing the characteristics of these phases may be difficult when described, but in practice rather clear.

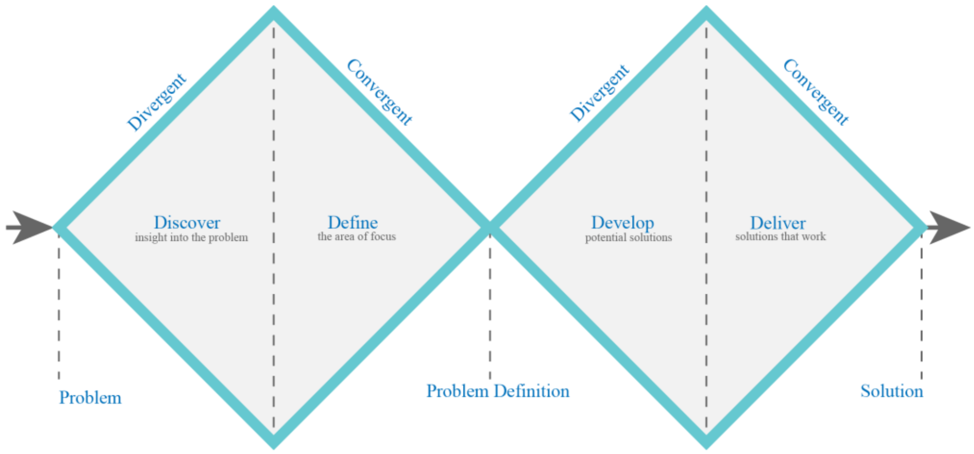

Figure 3: Double diamond methodology.

2.2 Phase 1: Research

The first phase was what initiated the whole project. To get further insight into the project and what it entails the following aspects were studied:

- Definition of the problem.
- Specification of requirements.
- Gathering of knowledge.
- Selection of components.
- Serviceability.
- Selection of tools and methods.

The first three steps were meant to create a better understanding of what the problem was, what had to be done and what were the requirements. The research started through deeper reading and learning of the subject so that all terminology and specification could be understood correctly. This was performed mostly through reading and reviewing academic papers on the subject and those recommended by supervisors. Further information was gathered through webinars and online articles. Meetings with supervisors and senior employees also provided qualitative information about their experiences and what to pay further attention too, something that was a frequent occurrence throughout the entire phase.

Next, was the process of selecting necessary components, based on their availability on the market. This introduced new information to further explore. A good understanding of the differences between the available components is required. This mean browsing through catalogs and understanding producer's often proprietary descriptions of parameters, then finding equations to calculate what sizes and dimensions are suitable and finally mailing and calling the manufacturers to discuss and hear their suggestions on the subject. Choice of components also implied analysis of serviceability.

The last step (though not definitely in this order) is the choice of necessary tools and methods to create and verify the design, which was at this point limited by the available tools at LTH and Cognibotics.

2.3 Phase 2: Implementation

This phase concerns both the complete Hexapod and each of the primary parts (actuator, joints, motor). It would contain the following actions:

- Sketching.
- Calculations.
- Investigation of dimensions and tolerances.
- 3D modeling.

Once the key components were specified through the previous phase, the first iteration of the design could start. Drawing sketches, calculating, and investigating dimensions and tolerances needed for that component to fit and finally designing around the assembly/disassembly/service requirements, were the common approaches of the development of each part.

When all available variables were accounted for, several design ideas would be sketched up and presented to the supervisor for review, which often led to further questions and new research/analysis of how to optimally approach the arrangement from a strength, mechanical and aesthetic perspective.

Once a basic idea was formed it would be brought to a CAD software (SolidWorks) and constructed in a 3D format. By downloading accurate models of the components in question a more realistic version of the ideas could be created with proper dimensions and sizes. Through iterative processes and verification in phase 3, the design would grow, leading to a satisfactory result.

2.4 Phase 3: Verification

This phase could include:

- Visual inspection of 3D models.
- Fitting analysis.
- Animations.
- Static/dynamic simulations.

- Safety analysis.
- Rapid prototyping.

Note: this phase refers to the developer's own verification and not external verifications and tests which occur after finishing the review processes and eventually building working models and prototypes.

This phase required critical analysis of achieved design results, mainly using 3D CAD software. It was necessary to control all fittings and necessary spaces between components, their strength and safety factors. Safety and serviceability had to be investigated. If available, rapid prototyping would have given further insights of the physical model, which might be missed otherwise. Furthermore, both static and dynamic simulations could verify choice of shapes, sizes and materials.

This verification phase often occurred simultaneously with phase 2. Steps like visual inspection, fitting and animations helped in the implementation stages of the design.

2.5 Phase 4: Review

After several iterations of phases 1, 2 and 3, followed by contacting the manufacturers about prices, production volumes and delivery times-and approval from the supervisor, the design moved on to the review stage. It would consist of a meeting or discussion with all relevant parties: engineers and designers at the company as well as managers and other employees who had previous experience in this project. The design was presented and explained by answering the following questions:

- How does it work?
- Why were the components chosen?
- What alternatives were there?
- What compromises were made?
- What were the costs and delivery times?

This led to discussions on whether it was feasible and well-designed or not, and what needed to change, both from a mechanical and aesthetic perspective, but also from a production point of view. Once a list of all feedback, changes and improvements had been made, the project was brought back to the start and went through every phase until it yielded a satisfactory result.

3 Hexapod general concept

This chapter will clarify and discuss the old and new concept of the Hexapod. The content of this chapter will create the basis for the remainder of the project.

3.1 Previous Design and Specifications

Note: Information gathered from colleges and supervisors at Cognibotics, through various means, such as mails, discussions or meetings are referenced with this: *(Cognibotics, 2022)*

3.1.1 **Requirements**

The project started at Cognibotics AB by investigating what had previously been accomplished. The first idea and design of the Hexapod was created by Prodtex AB who mainly develop smarter ways of industrial production. Actual work started at first in close co-operation with said company. Delivery of old project data in the form of previous 3D models, specifications, and drawings upon which the Hexapod would be further developed was expected. However, due to issues with their hardware design supplier Corebon AB, they were no longer allowed to share those. This led back to researching the data that Cognibotics AB had received previously.

Through studying an exemplary 3D model of the Hexapod, an idea of size and configuration could be made. Requirements were further extrapolated from old document (Jonsson & Borgenvall, 2020) from Saab Aero being primary customer and user of the first generation of Hexapods. Due to a new development, the Hexapod wasn’t directly related to Saab Aero anymore, those specifications where more guidelines rather than requirements.

The idea was, that several new Hexapods in a series, would make up a driven fixture with 6 degrees of freedom. They would support a single unit in production that requires regulation of dynamic loads. In the case of (Jonsson & Borgenvall, 2020) Saab Aero this meant, four robots holding up an airplane door. This scenario is what created these general specifications:

1. Remain in set position of +/-0,2mm when load is applied.
2. Expected accuracy of +/-0,2mm when moving and resetting position

3. Max downward force (at 0° offset) of 3000N
4. Max pulling force (at 0° offset) of 3000N
5. Push/pulling force from the side of 500N
6. Torsion 135Nm

   Note: These loads (points: 3-6) have been made using a safety factor of 3, according to the Saab Aero brief.

Further research made by Peter Helgosson at Prodtex AB in a case study of an airplane wing production yielded further specifications (Ilker, 2016):

7. Stiffness: Maintain position under +/- 500N in XYZ-axes (load for further parameters)
8. Repeatability: +/- 0.1mm (comparable to point 1 and 2 above)
9. Flexibility: 6 Degrees of Freedom
10. Reach: +/- 200mm in XY-axes and + 250mm in Z-axis
11. Cost: Minimize (further data not available, due to restrictions)
12. Lead team of components: 4 weeks
13. Capability to communicate with other equipment: Enable accessible data and structure transfer of information (software specification)
14. Knowledge Expectations: Minimize

These requirements would create a baseline guide for the design and decision making of all the components to come.

Cognibotics AB also specified that the previous version had issues with measured force regulation due to rough linear movement. Thus, smooth actuating will be considered when choosing components.

3.1.2 **Defining the concept**

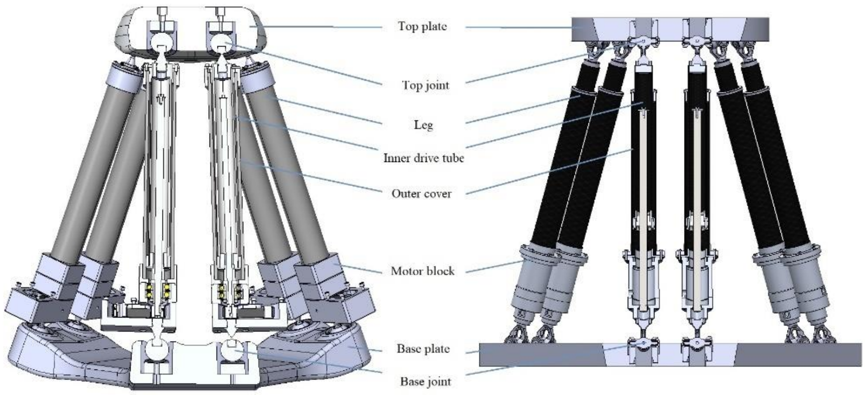

Figure 2: Concepts comparison. Corebon/Prodtex (left) vs. current proposal (right).

Figure 2 shows the previously developed Hexapod by Corebon Ab and Prodtex AB compared to an early concept of what is proposed in this thesis. By investigating these models several general areas of development were identified, these were:

- Base and top plate geometries.
- Joints.
- Drive solutions.
- Actuator designs.
- General layout.

3.1.2.1 Base and top plates

In the previous design, an optimized carbon geometry was used to enable a wider range of movement in the ball joints. The solution was quite good but required further development due to low stiffness in certain areas, leading to unwanted bending of the base plate. Furthermore, it is likely that the plates were made thicker due to the need of encompassing most of the ball joints. These problems will have to be considered in future designs of the plates while waiting for a more specific use case. An example of such could be enabling mounting of sensors or similar systems on the top plate, as well as adapting to standardized industrial fixture systems and the joints (Cognibotics, 2022).

Note: the development of the base and top plate will not be covered in this project.

3.1.2.2 Joints

For this type of robotic movement there are two major families of joints. Ball joints, and universal joints. Ball joints have a varying accuracy and getting the right type of these can be difficult due to their scarcity on the market. The previous design used specifically developed carbon ball joints. After disruption of co-operation with

Corebon AB these became unavailable for this project. Meanwhile, Cognibotics AB was developing their own unique set of universal joints which could be adapted into this device. Those were bigger and heavier than their counterparts, but there was an interest to investigate if they could be optimized. A benefit is that due to lack of axial movement, which is present in ball joints, they do not require rotational locking in the legs (Cognibotics, 2022).

The key components to investigate to create this adaptation of the joints were bearings. It was of interest to find the smallest bearings which meet the load requirements and see how their geometry influenced the strength of the joints.

3.1.2.3 Drive solution

While there are other solutions, like hydraulic or pneumatic drives, the electric solutions are most common. Electrical motors also meet the high demand of accuracy and flexibility requirements of the system, making it an obvious choice.

There are many kinds of motors available with various characteristics, and there are several ways of mounting and coupling them. The most common electric motor solutions are either servo motors, which can be DC or AC, or stepper motors.

Servo motors are good for high-speed applications with a constant torque curve. Their position can be controlled by an external driver and application with the help of encoders (or resolvers) incorporated within. For higher effects AC motors are commonly used.

Stepper motors are usually cheaper and do not need encoders to measure their position. Their angular resolution is proportional to the number of steps available, and they offer very high torques at a lower speed but to keep its position constant supply of current is required which can lead to unfavorable temperature and costs (Ring, 2014). However, Cognibotics AB was interested in developing a concept for a frameless type of servo motor for future references, and there aren’t that many varieties to choose from on the market.

According to guides made by motor producers (like Kollmorgen AB) and previous experience of fellow co-workers; the following components were deemed necessary for the motor design:

- Motor kit (including stator and rotor)
- Encoder
- Brakes
- Clamping hub
- Bearings

All except the clamping hub were key components that needed further research, which will be discussed in depth in chapter 6.1. Choosing the right motor kit creates the basis for the entire drive system. It was picked mostly based on torque requirements and size. The encoder is responsible for controlling the motor and measuring the position. The bearings are the primary part that carry axial loads in this part of the Hexapod and the brakes are the failsafe of the system which holds the robot in place when power is off.

3.1.2.4 Actuator design

There are different types of actuators; Screw driven ones, belt driven, rack and pinion actuators, rails systems. Among those only the screw driven solutions do not have a fixed length making them the only option for this application. Among the screw systems there are ball screws and sliding screws. Sliding screws are much simpler and cheaper, however are severely lacking in accuracy compared to ball screws. Their application is mostly for roughly moving objects, like opening and closing of a container for example. This leaves us with ball screws as the only option for high precision applications (Cognibotics, 2022).

Based on the previous model (by Corebon) and common practices of actuator design the following components were of interest during the research phase of this part:

- Ball screw
- Bushings and seals
- Covers
- Adhesives and gluing surfaces
- Load cells

Among those the key component was the ball screw, responsible for the robot leg’s linear movement. Choosing the right ball screw meant influencing the smoothness of the linear motion. This was a very important aspect as it allowed for easier control and force regulation of the robot. Load cells were also important for force regulation purposes, they were responsible for sensing the changes in loads on the actuators. Bushings, seals, and covers were mostly there for sealing purposes, to keep the ball screw clean from dust and debris. Finally, while gluing surfaces were of high interest due to mass optimization and strength of the actuator, the type of adhesive used was only studied for future references.

3.1.2.5 General layout

The layout was not a simple thing. The angle of the legs, their distance from each other, and their configuration (parallel and non-parallel configuration of legs), all those factors affect the robot’s movement, reach and workspace. These factors have not been taken into consideration at this stage in the project due to the lack of a specific user case. However, they are crucial steps to be investigated, especially for safety reasons. The layout should be designed with accordance to the European Machine Directive 2006/42/EC and guarantee the safety of the operator ("Directive 2006/42/ec of the european parliament and of the council," 2006).

3.1.3 **Force Calculations**

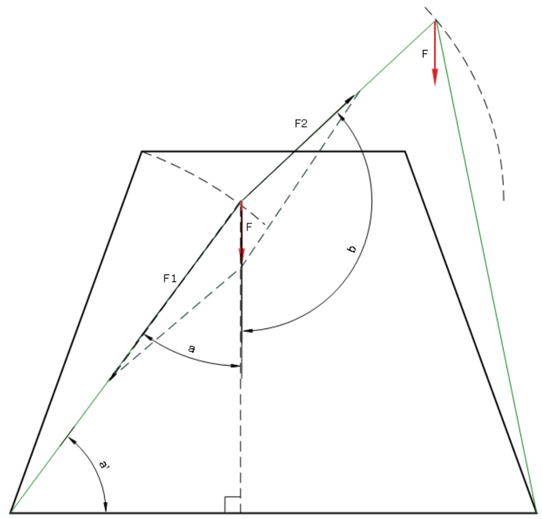

Figure 3: Model showing distribution of forces in assumed worst case scenario.

Assumptions were made that the worst-case scenario, as shown above, is when the Hexapod is positioned in such a manner that the angle $a+b$ becomes as large as possible. The larger the angle between the leg and the top plate of the robot, the greater the magnitude of the distributed forces. This angle is dependent on the permissible angles of the joints which can be altered to suit the requirements (currently +/- 50°). However, using these assumptions we can create an estimate of what these forces could be and how we should dimension the Hexapod parts.

To calculate these forces the parallelogram in Figure 3 (the one made up for forces: *F1* and *F2*) needs to be solved.

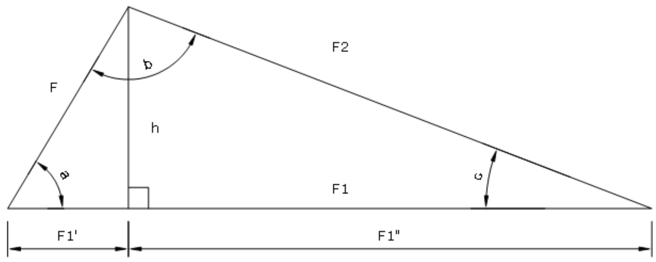

Figure 4: Representation of half the parallelogram from Figure 3.

$$F1 = F1' + F1'' \tag{3.1}$$

$$F1' = F\cos a \tag{3.2}$$

$$h = F\sin a \tag{3.3}$$

$$F1'' = \frac{h}{\tan c} = \frac{F\sin a}{\tan(180 - (a + b))} \tag{3.4}$$

$$F1 = F\ (\cos a + \frac{\sin a}{\tan\ (180 - (a + b))}) \tag{3.5}$$

$$F2 = \frac{h}{\sin c} = \frac{F\sin a}{\sin(180 - (a + b))} \tag{3.6}$$

From Figure 3 we can see that *a* is simply derived from *a'*.

$$a = 90 - a' \tag{3.7}$$

Which along with *a*+*b* are the angles given by the flexibility of the joints.

The forces F as shown in Figure 3, are an equal distribution of the load applied to the system. In our case 3000N divided over 6 legs: 500N.

Table 1: Axial forces in relation to angles in* Figure 4. *Green = applied parameters.

Given		
a'	**a+b**	**F**
45	135	500
50	140	
45	170	
10	170	
1	179	

Calculations		
a	**F1**	**F2**
45	707	500
40	766	500
45	2359	2036
80	2879	2836
89	28649	28645

The first row represents data from the previous solutions, based on their ball joint. The second row represents excepted values in this project with universal joints (green). The remaining ones showcase further extremities.

Additionally, calculations of the shearing force of 500N (on the entire system) should be done. However, they only represent 1/6 of the downward force (3000N on the system), so multiplying the above values by 1.165 should suffice. Furthermore, according to Saab Aero's specifications these forces were already multiplied by the safety factor of 3. Thus, the final axial force to be considered on each leg will be about 900N.

$$F_{\max axial} = 766N * 1.165 = 892N \tag{3.8}$$

This force shall be taken into account during selections of all the parts of the Hexapod.

4 Actuator

This chapter describes how the Actuator, a key part of the Hexapod was developed. While this process occurred iteratively as expressed in the chapter 2, for the sake of structure, it will all be constricted into the 4 phases regardless of what time they occurred at.

4.1 Phase 1: Research

4.1.1 **Ball Screw**

The ball screw is the key component of the actuator. It makes up the spine of the construction and defines how all the other components need to be designed. That is why this was the first major research point of the actuator. Exploration of this topic started by studying and understanding the different type of ball screws, how they differ and what their purpose is. The ball screw itself can by divided into two parts. The screw and the nut.

The screw can be manufactured in two ways. It can be rolled or ground. Rolled ball screws are the most common. They are much easier to mass produce due to the simplicity of the rolling system which makes them cheaper and much more available, in various sizes and lengths. On the other hand, grounded screws, are often called precision screws. These are post processed by grinding each grove in the screw to greatly increase its accuracy. Due to high accuracy requirements, the unfit parts of the screw are often cut off, which makes these screws very costly or unavailable at certain lengths (Layosa, 2015) and price is exponentially proportional to its length.

While the nut has thousands of varieties and types, depending on the manufacturer and their own ideas and technologies, they can often be categorized. Usually these are based on how the ball circulation in the nut looks like. A simple example of this would be the balls circle back to the start through a tunnel grooved-out in the nut, sometimes referred to as internal return system. While external return systems could have pipes that let the balls return to their starting points. These two categories often have subcategories which can also differ both in name and method depending on the producer.

Another important categorization is whether they are preloaded or not. Preload means applying a certain load axial, or radial in the production process which ensures constant contact of the balls between the nut and the screw. Essentially eliminating any play. The amount of preload varies and can have consequences on the lifetime of the product, which is why it is often the most important step in choosing the right nut for the application.

Choosing the right type of nut can be very difficult due to a lot of data which must be taken account for. It is possible to investigate and read up on the various types and their pros and cons, as well as do extensive calculations on max speed, acceleration, preload and other factors. In practice it is much simpler, knowing the requirements you have for the ball screw, to contact producer's sales engineers and let them do the selection.

Generally deciding what kind of screw and nut you need starts with determining the required lead, load rating and the total length required (Misumi, 2021).

The lead is the displacement of the nut during an entire screw rotation. Which is strongly related to the speed of the screw. The lead is a function where it must be equal or larger than the desired speed divided by the maximum rpm of the motor.

$$L \geq \frac{Vmax \times 60}{Nmax} \tag{4.1}$$

Where Vmax is the desired max speed in mm/sec, and Nmax motors max RPM (Misumi, 2021).

In the actual project the adjustment speed was not a critical parameter, so selection does not need to be based on minimum lead value. However, the lead cannot be too large due to possible limits of the resolution of available servo encoders. This will be further explained in later chapters. Finally, as precision is the main objective, choosing smaller leads is preferable.

As calculated in chapter 3.1.3, the applied forces on the screw are expected to reach 900N. This will be accounted for when choosing this element.

Finally, the total length should be selected. The critical parameter here is the length of the stroke, determined by the active area of the screw and the nut size. Other lengths such as the mounting length (in the motor block), possible space for bearing and other interfaces, have importance when establishing the length of the leg.

Those are the general steps for selecting ball screws, however in this case other parameters where of equal importance, like the screws smoothness which is allows easier force regulation of the system. Thus, selecting a precision screw is preferable.

4.1.1.1 Stroke Calculations

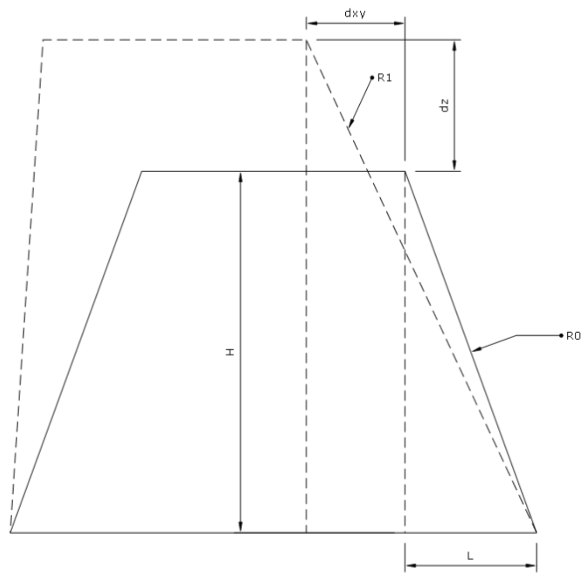

Figure 5: Models used to calculate the required stroke due to displacement of the top plate.

The necessary stroke of the actuator is given by differential of its min length and max length, expressed bellow as dR:

$$dR = R1 - R0 \tag{4.2}$$

Where *R0* is the total length of the actuator and can be expressed as:

$$R0 = \sqrt{L^2 + H^2} \tag{4.3}$$

Where:

$$L = \frac{L_{base} - L_{top}}{2} \tag{4.4}$$

Meaning, that the length of the actuator needs to fit in within possible parameters of the entire robot, depending on use case. This length, expressed as R0, is defined by the minimum length of the ball screw and its housing, two times the length of the joints (minus their depth of installation) and the size of the motor block.

R1 is the length of a completely extended actuator reaching its desired extremity of *dxy* and *dz,* which can be expressed as:

$$R1 = \sqrt{(H + dz)^2 + (L + dxy)^2} \tag{4.5}$$

This means the desired stroke can be calculated as:

$$dR = \sqrt{(H + dz)^2 + (L + dxy)^2} - \sqrt{L^2 + H^2} \tag{4.6}$$

By using values from the previous Hexapod design an estimated stroke of 308mm is needed as shown in *Table 2Table 2*.

Table 2: Stroke calculations based on given robot geometries.

Stroke								
Given					**Calculations**			
dxy	dz	Lb	Lt	H	L	R0	R1	dR
200	250	515	200	475	157,5	500	808	308

4.1.2 **Bushings and Seals**

Choosing one depended on what was really needed. Bushings have some sealing capabilities due to them having such a small tolerance to the inner shaft, they prevent larger debris to enter the system. However, they struggle where seals shine, that is when micro dust and liquids get involved. Bushings can have other purposes. Like taking up forces and vibrations perpendicular to the linear screw direction while allowing a near frictionless movement. In the previous Hexapod design, they were also used for rotational locking of the inner shaft.

Choosing bushings can get very technical too. There are a lot of different materials and composites to choose from which have many different aspects. Such as self-lubrication, maintenance free operation, dirt environments, corrosion resistance, max temperatures, and max loads (SKF, 2010). Furthermore, it turns out plastic bushings work well with carbon fiber parts. Another major requirement was the shape and how it was supposed to be fitted. There are versions with flanges and there are versions without, which drastically changes how they can fit and be mounted.

4.1.3 **Covers**

The most common design for actuators is to have a static house around the screw, so that when it is in its lower position it covers the entirety of the inner shaft. Historically other solutions have been used, a common one being bevels. The purpose of covers has mostly been to protect the ball screw from contaminations in-between the two sealed points, for this purpose bevels are by far the superior choice. They are lighter, cheaper, more flexible, and only really lack in esthetics, arguably. However, some actuators, like mentioned earlier, house bushings which are meant to pick up perpendicular loads. In those cases, hard covers become the only option. Another interesting find made during this research phase was something called telescopic springs. They are like bevels in the sense that they extend and contract with linear movements. They are made of a thin metal sheet that wraps around itself in a spiral. Due to linear movement of the screw, it elongates along the axis of the spiral. These covers are common in mills due to their strength of keeping larger debris and chips away as well as liquids, common contaminations in the milling process. Though they lack in keeping out finer dust. They are also, arguably, more pleasing to look at than plastic bevels (Cognibotics, 2022).

4.1.4 **Adhesives and Gluing Surfaces**

For housings and elements made of metal, carbon fibres or other non-fat techno plastics the use of glues or other adhesives can be necessary. It has been researched only partly in this project, as the final product design is not ready yet. Glues and adhesives will become more relevant at the manufacturing preparation stages of the project.

Generally, there are many things to consider when selecting adhesives. The material it will be adhering too, the work and storage ambient temperatures, UV, fluids and chemicals exposures, etc. It’s worth investigating what kind of stress will be put on, vibrations, permanent load, shock or shear forces.

On the other hand, gluing surfaces (area, shape and grading) should be accounted for. For example, the shape of the area has a direct influence of how much force can be applied to the adhesive before it loosens up. Cognibotics AB had made a test study on this in other projects using inserts in carbon tubes. They found out that using conically shaped inserts with a 1-2° angle gave them 4 times greater max load in comparison to straight cylindric shapes. In fact, in many of their tests the metal of the insert tore up before the adhesive let go (Ranefjärd, 2022).

4.1.6 **Load cells**

Load cells are small devices which are used to measure the forces and torques applied on the subjects to which they are attached too. These are necessary to measure and regulate the forces and torques created or received by the Hexapod.

Generally, there are four types of load cells:

- Strain Gauge load cell (Wheatstone Bridge)
- Piezoelectric load cell
- Hydraulic load cell
- Pneumatic load cell

Strain Gauge load cells are the most common choice dude to its durability, high versatility, and cost efficiency.

Piezoelectric load cells work by generating voltage through a piezoelectric material relative to the applied load (hence the name). It is commonly used for dynamic or frequent measurements, unlike static loads.

Hydraulic load cells are used when the environmental conditions do not favor electrical parts.

Pneumatic load cells are mostly used to regulate pressure that escapes through a nozzle or other venting solutions.

Load cells can be further defined by their shapes depending on their application, for example: S-beam load cells are commonly used for tension measurements, and Button load cells are good for application where there is a lack of space (Scales, 2019).

Choosing the right sensors revolves around understanding your application. Whether it supposed to last a long time under unforeseen stresses, work remotely or have high precision. Furthermore, deciding on the type of load, capacity, size and weight of the cell is important. Lastly, adapting to the needs of the software engineer, in charge using the sensors to regulate the forces, might be necessary.

Strain Gauge and Piezoelectric load cells are applicable for the Hexapod. The first one is easier and more forgiving in its installation and use. The latter one works with higher frequencies of force changes which might be necessary for our application but requires a more complicated mounting interface. The final decision on this matter was not specified and will be further researched in the future.

4.2 Phase 2 & 3: Implementation and verification

4.2.1 **First Version**

Once the specifics of the main component of the actuator (ball screw) had been established the implementation phase started. This was done through downloading the appropriate CAD model from the manufacturer's website, then designing the interfaces and housing, to which ball screw will be attached.

The chosen nut to be used was the SDA-V 1605V-3 by THK Ltd (see figure below), with a precision ball screw for more smoothness to further enhance force regulation of the system.

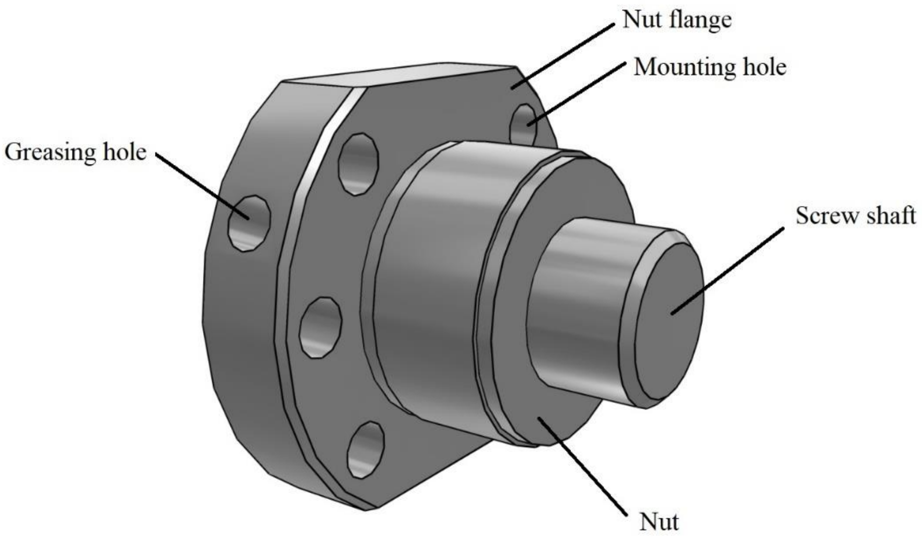

Figure 4: SDA-V ball screw explained (courtesy THK Ltd)

Based on research on earlier designs of actuators, the screw shaft had to be attached stiff to the axle of the rotor of the driving motor. Meanwhile, the nut would be attached to the driving tube. The bottom part of the motor was attached through joints to the base of the Hexapod, while the driving tube, responsible for the Hexapod's movement, would be attached by joints to the top plate of the Hexapod.

Ideas of attaching the nut trough adhesives or heat fitting to the driving tube, to save weight was analyzed. However, it was not recommended by the manufacturers due to potential heat buildup caused by the ball movement, which could lead to deformations. Furthermore, permanently attaching the nut to the driving body of the actuator could lead to service issues. Should the gluing be misaligned, or some component of the actuator damaged then the whole ball screw would have to be replaced, which would be very costly as this was the most expensive part of the design.

The solution was to create an insert that would be glued, or heat fitted to the tube instead. This adapter would then be screwed on to the nuts mounting holes. Cheap, replaceable, and easy to manufacture.

This insert could be mounted exterior or interior to inner tube and glued or heat fitted. The sizes needed to be chosen according to transferring forces.

The moving part being the inner tube of the actuator, made it rather obvious to be fitted on the inside. Both adhering options are good, however for carbon fiber the most common practice is gluing. The thickness had to be as thin as possible to minimize wight while accounting for the screw clearances and tolerances. This was decided according to testing done at NASA (Rivera-Rosario & Powell, 2017). The rest of the actuator was constructed based on common actuator designs as well as the previous versions made by Corebon.

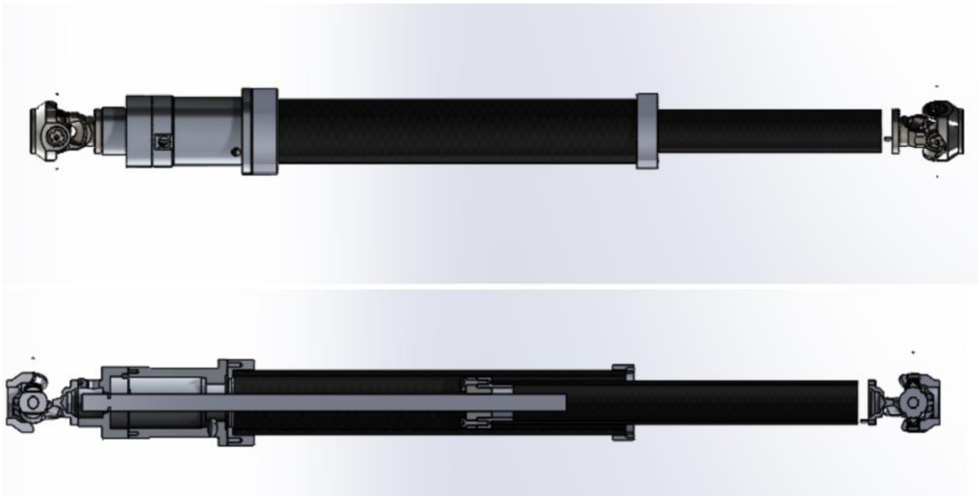

Figure 5: First actuator design, with placeholder joints and motor housing proprietary to Cognibotics AB.

Disclaimer: *Bushings were deemed unnecessary in this project. However, it was an important realization to be made, how that decision was achieved is described in the next chapter (4.2.2). If this is not of interest skip to 4.2.3.*

The challenge of the first version started when trying to figure out where the bushings should be. In the previous design made by Corebon the inner tube was made from aluminum, however in this project we were trying to make as many parts as possible in carbon fiber, to push its limit and for weights saving purposes. An aluminum tube allowed to create specific slots and fittings for the bushings, however carbon fiber cannot be post processed in the same ways, which meant the bushings couldn't be on the tube anymore. The only parts where bushings could be eventually mounted on were the aluminum inserts. The following sketches contain proposals for such arrangements.

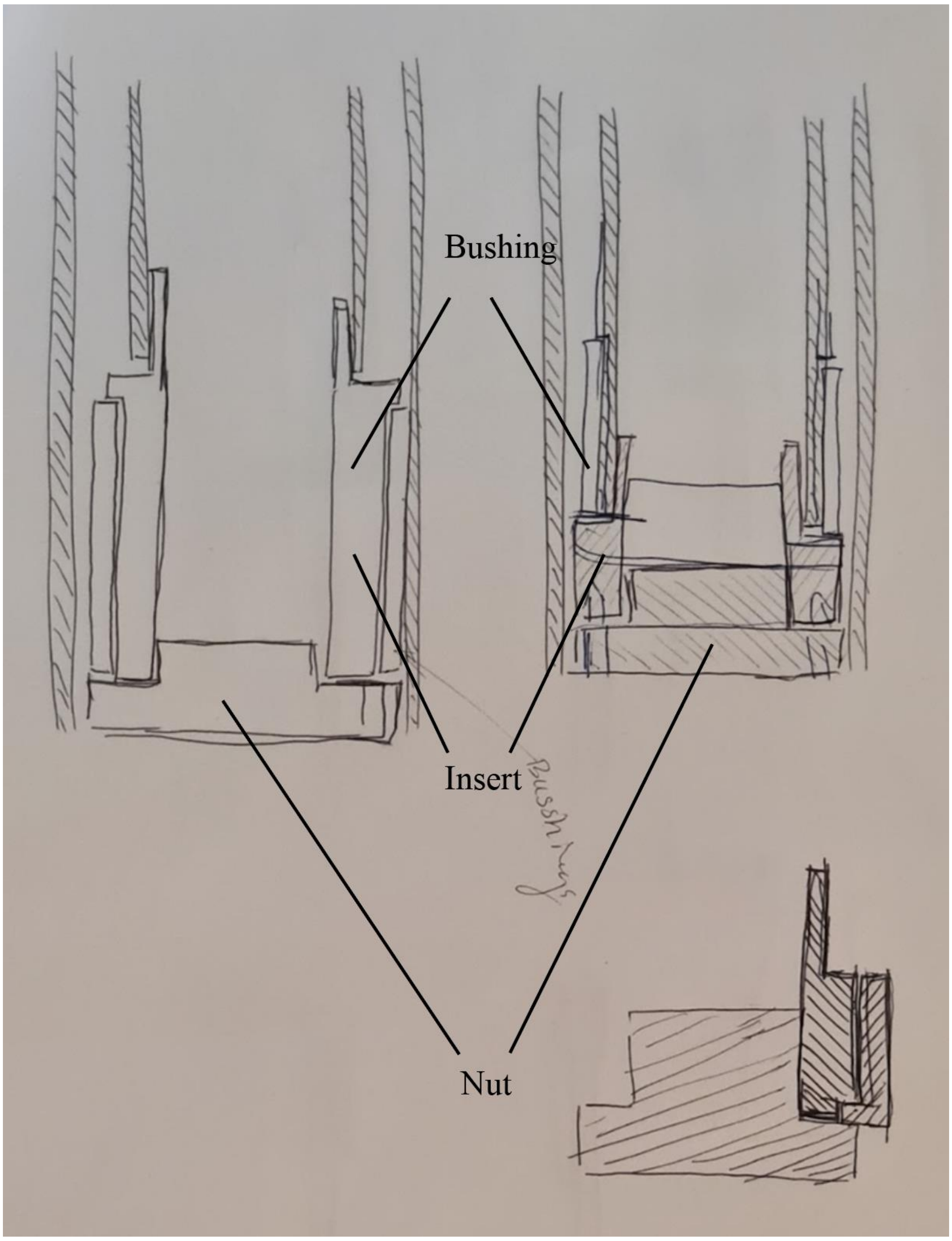

Figure 6: Some sketch ideas of how to modify the bottom insert to mount bushings and the ball screw nut.

Several ideas were considered but the optimal seemed to be to have the nut on the outside of the inner tube, allowing use of its aluminum surface for the bushing to mount on. It would then be fastened by screwing on the nut to the adapter pressing the bushing in place.

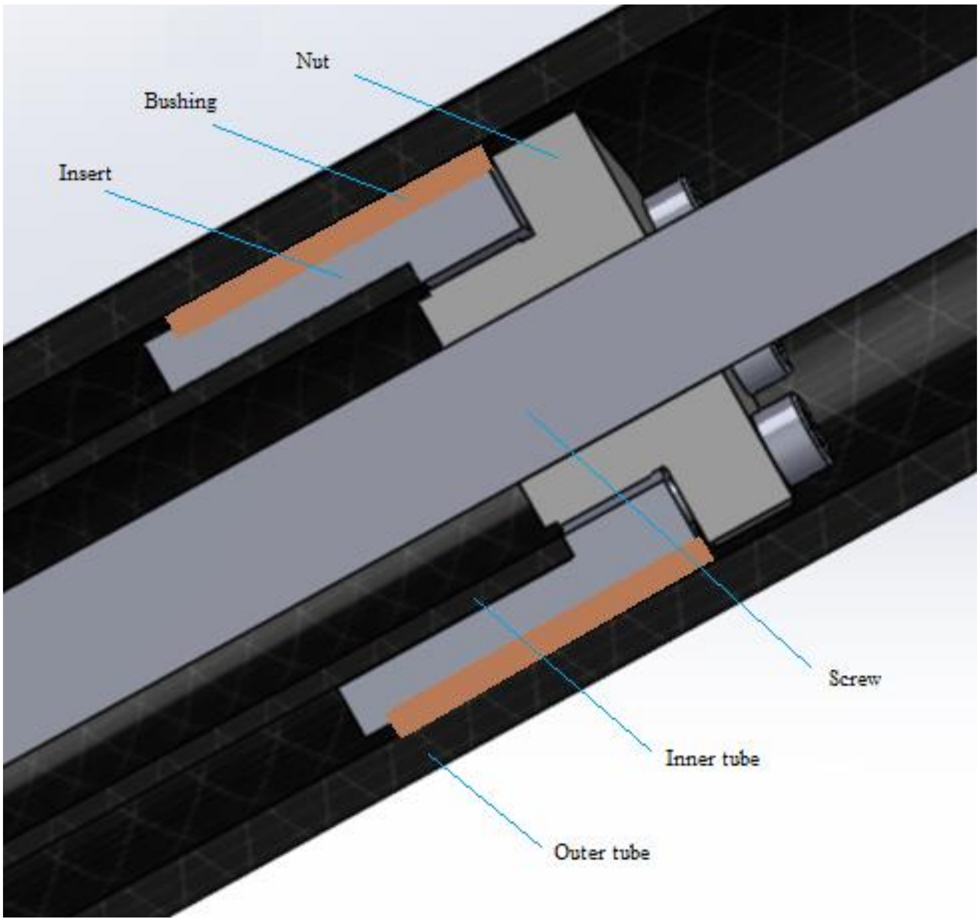

Figure 7: Bottom insert attached to the outer side of the inner tube with a slot for a bushing.

There was also a need for a bushing on the top end of the outer tube, as seen Figure 8. Mounting it with a flange was easier here as there was more space available.

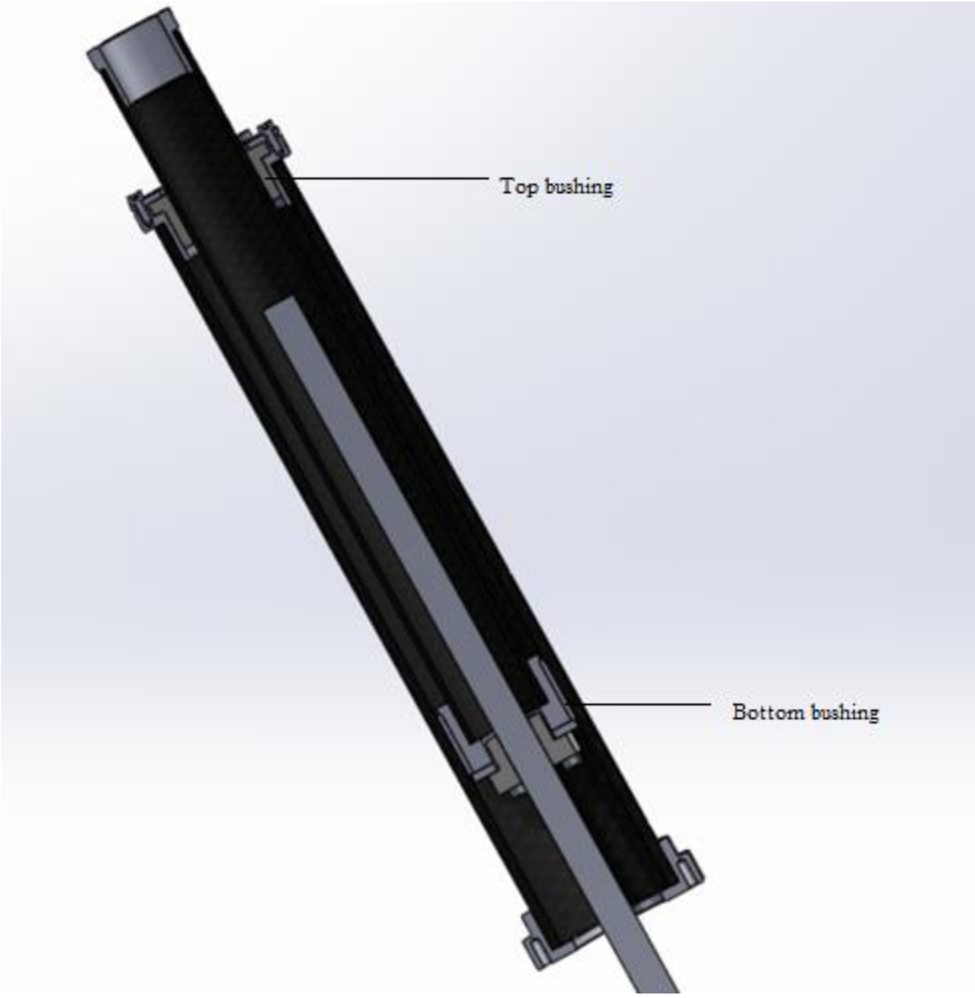

Figure 8: First iteration of the actuator.

4.2.2 Bushing analysis

These bushings caused some difficulties. The available dimensions of the bushings on the market were not ideal. Both bushings had to fit inside the space in between the two tubes and most combinations of bushings made the outer tube excessively large. Furthermore, this solution required longer legs to meet the required stroke constraints as inserts collided with each other in the stroke's terminal position, so the tubes needed to be made longer.

The second problem was that disassembly of the inner tube after gluing the top insert to the outer one was not possible. Another issue was that lubrication of the nut was impossible in this configuration. Accessing any components in the actuator would require breaking parts.

Furthermore, presenting this idea to the supervisor and discussing the difficulties of having to create bushings mountings and choosing the right type of bushing, material, manufacturer, size etc., made me realize, that it wasn't clear why they were necessary between the tubes in the first place.

This attempted design was based on other actuators on the market and previous versions made by Corebon AB, so the first assumption was that bushings were needed.

Discussion with other coworkers, who designed bushings for actuators in other robots, clarified that they were mostly used to dampen vibrations. For robots that continuously move fast over a certain path can these vibrations become a serious problem. However, the Hexapod doesn't move all that much. It's more about moving to a certain position and staying there, which means whatever happens during the movement isn't as critical, thus if vibrations occur around the ball screw it has negligent impact on its performance.

Corebon also used specially designed bushings with a flat side to lock the tubes from rotating during the screw's movements. However, with universal joints, instead of ball joints, which were to be used, you avoid that issue entirely.

This rendered the use of bushings in our case, entirely unnecessary. Furthermore, the outer cover wasn't needed other than to keep dirt and pollution away from the screw, so the top bushing could be replaced with a seal.

4.2.3 Second version and cover analysis

With the bushing analysis done, the focus of the challenge shifted on to the covers and sealing of the system. As well as just finishing up all the details of the actuator: lock rings, screws, tolerances, attachment points, thicknesses, wight optimization etc.

Figure 9: The second version of the actuator using a carbon tube as cover.

The first step was to get rid of the bushings. This led to a simpler design of the nut-adapter insert. The upper bushing was replaced with a simple plastic seal.

To allow easy de-assembly and maintenance, the bottom outer-tube interface and the upper inner-tube interface was split into two parts (right side and left side of Figure 9, respectively. This can also be seen better in Figure 14 & 15).

As for covers, there were several options, for example bellows, as in this old ABB robot (Figure 10). The outer cover did not carry any perpendicular forces, so there are no real requirements about stiffness of this part. However, since this was supposed to be one of the first models and it was meant to attract sponsors, aesthetics was very important. Bellows, while light, good at sealing and cheap were not aesthetically pleasing. They are also prone to wear and need to be checked periodically. Furthermore, there wasn't any geometrical restriction which would impose contraction of the covers, unlike the example shown below. Additionally, for bellows to contract from 300mm to 20mm or so, they would have to be quite wide. As mentioned in chapter 3.1.2.6 this could prove problematic due to the legs being close to each other.

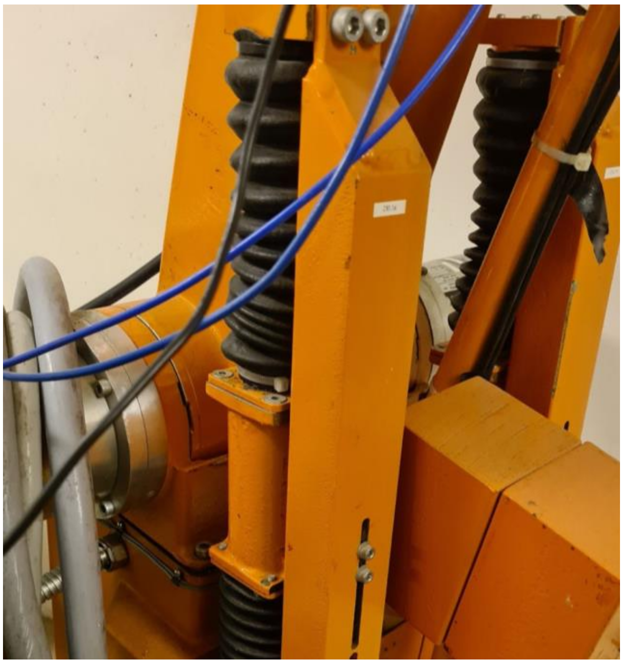

Figure 10: Bellows on an old ABB robot.

However, this inspired other ideas, namely, to look for something like metal bellows. Which led to the metal telescopic spring covers.

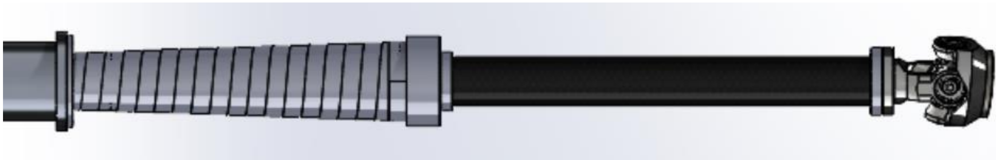

Figure 11: First version with metal spring covers.

This was an interesting idea, and especially due to being unique. It stood out from other solutions with bellows or tubes, and differentiated itself from previous designs by Corebon, which would avoid any identity disputes.

Though, they were not as ideal as one could have thought. First, they were severely lacking when it comes to sealing. They wouldn't protect against finer dust and debris. Furthermore, unlike initially thought, for them the work properly the thicker part of the spring had to be placed in the upper position when extended, which wasn't very aesthetic (see Figure 11). The idea was that the actuator is somewhat ticker and heavier at the bottom and slimmer at the top, so having the spring the other way around would have created a smooth transition between the two. Having it the way as shown above makes the actuators seem uneven and bulky.

Lastly even through scouring the internet for as thin metal sheet variants as possible, having a spring increased the weight by 250g over the carbon tubes. Which for the whole system meant 1,5kg increase in weight.

Though, there was an interesting motion of perhaps making them in carbon. This would have to be further investigated, but at this stage we wanted to keep the possibilities open, so the actuator was designed with both cover options in mind. It was made interchangeable, which is why the nut insert and the bottom out tube insert have an extra layered part, which works as an adapter between the two options.

4.3 Phase 4: Review & latest version

4.3.1 **First review**

After some minor discussions with the supervisor, it was decided that it was time for the first review. Meaning we sought the help of other experienced coworkers to hear their insight on the project. The review started with a presentation of how the actuators was designed, why we had done things like getting rid of the bushings and presented the various ideas we had with the different covers. There were some interesting discussions regarding the covers. A lot of the opinions where similar to ours, meaning that bellows seem to be the best option from a technical point of view, or just some cheap plastic cover. It was also agreed upon that the carbon fiber tube is the most pleasing option and fixating on the telescopic springs seemed rather non beneficial.

Following that several points of improvements were brought up by engineers and other co-workers at the review at Cognibotics AB:

1. Close the gap between nut and the bottom inner interface.
2. No inner ring (for the telescopic spring) in between the bottom inner interface, instead have it below.
3. Top insert should have a circular pattern of holes rather than one hole.
4. All inserts/adapters need centering flanges
5. Insert with the seal needs to be in two parts with a lock ring from above.
6. Critical gluing surfaces should have 1-2^{o} angle chamfer for significantly better strength and fitting.
7. Have easy access to the ball screws maintenance (greasing) port.
8. Look into how the ball screw stops at the end of the nut
9. Consider use of the stopping material, metal to metal is not ideal.
10. Find a good way to attach a force regulation sensor and to have the cables to it attach to the actuators without getting in the way.

4.3.2 Final version of actuator

First point; The bottom inner interface, to which the nut was screwed on, had a deliberate gap between the walls to allow for possible heat expansion of the nut. This was deemed unnecessary as the heat build up in the nut was thought to be negligible.

Second point; there was a need to have an adapter attached to the nut to fit the telescopic spring, should we want to use it in the future. This was initially placed between the nut and the bottom inner interface, to cover the nut with the telescopic spring cover. However, this meant that an extra washer would be needed to not change the geometry of the actuator when a regular cover tube is used instead. An unnecessary increase in weight, and it was believed that most of the contaminations happens through dirt on the screw turning into the nut during actuation. Thus, covering the screw would suffice, meaning this adapter could instead be attached below the nut if needed.

Point three and four: Where just a quick change that made sure all components are aligned properly in the actuator.

Point five was brought up, as the initial thought was to replace the seal by taking of the cover, which would simplify and minimize the wight of the top outer interface. However, the seal would need frequent replacements so an easier access, in the form of a two part/lock ring interface, was more desirable.

Point six was discussed after the review with members designing actuators for other robots. They explained that their previous research and testing showed that an angled gluing surface increases the strength of the adhesive by factor of 4, as mentioned in chapter 4.1.4.

Point seven further mentions the idea of serviceability, to create an easy access to the nut for greasing purposes without having to remove the whole cover, as this must be performed quite often according to Cognibotic's experience.

Point eight and nine address the stopping mechanism. The stopping mechanism needed to be added to the actuator, as the nut did not have such a function built in. Stopping material made of metal was shown to be necessary for calibration requirements. However, eventual use of POM-C or other techno plastic with good dimensional stability could be considered, since metal to metal stopping isn't optimal during regular operation of the actuator.

The last point; the force cell and cable management could not be resolved at this stage due to lack of data on this subject (will come in further iterations of this project). However, the space needed to mount the load cells was considered. The sensors and their position may change with further requirements created by the software engineers.

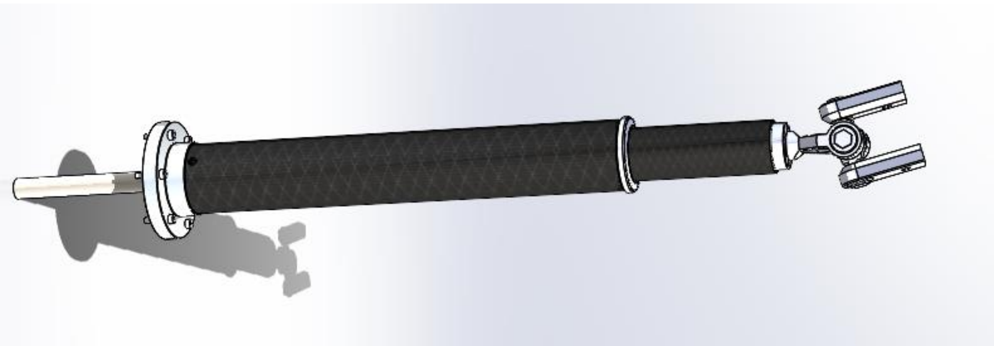

Figure 12: Latest version of the actuator.

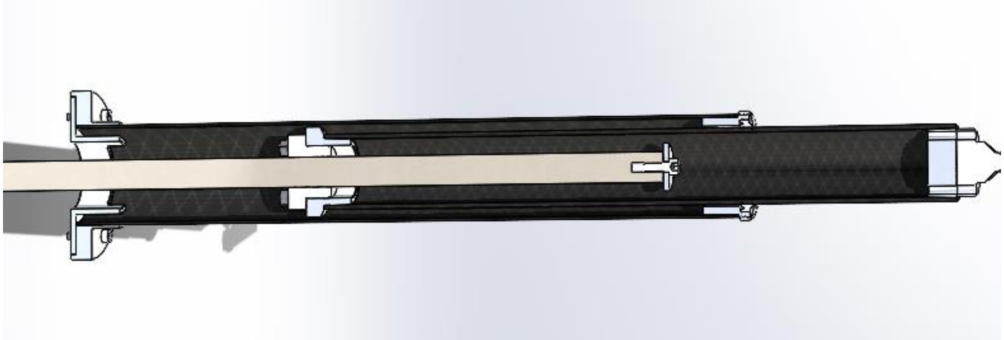

Figure 13: Cross section of the latest version.

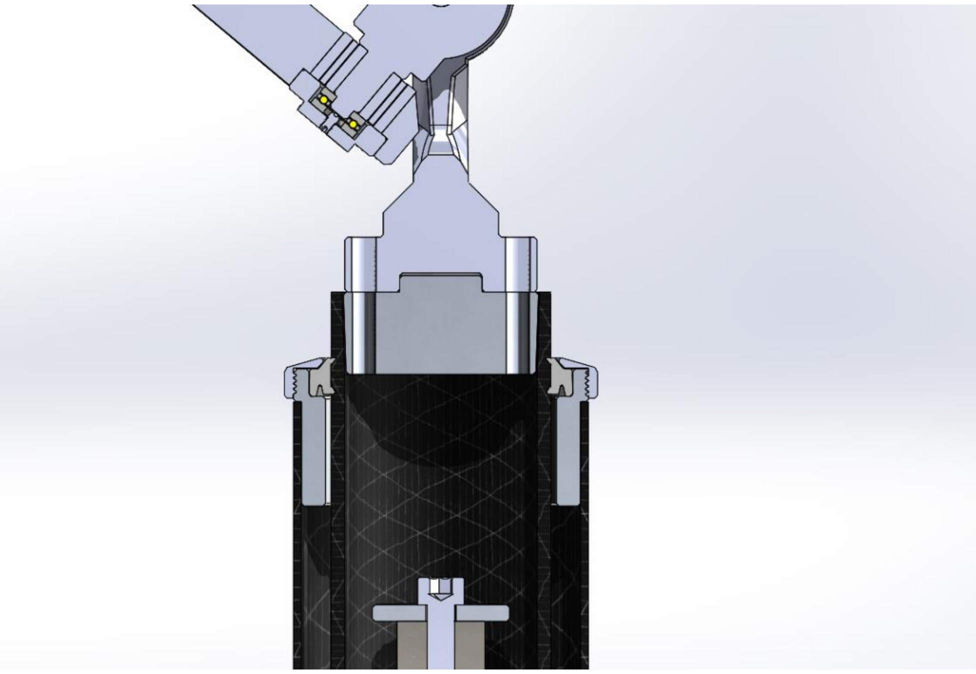

Figure 14: Detailed view of outer tube insert with seal and lock ring, as well as screw lock ring, and the innertube top insert attached to the universal joint.

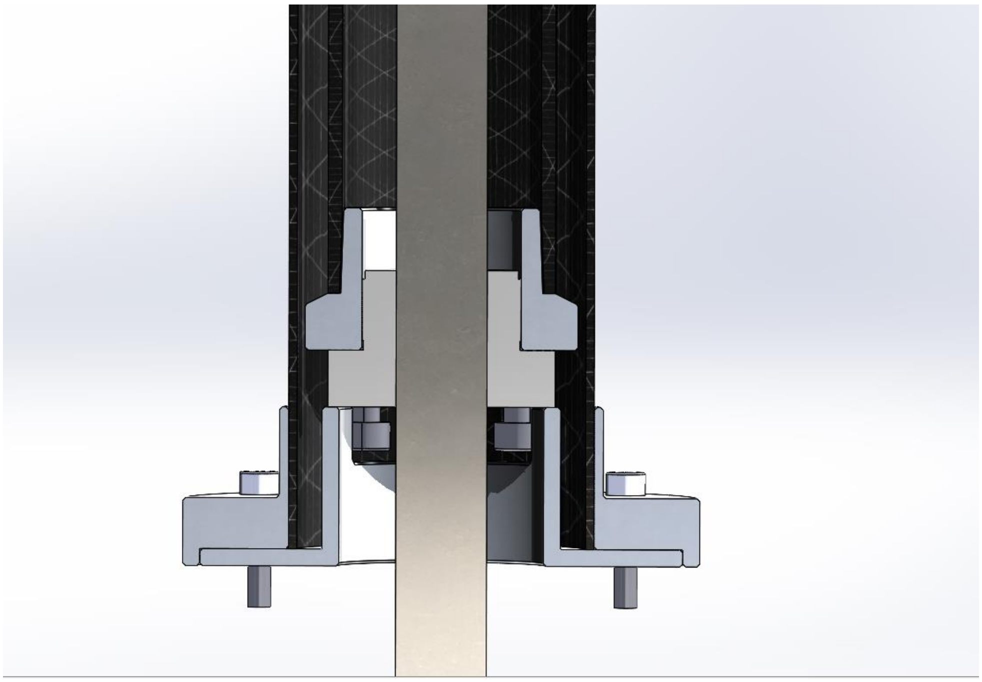

Figure 15: Detailed view of bottom inserts and cover/stopping mechanism, Inner tube insert/nut adapter.

5 Joints

This chapter describes how the joints were adapted from their original version made by a fellow co-worker at Cognibotics AB. As in the previous chapter, it will be structured by following the methodology rather than chronical time.

5.1 Phase 1: Research

The requirements and specifications for the joints are the same as the ones established in 3.1.1.

5.1.1 **Previous design and universal joints.**

A universal joint (U-joint) is a device which allows to transfer rotary motion through inclined axes. It can be even used as flexible connection between two components which moves against each other in angular manner (Cognibotics, 2022).

U-joints are made up of two U shaped hinges oriented 90 degrees to each other. These are rotating around two shafts which are intersecting (cross shaft). This allows for one of the U-shaped parts to practically move in a half sphere workspace around the other part, depending on the allowed max angles of the hinges tilt. Unlike ball joints, which in the Hexapod's configuration can rotate indefinitely around the actuator's axis, universal joints, due to being statically fixated to the top and bottom plate, cannot. This rotational locking was beneficial for our system, as there weren't any such mechanisms built into the actuator (Cognibotics, 2022).

The previous Cognibotics AB design of U-joint was studied by creating an exploded view of off it.

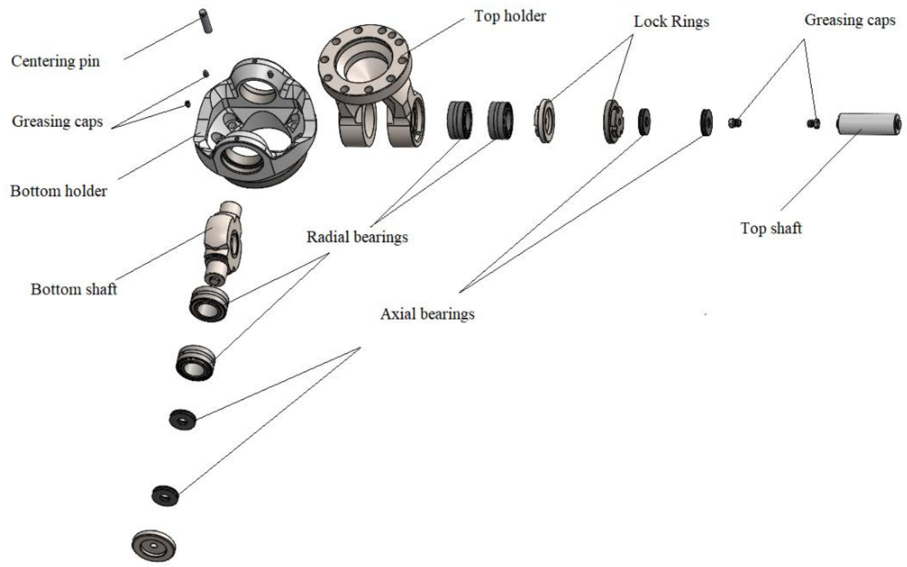

Figure 16: Exploded view of universal joints designed by Cognibotics AB.

Through this example design, it was possible to study what components were needed, what they were made of and how they were aligned to work together. Especially important was analysis of the components shape, giving as much movement as possible around each axis. As explained by the creator some parts ware intriguingly designed such as, to make it possible to install the parts while keeping this design as compact as possible.

At first glance the joints themselves seemed well designed, practically perfect for our purpose. The only problem was their size, about twice as big for our application, thus the ideal goal would be to simply scale them down to the appropriate size. For that, it seemed, that all that was needed to be done, was to find corresponding standard components, in this case the bearings, in smaller sizes.

5.1.2 Shaft deformation and stress calculations

Due to the complicated geometries of the joint shafts, calculations of stress and bending are very difficult without using simulations in CAD.

To estimate necessary diameter and length of the shafts, calculations were done only on the top shaft, because its geometries are manageable.

The formulas and stress limits for the following calculations were taken from *Formler och tabeller för mekanisk konstruktion (Björk, 2013)*.

5.1.2.1 Max stress

$$\sigma_{b_{max}} = \frac{M_b}{W_b} \tag{5.1}$$

For point load in the middle of the top shaft (equally divided load from the bottom shaft, for simplicity as a middle force) freely supported at the ends (where bearings were placed), we have the bending torque:

$$M_b = \frac{FL}{4} \tag{5.2}$$

Section modulus for round shafts is:

$$\boldsymbol{W_b} = \frac{\boldsymbol{\pi d^3}}{\boldsymbol{32}} \tag{5.3}$$

This gives max stress:

$$\sigma_{b_{max}} = \frac{8FL}{\pi d^3} \tag{5.4}$$

And diameter:

$$d = \sqrt[3]{\frac{8FL}{\pi \sigma_{b_{max}}}} \tag{5.5}$$

Where L is the length between the bearing's centers, d is the shaft's diameter and F is the applied force of 900N (according to calculations in chapter 3.1.3).

Allowable stress in static loads for engineering steel SS-1655 (S235JR EN10025-2), for diameters smaller than 16mm is 240N/mm^2.

From these values the combination of minimum diameter and length can be obtained.

For example, for a chosen length of 20 mm, the minimum diameter using the above-mentioned steel is 5.76mm.

5.1.2.2 Max deformation

To calculate deformation the following formula was used:

$$\delta = \frac{FL^3}{48EI} \tag{5.6}$$

Where E is young's elasticity module and I is the area moment of inertia, given by:

$$I = \frac{\pi d^4}{32} \tag{5.7}$$

For engineering steel SS-1655 the elasticity module is 210GPa (210000N/mm^2)

These calculations give an estimated max deformation of 0,006mm.

According to the requirements (chapter 3.1.1), the accuracy was +/- 0.1mm. Assuming no additive fault of legs positions, it translates to a leg length accuracy of 0.1mm. This was further split between all parts of the leg (encoder, screw, joints and possibly other interfaces). In the worst case, according to the calculations the total displacement of the joints was 0.024mm, (assuming the shafts are the weakest links). This gave about 0.076mm as a required positioning accuracy of the remaining parts (see later chapter for further analysis).

5.1.3 **Bearings**

This was probably the most challenging aspect of the joint design. As stated above the goal was to find the corresponding bearings in maybe half the original size. However, this was practically impossible.

Our application was looking to use needle bearings, both radial and axial. While the most common bearing is probably the ball bearing, its application is mostly for high-speed scenarios while having low friction, noise and vibrations, which reduces their wear and tear (CMC, 2021). Needle bearings on the other hand offer a more compact design while withstanding 2 to 8 times greater load than ball bearings (NTN-Europe, 2022). Since the joints won't be rotating with high RPM, in fact they won't be doing any revolutions at all, the bearings speed or friction doesn't matter to much. While, having a compact design that can support the full forces applied to the system is preferred.

As calculated previously, the max load of 900N was split equally between each bearing which gives 450N. Times the highest safety margin of 3.5 gives a minimum static load rating of 1575N, which was our first requirement for choosing the right bearings.

With the static load rating established, it was just a matter of finding the right sizes now. The radial bearings being the larger of the two, were more significant when it came to the joints design. The original bearings used were the NA4903-rsr-xl needle bearing by Schaeffler. Their outer dimensions being: 30mm in outer diameter and 14mm in width. So, the ideal would be to find something with 15mm outer diameter, and 7mm width. The smallest available in the same series had a diameter of 22mm and the same width of 14mm. While searching for other option the RNAO6X13X8-TV-XL by Schaeffler were found with a diameter of 13mm and width of 8mm. While seemingly ideal, these bearings did not have an inner ring. This meant that the shaft these bearings would be mounted on would have to have an extremely high precision and be manufactured in a special way, which would be extremely costly and risky as installing the bearings would require special precision to avoid gaps and other faults.

The alternative was: NAO6X17X10-TV-IS1-XL from the same producer, 17mm in outer diameter, 10 mm in width. Which meant 0.56% of the original diameter and 0.72% of the original width. Meaning a scale down of ¼, far from what was desired, but the only thing possible. With its inner diameter of 6mm and an acceptable static load of 5500N it met our requirements (see previous chapter).

Other types of bearings, like axial-spherical roller bearings, or tapered roller bearings where considered, instead of the combination of axial and radial, however all the available bearings in those categories were generally of larger sizes.

An alternative option of using bushings hasn't yet been considered, but after short research it seems to be worth to investigate during future iterations.

For the axial bearings, smallest ones from the same series as the original ones where chosen, since they were no longer the limiting factor: axk0515-tv 15mm outer diameter and 5mm width.

5.2 Phase 3 & 4: Implementation and verification

5.2.1 **First version**

After the bearings were chosen and the general idea for how much scale down could be done the 3D modeling could start. First step was to understand the underlying geometries and structures that made up these universal joints, which differ quite a lot from the commonly available ones. While the principles of the two U-shaped holders and shafts remain the same, these joints are incorporating a larger bottom holder in which the top holder can move about as freely as possible. A basic structure with a heavy round baseplate was created with the shafts very similar to the original version and the new bearings mounted on.

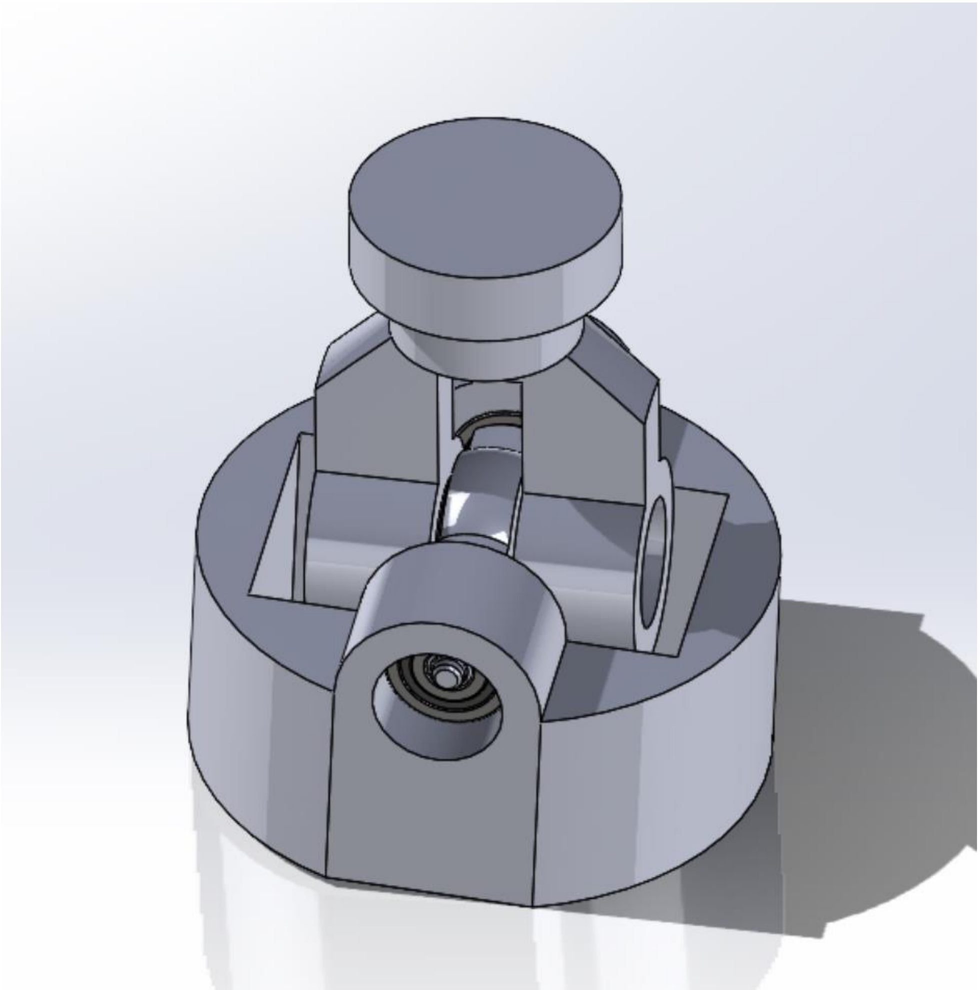

Figure 17: Very first iteration of the new joints.

Once this was done, one could start chipping away material where it wasn't needed to slim down the joints as much as possible.

5.2.2 **First analysis**

While working on optimizing and making the joint more compact, the thought occurred as to why such a heavy base part was needed. After all, what was necessary was a way to fasten the bottom holders with the bearings to the base and top plate of the Hexapod. So why not incorporate the joints into these plates directly. However, there were worries that it would be hard or impossible to fit in (or replace) some parts of the joint when its bottom holder was integrated with the plates of Hexapod.

So finally, the bottom holder was split into two symmetrical pieces.

5.2.3 **Second version**

With the new ideas in mind, a simple (placeholder) base and top plate for the Hexapod was created. A semi-triangular, flat plate, something to be further re-shaped in later stages of Hexapod project. In it slots were created to fit the bottom holders of the joints, which were now split into two separate parts and the larger cut-outs which allow for wider rotation range of the top part of the universal joint (see Figure 18).

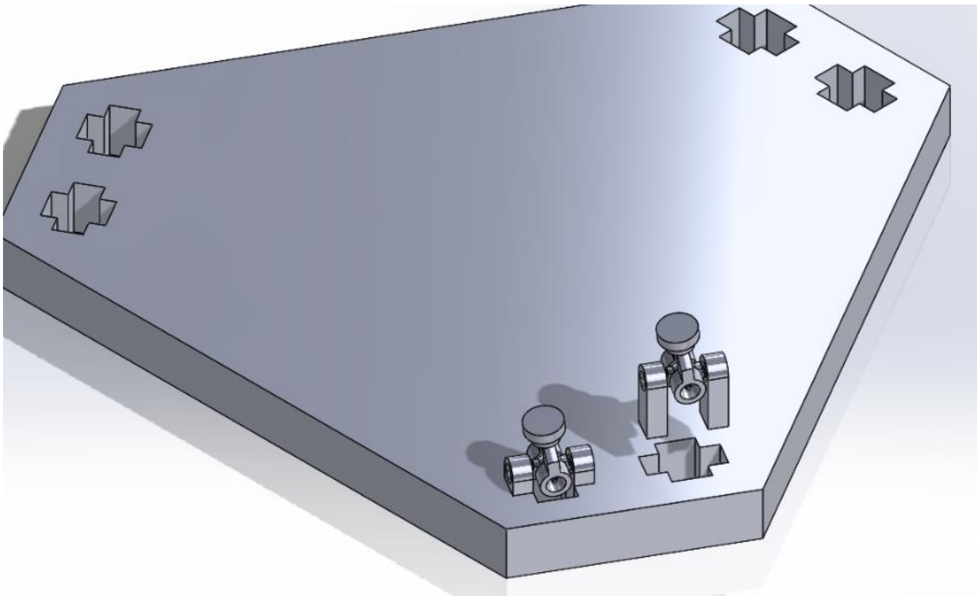

Figure 18: Basic base plate with the new joint designs.

Figure 19 shows an early model of the whole Hexapod with this joint solution.

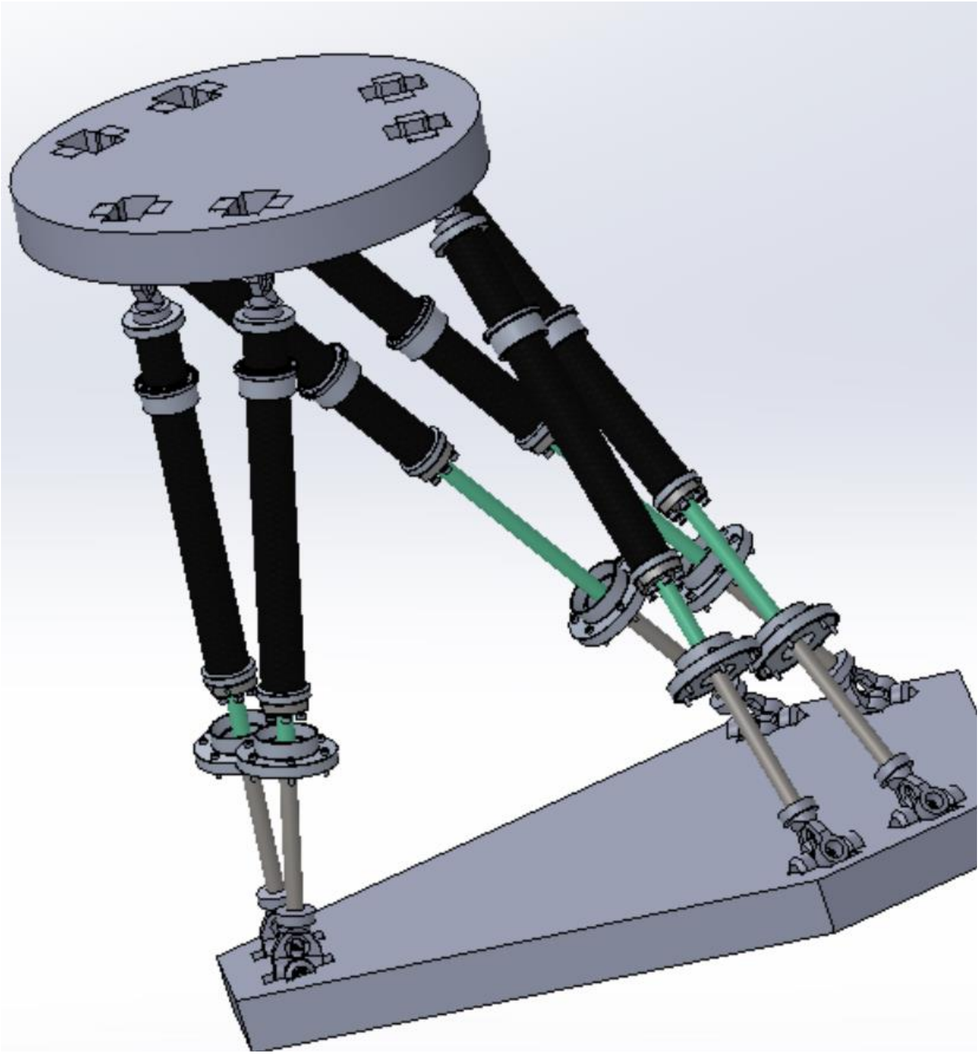

Figure 19: First look at a complete concept of the hexapod

5.2.4 **Final analysis and version**

After verification of the previous version, chamfers, rounding and further shape improvements were done to maximize movement range and minimize wight.

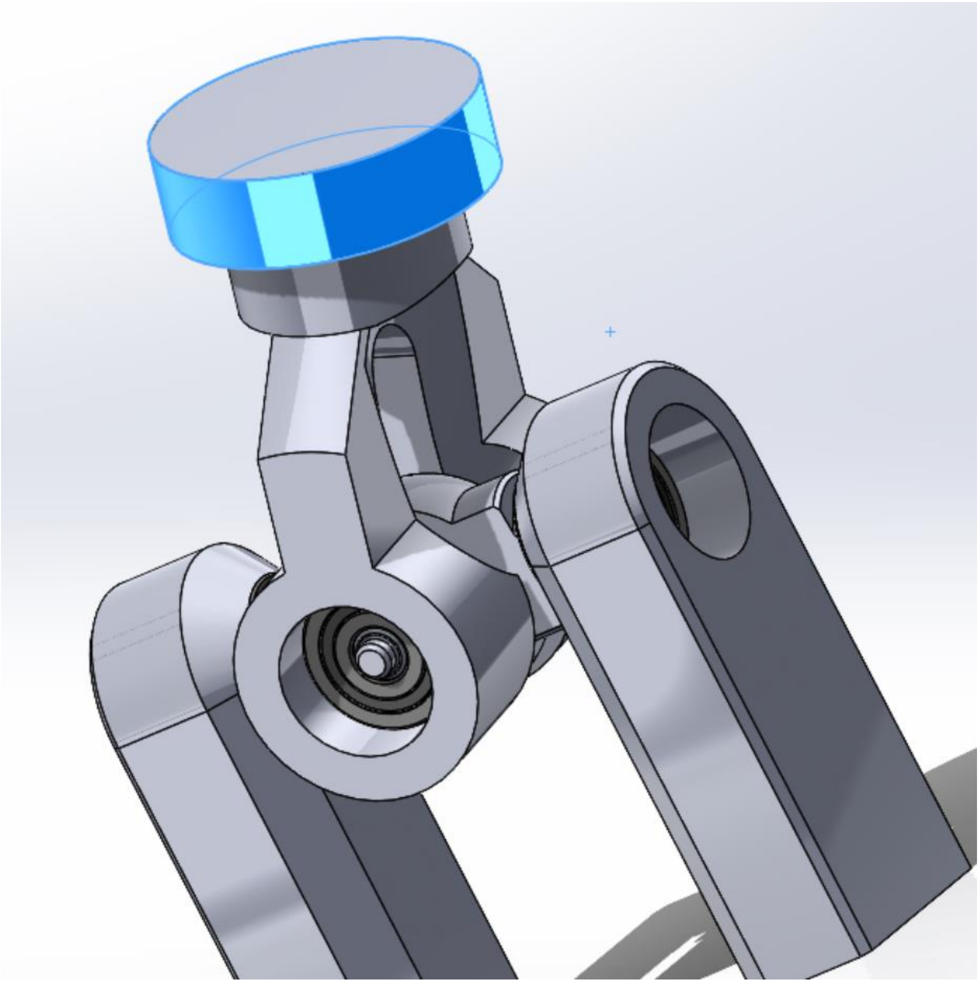

Figure 20: Attempting to optimize the joints movement by creating chamfers and pockets in its parts.

Once a satisfactory result was achieved final touches where made, cutting down on some unnecessary material, adding the second bearings sets, and creating locking rings for them. Which created the final version of the joints.

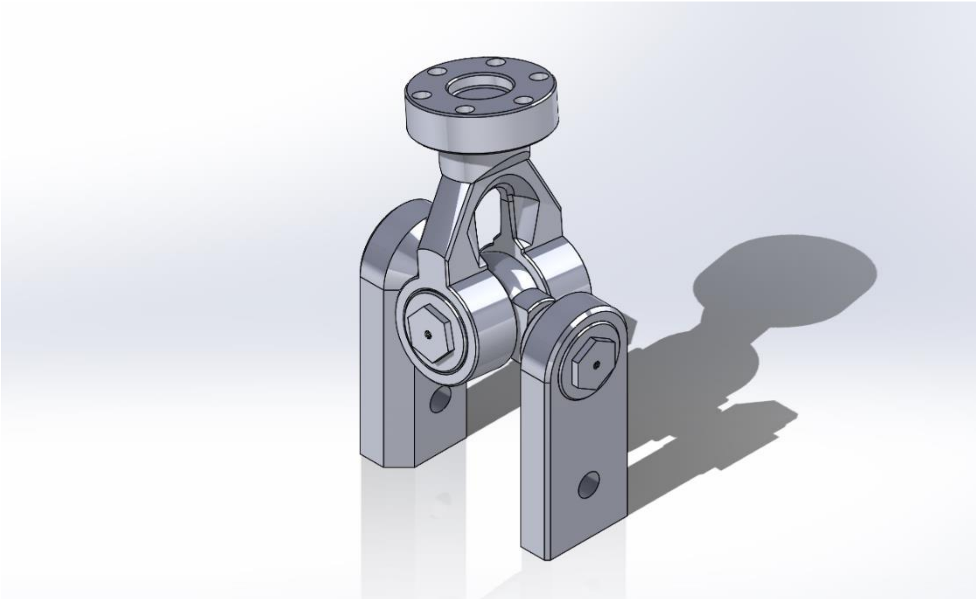

Figure 21: Final version of the universal joints for the Hexapod.

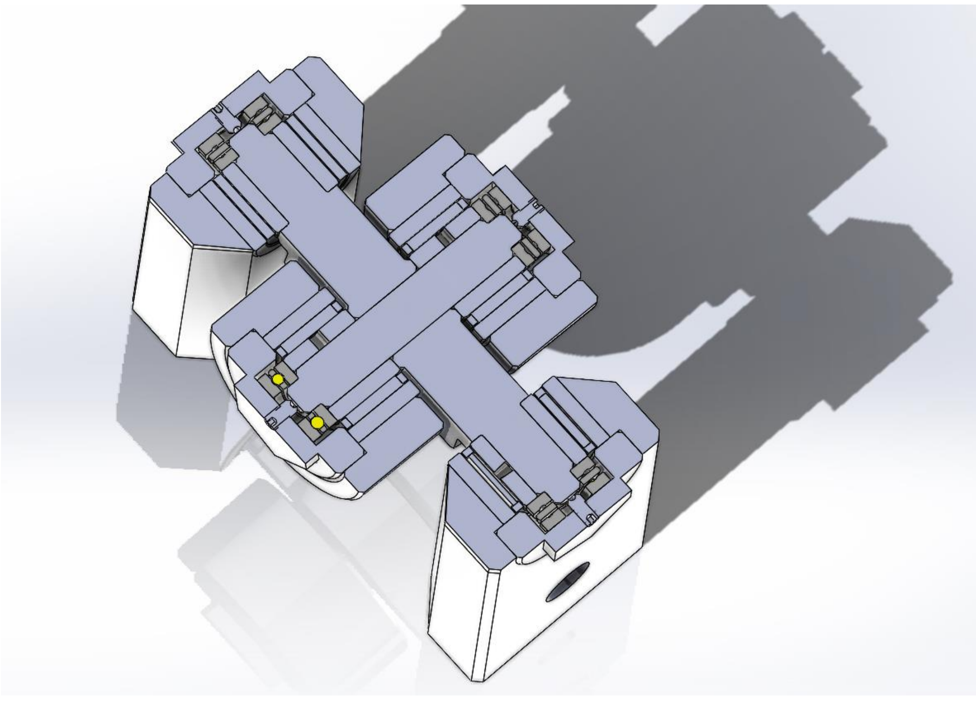

Figure 22: The insides of the joints with the various bearings and axels.

5.3 Phase 4: Review

During review phase following improvement / investigation areas were detected:

- Mounting of bottom holder parts into Hexapod plates.
- Accuracy needed to mill slots in said plates.
- Re-usability.
- Choice of materials.
- Use of bushings instead of bearings.
- Static load CAD analysis.
- Weight optimization.

The mounting holes are not defined at all. The shape of these plates will probably not be flat to achieve better movement, thus making the mounting solutions uncertain. The design of these plates was outside the scope of this thesis.

The latest design split the bottom holder into two parts. This could prove to be an issue due to misalignments caused by imprecise milling of slots in the Hexapod plates. Additional inserts or fixating solutions would have to be used to ascertain consistency, especially when carbon fiber is used for plate material.

Lack of possibility of re-use of created joints in other Cognibotics projects as a standard part. Making it available for other projects would be beneficial for the company.

The parts of the joints are designed in engineering steel, but it would be interesting to see if and what can be done in lighter materials (aluminum, carbon fiber, POM-C, etc.).

Modern technical plastic bushings might be applicable in this project. They are cheaper, and smaller in size than the needle bearings, while withstanding similar forces. If excessive movement is not necessary this could prove an interesting solution, to be further investigated (Cognibotics, 2022).

Finely, it will be required to perform static load simulation on the joints to confirm our estimated calculations. While doing so it could be beneficial to also simulate weight optimization with different materials, to see if the weight and size can be further minimized.

Due to the notices of this review, additional iterations will be expected.

6 Frameless motor

This chapter describes how the key motor components were chosen and the design process of creating the housing for them.

6.1 Phase 1: Research

6.1.1 **Motor kit**

6.1.1.1 Torque Calculations

To choose the right motor for this application the critical parameter was the torque.

By looking at how the axial, driving and friction forces are applied on the screw we can figure out the required torque.

Assuming square shaped teeth for simplicity at this stage, the following geometries can be defined.

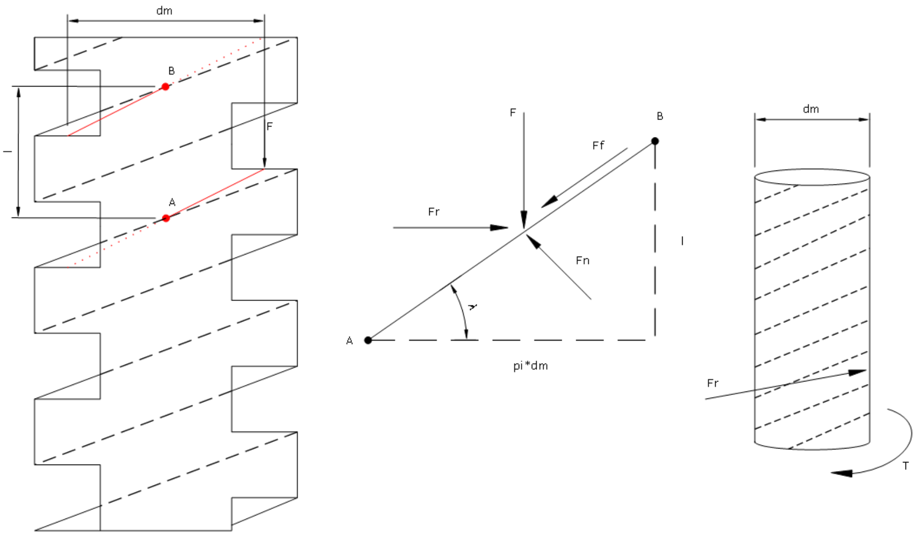

Figure 23: Showing relevant geometries for torque calculations.

A to B, is the length of the middle path of a revolution of the screw. Lambda (λ) is the lead angle of the thread, l is the lead of the screw and d_m is its mean diameter. F is the axial force on the screw, 900N from previous chapters, F_r is the force exerted by the nut on the screw, T is the torque on the screw and F_n is the normal force. F_f is the friction force and is defined by:

$$F_f = \mu F_n \tag{6.1}$$

The sum of force in the y-direction:

$$\sum F_y: -F - \mu F_n \sin\lambda + F_n \cos\lambda = 0 \tag{6.2}$$

Which gives:

$$F_n = F + \frac{\mu F_n \sin\lambda}{\cos\lambda} \tag{6.3}$$

The sum of forces in the x-direction:

$$\sum F_x: F_r - F_n \sin\lambda - \mu F_n \cos\lambda = 0 \tag{6.4}$$

Which gives, by substituting in F_n:

$$F_r = \frac{F(\sin\lambda + \mu\cos\lambda)}{\cos\lambda - \mu\sin\lambda} = \frac{F(\tan\lambda + \mu)}{1 - \tan\lambda} \tag{6.5}$$

From the figure above the following can be derived:

$$\tan\lambda = \frac{l}{\pi d_m} \tag{6.6}$$

$$T = F_r(\frac{d_m}{2}) \tag{6.7}$$

This gives:

$$T = \frac{F d_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right) \tag{6.8}$$

Finely for screws with angled teeth:

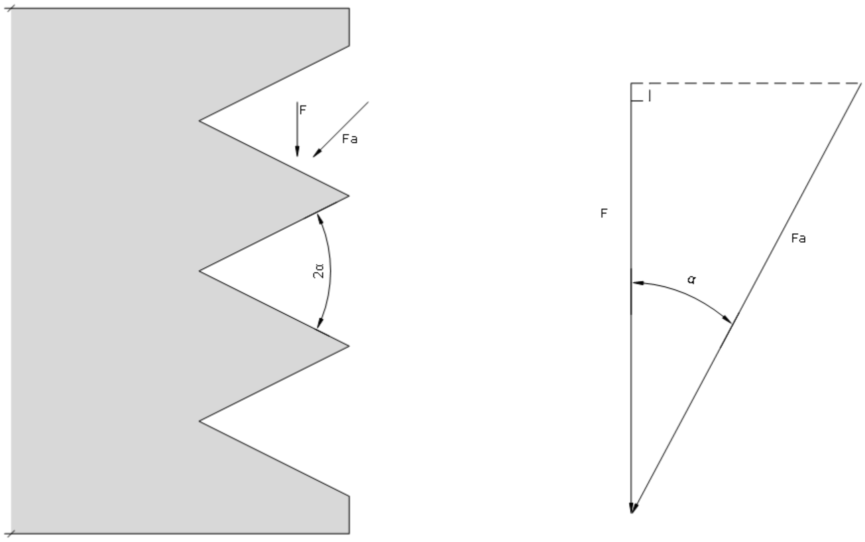

Figure 24: Axial force translation on angled teeth on ball screw.

Where:

$$F_a = \frac{F}{\cos\alpha} = F\sec\alpha \tag{6.9}$$

Substitute F_a for F in the previous equation gives the general expression for torque:

$$T = \frac{Fd_m}{2}\left(\frac{l + \pi\mu d_m \sec\alpha}{\pi d_m - \mu l \sec\alpha}\right) \tag{6.10}$$

However, the screw manufacturer of our choice (THK) does not provide the friction coefficient (μ) for the screw the general formula cannot be used.

On the other hand, they provide the following simplified formula for toque calculations:

$$T = \frac{Fl}{2\pi\eta} \tag{6.11}$$

Where η if the efficiency grade of the screw provided by the producent.

Given a lead of 5mm, applied force of 900N and an efficiency grade of 0,9 for the selected screw we require a minimum torque of 0.8Nm.

6.1.1.2 Choosing motor

There weren't that many choices for the selection of frameless motor kits. The two companies, selling said kits, that Cognibotics had previous communication within other project were Kollmorgen Automation AB and Wittenstein SE. Below is comparative data for the two smallest kits that meet the torque requirements.

Table 3: Comparison date of the two main motor kit contenders. G=pros, R=cons, Y=equal.

	Wittenstein 050-040	Kollmorgen KBM14x01
Max torque	2,66	3,6
Max Current	20	4,32-19,4*
Rated Pwer	349	700-915*
Rated Torque	1,09	1,22
Rated Current	9	1,53-6,25*
Max Temp	140	155
Mass	0,73	0,89
Outer Dia	50	74
Stator L	52,7	60
Speed	3000	12000

(*Depending on selected winding)

Since there weren't any expectations regarding the drive system, the choice was made to go with the smaller kit. Thus, choosing the Wittenstein cyber® kit 050-040. Both motor kits are made up of a stator and a rotor.

6.1.2 **Encoder**

6.1.2.1 Encoder characteristics

A rotary encoder is a device that translates angular positioning of a motor into a digital or analog signal. Encoders can be either absolute or incremental. Incremental encoders usually need a homing calibration to determine the start and angular position of the motor while absolute encoders usually do not (at least when the error isn't greater than one revolution). Generally absolute encoders are more expensive, however favorable in applications similar to ours (Murray, 2021). In the Hexapod (see assessments in next chapter), calibration with an absolute encoder would only require an accuracy of +/- 2,5mm (for a lead of 5mm of the selected screw) while an incremental encoder would need a calibration accuracy of +/- 0,025mm everytime the motor would restart.

Fundamentally the encoder can be used to determine 3 things: the motor's angular position, which direction its turning and what velocity it is moving at.

What was of interest to us in this project was the angular positioning of the screw and more specifically what it means for the accuracy of the Hexapod (not to be confused with the accuracy of the encoder). Its precision could be determined by these two key characteristics:

- Resolution of encoder.
- Accuracy of encoder.

The resolution of a digital encoder is defined as the number of measuring segments per revolution, measured as pulses per revolution (PPR) or bits (for absolute encoders). For the Hexapod to have a certain accuracy, 0,1mm as stated in the requirements, the resolution of the encoder needs to allow for a precision of at least 0,076mm when accounting for possible load displacements in joints (0,024mm). Though more likely a resolution of 0,05mm is needed to have margins.

The accuracy of an encoder is the difference between its read-out and the true values of the angular position of its shaft. This is measured in degrees or arcminutes and arcseconds. While an encoder can have a high resolution, with low accuracy there will still be "jumps" in its measurements, lowering the systems precision (Collins, 2017) (see Figure 25). Choosing a high accuracy encoder is obviously beneficial, however can be costly and it will not solve the accuracy of the entire Hexapod if there are errors elsewhere. However, choosing an inaccurate encoder might create unwanted issues.

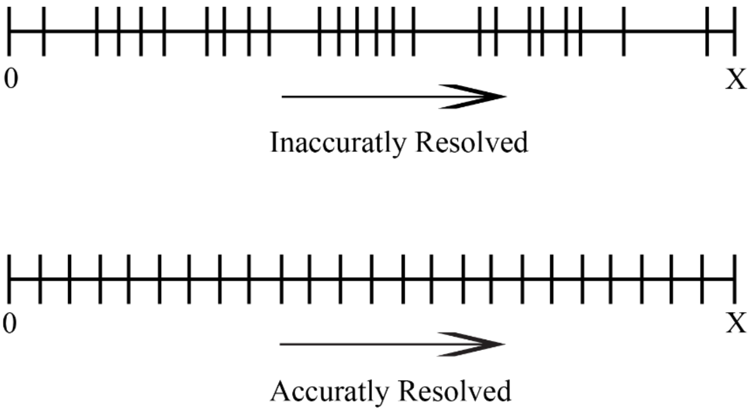

Figure 25: Difference between an inaccurate encoder measurement and an accurate measurement with the same resolution.

6.1.2.2 Determining the required encoder resolution and accuracy.

By dividing the required positioning of the leg (0,05mm – see comments in previous chapter) by the displacement during one revolution (selected 5mm lead) you can determine the angular precision to be 1/100th of a revolution. Using binary encoders, the required number of bits would be at least 7 which gives 1/128 ($1/2^7$) which equals to around 2.8 degrees. The accuracy of the encoder must be at least 1/100th of the resolution thus at least 3,6 degrees, which translates to 216’’ (arcmin) according to this formula:

$$Degrees = \frac{1}{60} \times arcmin \tag{6.12}$$

Additional bits may be required to count what revolution the shaft is on. For our stroke of 310mm with a lead of 5mm this would mean 62 revolutions, which gives 6 additional bits.

6.1.2.3 Selecting encoder

There are many encoders available on the market. Heidenhain Corp. was chosen as the provider due to contributing in previous Cognibotics AB projects.

The smallest and cheapest encoder that meets our requirements was chosen at this stage.

Table 4: Most suited encoders suggested by Heidenhain Corp (green = selected).

Hedeinhain	EQI 1131	IdNr 1164813-02 Multi turn
Hedeinhain	ECI 1118	Single turn
Hedeinhain	ROC 1013	IdNr 606691-01 Single turn
Hedeinhain	ROC 1023	IdNr 606693-01 Multi turn
Hedeinhain	ROQ 1025	IdNr 606694-01 Single turn
Hedeinhain	ROQ 1035	IdNr 606696-01 Multi turn
Hedeinhain	KCI 1319	Single turn
Hedeinhain	KBI 1335	Multi turn

6.1.3 **Brakes**

6.1.3.1 Brake types

Electric motor brakes which are in the category of electromechanical brakes, can generally be defined by two types: spring set and electrically set (Inc, 2020).

The spring type works in the opposite way, a set of springs mechanically actuates the braking plates together which are then released when power is applied to an electromagnet. Thus, it brakes as soon as power is cut off from the system making it useful for emergency stops or holding things in place when the motor is not running. It doesn’t offer much control over the stopping speed.

The spring setting brake can be further split into DC brakes, 3 phase brakes and single-phase AC solenoid brakes. They have springs pushing an armature (friction plate) against the brake-hub creating torque. The armature is then released through magnetic force by powering the system. Instead of springs, permanent magnets can also be used.

The electrically set type brakes when power is applied to a magnetic coil of some sort, which presses the braking plates together generating a torque. These types of brakes can have a varying torque making them applicable where softer breaking and releasing is necessary.

There are also tooth brakes which use teeth instead of friction plates to create the stopping torque, however due to their high backlash they are really irrelevant for this application (Sapec-Inc.).

6.1.3.2 Selecting brakes

When selecting brakes, the key features to consider are the allowable torque and the RPM of the motor. For our application it was also important to consider the backlash, which is the amount of movement the brake allows after it has been initiated (play due to inertia). Generally, all spring set brakes have some backlash. From 1 degree for larger brakes to 0,25 degrees for smaller brakes, which in our case translates to an inaccuracy of 0,016mm, 0,0042mm respectively. For backlash to be further minimized permanent-magnet friction brakes are recommended, however, costly and limited in size (Sapec-Inc.).

This project also attempts to save on weight and size so choosing the smallest brake was favorable. Other criteria like the expected life cycle, available power supplies, serviceability and mounting constraints should be considered too. Finely, the environmental parameter, like noise, temperature etc. should be considered.

Sepac Inc was the company which has previously cooperated with Cognibotics AB, thus selecting their brakes for the Hexapod was advantageous. However, many other types of brakes are available on the market which can be further investigated in the future.

Table 5: Considered brake choices (green = selected)

Sepac	Permanent magnet	Smallest backlash
Sepac	Ultra-thin	Smallest size

6.1.4 **Clamping Hub**

A clamping hub was intended for attaching the screw shaft to the motor shaft. These hubs are standard components produced, among others, by ETP Transmission AB. By tightening a screw, the hub expands radially through hydraulic pressure, clamping the outer and inner component together. This allows for quick assembly/disassembly of components without any permanent damage. It is essentially like heat fitting but adjustable (Cognibotics, 2022).

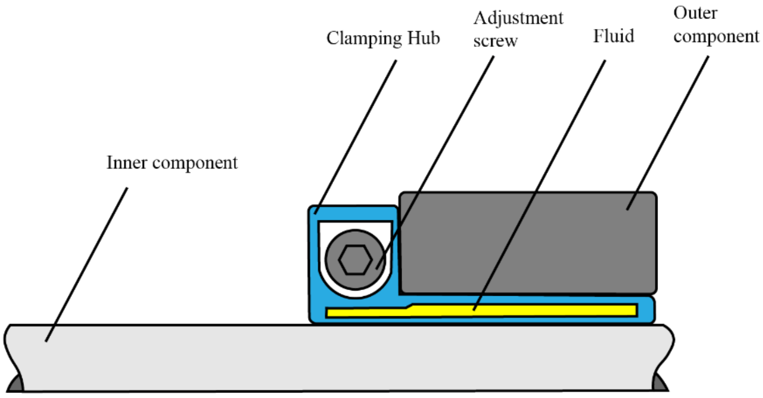

Figure 26: Clamping hub in between a shaft and an outer component. Picture adapted from ETP Transmission AB.

There are several types of clamping hubs provided by ETP Transmission AB, some featuring holes for steering pins, used for angular orientation of the hub.

Table 6: Considered hub selection (green = selected).

ETP	ETP-POWER-15	Centring pins
ETP	ETP-TECHNO-15	
ETP	ETP-EXPRESS-15	Smallest

6.1.5 **Bearings**

According to SKF guide on roller bearings and seals in electric motors there are many designs parameters to be considered when choosing the right type of bearings for the application (SKF, 2013):

- Mechanical parameters
    - Arrangements of bearings
    - Dimensions
    - Materials (even surrounding)
    - Motor coupling
    - Mounting orientation
- Load
- Speed
- Environment parameters
    - Vibrations
    - Noise
    - Temperature
    - Sealing
- Maintenance
    - Bearing life
    - Lubrication type
    - Monitoring of wear and tear

As stated by Todd A. Hatfield, a failure analyst at Heco Inc, a company specialized in electric motor maintenance; the most common motor failure (51%) are caused by faulty bearings (Hatfield, 2018). Choosing the right bearings will be critical for the final Hexapod to reach the market as a quality product. However, this design is on a conceptual level and going through all the above stated parameters would be a thesis in itself, so only the first two points will be considered at this stage.

6.1.5.1 Mechanical parameters

Generally, to support a rotating shaft two bearings are required at a certain distance from each other. These can be arranged in different ways depending on the application:

- Locating/non-locating bearing arrangement
- Adjusted bearing arrangement
- Floating bearing arrangement

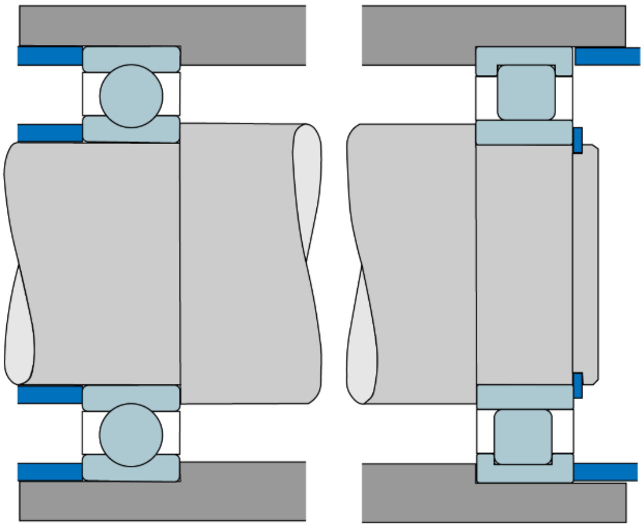

Figure 27: Locating (light blue left side) / non-locating (light blue right side) bearing arrangement. Right side shows a NU cylindrical roller bearing. Adapted from Schaeffler.

Locating/non-locating bearings arrangements are made up of two bearings apart from one another. As a result of material movement due to temperature, motor vibration or manufacturing mistakes, one of the bearings mustn't lock the position of the shaft. The locating is dimensioned and mounted in a way to support both radial and axial forces on the shaft. The non-locating bearing is mounted with a cylindrical roller bearing with a N or NU caged solution, which allows axial movements and doesn't carry any axial loads (Figure 27).

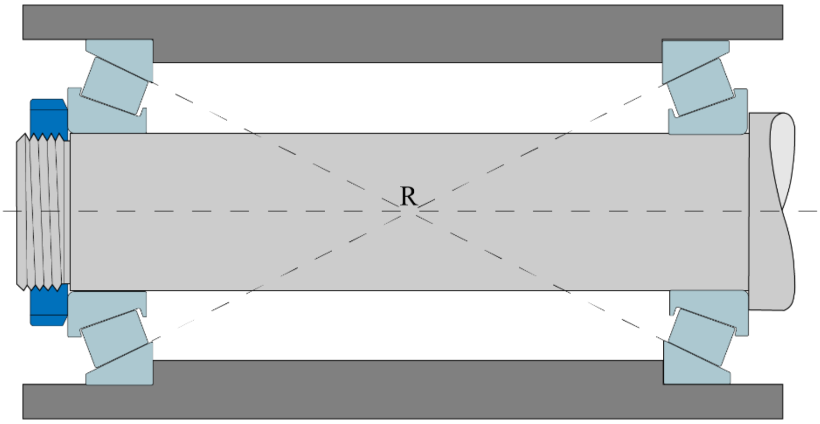

Figure 28: Adjusted bearing arrangement using tapered roller bearings (light blue). R = roller cone apex. Adapted from Schaeffler.

Adjusted bearing arrangement is made up of two angular contact bearings (either angular ball bearings or tapered roller bearings), mirrored towards each other. By adjusting the distance between the two bearings (this can be done with a lockring screw on the shaft for example) and depending on their arranged angles you can derive the apexes of the bearings. In simpler terms, the distance and angular position of the bearings decides the compensation of thermal tensions, while fixating the shaft. Depending on the distance of those apexes you can get different results. For example, should the apexes coincide then the radial and axial thermal expansions will cancel each other out (Shaeffler, 2022).

The final arrangement described in Schaeffler bearing arrangements guide, is the floating bearing arrangement. This arrangement is simply defined by a clearance gap on one side of the bearing set to allow for thermal expansion and other movement.

The simplest for this project is the locating/non-locating or the floating arrangements. The adjustable arrangement adds an unnecessary level of complexity to the system. The first one is recommended by Schaeffler bearing arrangement guide, for electric motors of medium power rating (Shaeffler, 2022).

The "Design of bearing arrangement" by Shaeffler (Shaeffler, 2022) shows an example of how this arrangement could look, which will be considered at the implementation stage.

6.1.5.2 Loads

For the non-locating bearing, NU cylindrical roller bearing, only radial loads apply which gives:

$$P_0 = F_r \tag{6.13}$$

Where P_0 is the equivalent static bearing load and F_r is the radial load exerted on the bearing by the torque of the motor. Given a high safety factor 3,5 the basic static load rating (C_0) is given by:

$$C_0 = 3.5 \times P_0 \tag{6.14}$$

The basic dynamic load taking (C) can be calculated in similar ways by simply replacing C_o and P_0 with C respectively P (equivalent dynamic bearing load). Though, these can be omitted due to a mostly static application.

For the locating, deep ball grove bearing, both radial and axial loads apply. The static load rating can be given calculating the equivalent static bearing load (P_0). Given by:

$$P_0 = 0{,}6F_r + 0{,}5F_a \tag{6.15}$$

However, in our system there are no radial loads, as they are countered by the axial loads of other legs of the hexapod. Thus, assuming the previously calculated axial loads of 900N times 0,5 times a high safety factor of 3,5, gives a required minimum static load rating of 1,575kN for the locating bearing and 0kN for the non-locating bearing.

6.1.5.3 Selecting bearings

Table 7: Bearing considerations for the design (green = selected)

Schaeffler	6002-C-2HRS	Deep groove ball 15mm
SKF	NU 202 ECP	NU 15mm
SKF	61905-2RS1	Deep ball groove 25mm
SKF	NU 205 ECML	NU 25 mm (second v)
SKF	6001-2RSH	Deep ball groove d12 mm b 9
SKF	6004-2RSH	Deep ball groove d20 mm b 12
SKF	6003-2RSH	Deep ball groove d17 mm b10
SKF	61804-2RS1	Deep ball groove d20mm b7 (thinnest) (second v)

SKF	61805-2RS1	Deep ball groove d25 b7
SKF	NU 204 ECP	NU d20 b 14
SKF	NU 202 ECP	NU d15 b11

6.1.6 General layout

Kollmorgen Automation AB provides interesting recommendation and examples of how a general frameless motor could be designed (Hurlay Gill, 2017) (Hurley Gill, 2017). These instructions need to obviously be reevaluated with the required tolerances, air gaps and other parameters of the Wittenstein motor kit. However, the figures provided in these documents served as references for the general implementation plan discussed in the next chapter.

6.2 Phase 2 &3: Implementation and verification

6.2.1 **General implementation plan**

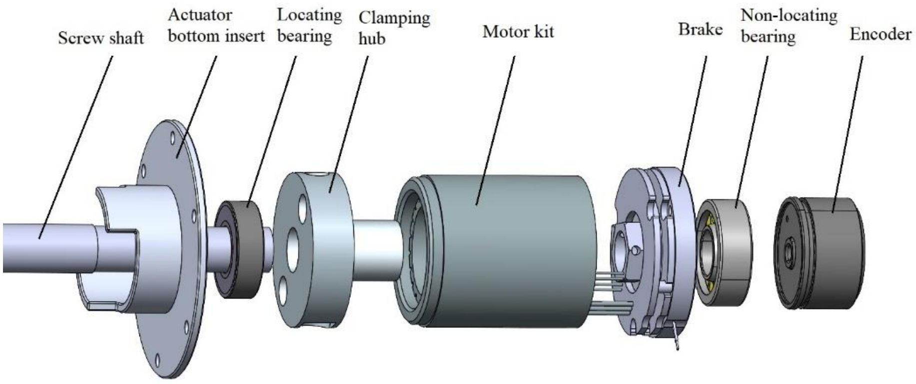

Figure 29: General layout plan for necessary components of the motor.

The following was considered in the implementation plan:

- Building shaft assembly.
- Creating housing assembly.
- Manufacturability, including producer's assembly requirements
- Serviceability

The first step was to create a motor shaft, according to a general overview of how the motor should be built up, based on previously done research. This shaft should be attached to the rotor through adhesives and would need to accommodate two bearings, an encoder and a brake while somehow connecting to the ball screw shaft. By using a clamping hub, the screw/motor could be disconnected from each other for maintenance purpose.

The next step was to design a motor housing, connected to the actuator's outer cover, that would hold the stator and the two bearings in place, as well as other non-rotating components (brake and encoder, depending on specifications).

The housing and shaft needed to be designed in such a way that assembly and disassembly was as easy as possible. Furthermore, all parts had to meet the assembly requirements made by their respective manufacturer.

Primarily bearings and brakes, but also the rotor/stator and connection hub had to be accessible for maintenance. For example: lubrication, replacements or adjustments.

6.2.2 **First version**

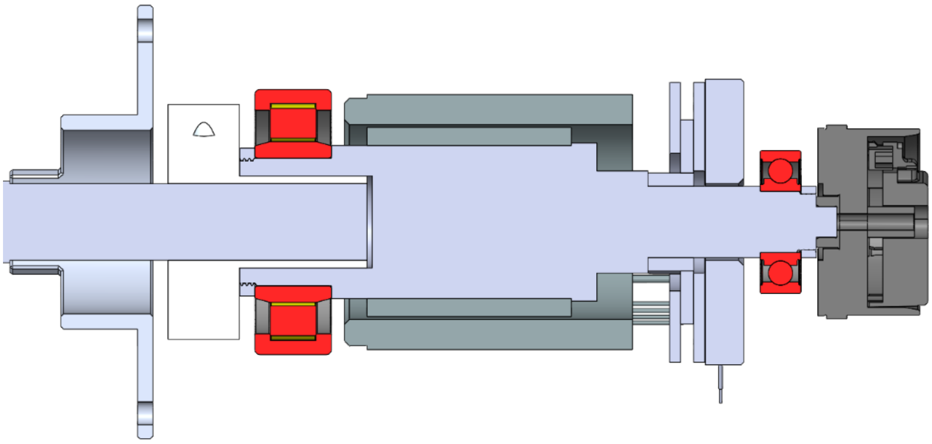

Figure 30: Cross section of first version of motor shaft with all components.

A motor shaft was created starting from the rotor. Following this, all necessary components were mounted in such a way, that they could be taken apart one by one. Components were spaced to allow fixation to the housing and shaft, while retaining their removability and to allow access to mounting holes in the brake and encoder.

Changes from initial plan (see Figure 29):

- Smaller hub due to not being able to create interface for centering pins (see table in chapter 6.1.4).
    - Centering would have to happen through housing.
- Moved left bearing closer to motor.
- Switched bearing places, due to size limitations of NU (no smaller sizes than 15mm inner diameter). Shaft should progressively become smaller for installation purposes, which wasn’t possible due to brake inner diameter.
    - Smaller locating bearing still within required parameters.

Proposed assembly procedure:

1. Insert shaft from right with adhesive into rotor.
2. From left (see Figure 30):
    a. Place NU bearing and locking ring (fastening of bearing from housing needed)
    b. Insert clamping hub for screw shaft.

3. From right:
    a. Slide on brake (access to tightening screw and cable hole needed in housing)
    b. Slide on bearing and locking ring
    c. Finally place encoder in centering hole and screw it on through middle hole.

From this iteration, a general design idea for a 3-part housing was developed:

- Main part holding stator (through adhesives).
- Top part attached to actuator and main part to allow access of bearing and clamping hub.
- Bottom part attached to main part and bottom joint allowing access to brake, bearing and encoder.

6.2.3 Second version

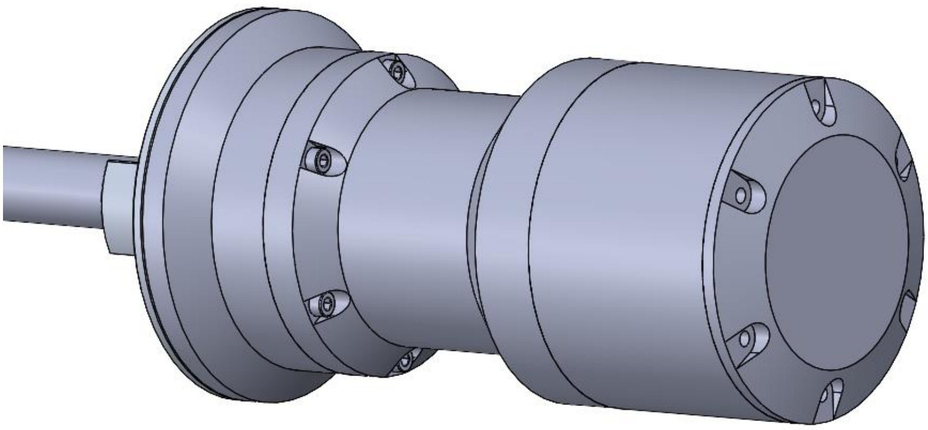

Figure 31: Outside look of the second motor version with housing.

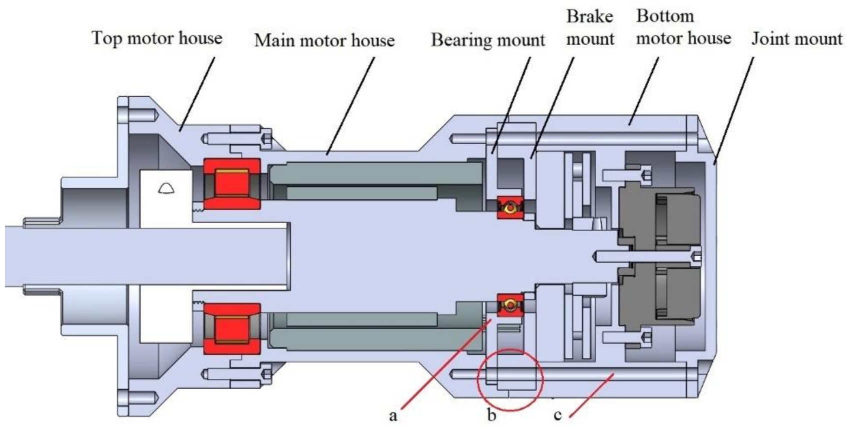

Figure 32: Cross section of second version of motor with housing. Weak design points are marked a, b and c.

In this version a housing for the motor was created and some corrections to the shaft were done. According to previously stated plan, the housing should be made of 3 main parts (top, main and bottom housing). This would allow for an easy assembly/disassembly and serviceability of components. The bearing and brake mount were further added to lock the bearing in place and mount the brake onto the house. The last piece (join mount) was basically just a cover to access the encoder, it would also be where the joint should be mounted too.

However, several issues with the second version where recognized (which would be addressed in the following version):

- Shape complexity of housings increased due to necessary mounting of bearing (b in Figure 32).
- Bottom bearing mount was too fragile (a).
- Bottom motor house screw too long (c).

Due to having to mount the bottom bearing axially and radially, while still being able to assembly/disassembly and service it, a mounting flange had to be added. This was not optimal, it increased the outer diameter of the house substantially, due to the need for further centering phases (b in Figure 32). Furthermore, the bearing and brake mounts were only 3mm thick. If these were made from aluminum, it could be a serious rigidity issue, as these components should carry substantial axial loads.

Lastly the design required a 65mm screw to connect the joint mount through all element of the housing. Standard M3 screws available for DIN-912, usually go up to 40mm (compared to Eugen Wiberger AB online stock - company selling standard components). To get over 65 mm an m8 from this series would be needed, which would substantially increase the size and weight of the motor. Other alternative would be a specially manufactured screw or a change in the motor design.

6.2.4 **Third version**

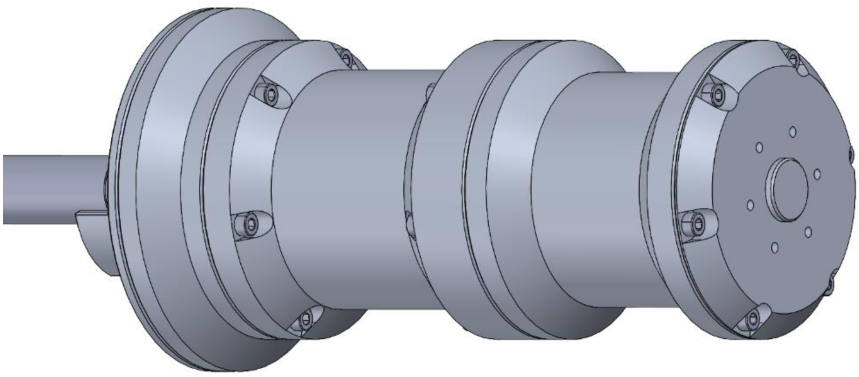

Figure 33: Outside look of the third motor version with housing.

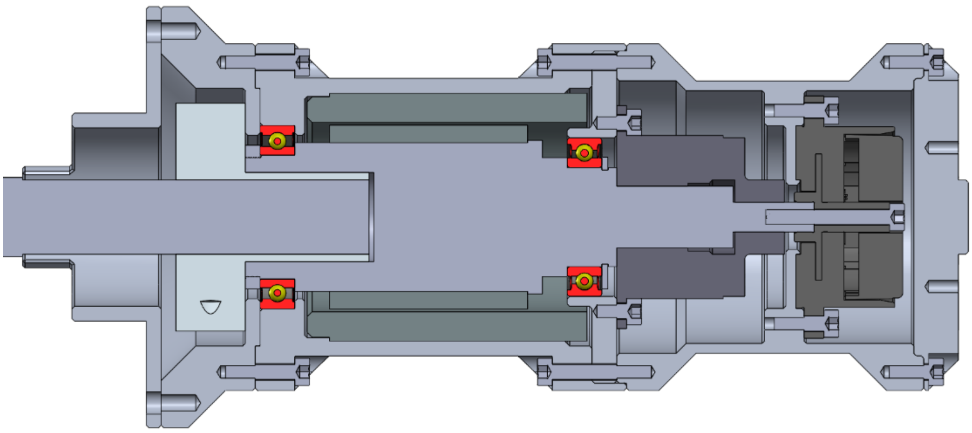

Figure 34: Cross section of third version of motor with housing.

Changes made to previous version, based on previously stated improvements and new component research:

- Moved locating bearing to top side.
- Non-locating bearing changed to deep groove ball bearing with no axial fixating.

- Spring brakes changed to a permanent magnet brake.
- Removed bot bearing mount.
- Snap ring used instead of locking nut at the bottom (no space at the top).
- Housing remade to fit new parts and arrangement.

In this version the locating bearing, which carries the axially loads was moved to the top side as per previous observations. The non-locating bearing was changed from an NU to a ball bearing that isn't axially fixated. This is to simplify the housing design by getting rid of the weak bot bearing mount which increased the complexity and size of the motor.

The previously used ultra-thin spring break was replaced with a permanent magnet friction brake to minimize backlash. This brake is ticker but has a smaller diameter, which allowed the housing to slim down in the bottom section, solving the issues with the long screw as stated in the previous chapter.

A snap ring was introduced as a locking mechanism for the bot bearing, due to being a standardized component available on the market. However, fitting another one on the top bearing proved difficult.

6.3 Phase 4: Review

The following points of interests/improvements for the motor were recognized during this project phase:

- Consider using screws with smaller lead.
- Consider using external position sensors (giro and optical) instead of built-in encoders.
- Material considerations.
- Weight optimization of inner shaft and housing.
- Standardized locking solutions for bearings.
- Cable management solutions
- Investigate implications of bearings position and arrangements in respect to other components.

According to previously done calculations of encoders, using smaller leads would mean higher accuracy for lower resolutions. Thus, it could be possible to use a hall sensor (encoder) provided by Wittenstein (as an addition to the motor kit), which has a lower resolution, if a smaller screw lead would be used. It could also be possible to achieve higher resolution with this encoder by compensating with software. Using the hall sensor provided by the motor producer would be cheaper and more compact, thus worth investigating further.

External positioning sensors. Using them would perhaps require more advanced software, but it could be both a cheaper and more accurate positioning system, as encoders do not give data and allow compensation for other inaccuracies than motor position. Type of sensors to be considered here is out of the scope for this project iteration and requires research in co-operation with measurement and software engineers.

Material was important to consider and would need more analysis. On one hand aluminum is a lighter solution, greatly benefiting this project, however it has double the expansion rate of other steels. This can prove to be problematic for the fitting of the bearings and motor kit if temperatures rise a lot. The motor shaft would most likely have to be made of steel for this reason and to properly carry the axial loads. It would be interesting to see how other materials, like carbon fiber or techno plastics, could be implanted in the motor housing design (Cognibotics, 2022).

Note that high temperature differences would probably not occur during normal industrial work conditions for the Hexapod and motor will not generate its own heat as it is not running continuously.

In the final design there is a lot of empty space and thick blocks that in future iterations could be optimized to reduce size and weight of the system. An alternative to the motor kit was a rotor with much smaller inner diameter, where the rotor was a substantially ticker component, however, to reduce weight while keeping its rigidity, Wittenstein had implemented a specific hole pattern. It would be of interesting to see how this can be implemented into our design of the shaft.

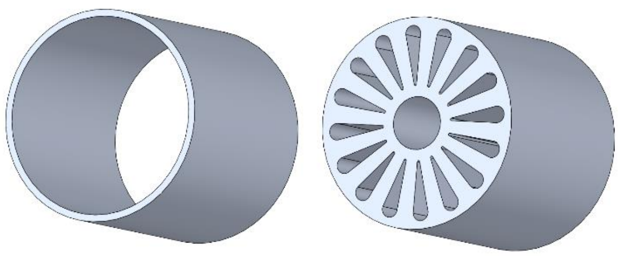

Figure 35: Comparison of small and big shaft diameter based on the frameless cyber® kit by Wittenstein SE.

In the third version, a standardized snap ring was introduced instead of a manufactured lock ring for the bottom bearing. Due to space limitation this was not possible for the top bearing. Looking into adapting the design or finding other standardized locking mechanisms would be beneficial for simplicity and cost effectiveness.

The main missing component of the latest motor design is cable management. Designing a way for the cables from the motor kit, brake and encoder to exit the housing without interfering with any rotating parts, is required. Furthermore, it would be beneficial to create a mock-up/prototype, perhaps through 3D printing, to see how the cables would behave during assembly/disassembly, as cable behavior simulation is not optimal in 3D software like SolidWorks.

Finally, it’s worth testing and discussing how various positions of the bearing affect their performance. What side the locating/non-locating bearing should be on and how the distance from the motor kit, screw shaft or other parts affects them.

7 Result

This chapter describes the resulting concept of this thesis.

7.1 The resulting concept

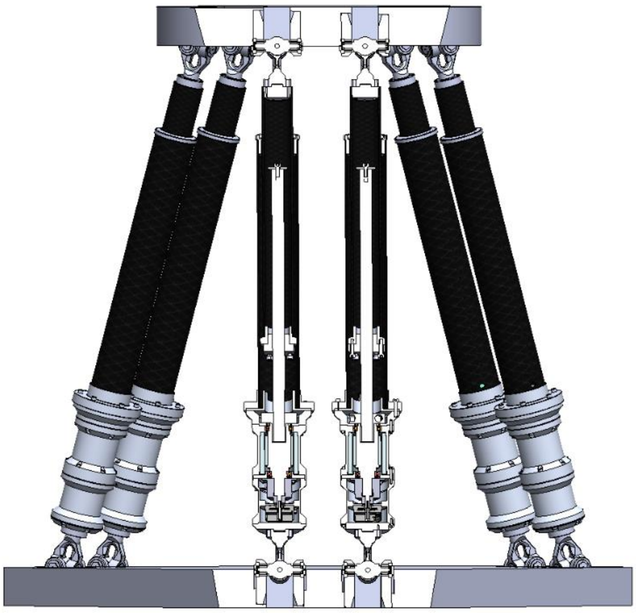

Figure 36: Latest Hexapod concept design.

Stewart platform - the Hexapod, developed for Cognibotics AB is made up of six legs set up between two plates. The main objective of this thesis was to construct a conceptual model of these legs, based on available components on the market and previously defined requirements.

In this concept, the legs (which are all the same), are made of an actuator with a frameless motor inline, to which two universal joints are connected, attaching the legs to the base and top plate.

The actuator is made up of a precision ball screw manufactured by THK. It's covered by two carbon tubes, one which simply works to seal the system from dust and debris, while the other, attached directly to the nut of the ball screw, drives the top joint (which pushes the top plate). Between these components are aluminum interfaces design to mount the various parts together. There are also stop mechanisms and sealing housing connected to these interfaces.

The joints (top and base, both the same), are based on previously designed universal joints made by Cognibotics AB for other projects. They are redesigned to become as small and light as possible within the limitations of standardized bearing components and the accuracy and load requirements.

The frameless motor is developed around the motor kit provided by Wittenstein SE. The kit is made up of a stator, attached to an aluminum housing, and a rotor, attached to a steel shaft. These are the two interfaces (housing and shaft) which are also designed to fit a permanent magnet brake, absolute encoder and a set of deep ball groove bearings. The motor is attached to the bottom of the actuator and to its screw shaft through a clamping hub and is the driving power behind the Hexapods movement. It is then attached to the base plate through the universal joint.

By actuating these six legs asymmetrically the top plate moves around with 6 degrees of freedom.

7.2 Selected standard components

Actuator:

- Ball screw: SDA-V 1605V with a 310mm stroke by THK.
- Seal: Polyurethane Rubber Rod Sealing Wiper (available by many manufacturers)
- Carbon Fiber tubes (available at various sizes) by Rock West Composites

Joints:

- Radial bearings: NAO6X17X10-TV-IS1-XL by Schaeffler
- Axial bearings: axk0515-tv by Schaeffler

Motor:

- Frameless motor kit: cyber® kit 050-040 by Wittenstein.
- Brake: Permanent Magnet Friction brake (1Nm torque) by Sepac USA.
- Encoder: EQI 1131 (IdNr: 1164813-02) by Hedeinhain.
- Locating/Non-locating bearing: 61805-2RS1, 61804-2RS1 by SKF.
- Clamping hub: ETP-EXPRESS-15 by ETP.

Screws and snap rings by Eugen Wiberger AB.

7.3 Resulting parameters

Actuator:

Table 8: Mass and geometries of Actuator.

Mass (kg)	Hight min (mm)	Hight max (mm)	Outer Dia (mm)
1,4	388	698	91

Note: Outer diameter is for the widest part (interface mount to motor), the tubes are of smaller diameter.

Joint:

Table 9: Mass, freedom of movement (angle) and geometries of joint.

Mass (kg)	Hight (mm)	Width (mm)	Length (mm)	Volume (mm3)	Angles (+/- deg)
0,3	105	67	43	3.0 x 105	50

Note: Angles is in both directions (for both shafts)

Motor:

Table 10: Mass and geometry of Frameless motor.

Mass (kg)	Hight (mm)	Outer Dia (mm)	Volume (mm^3)
2,1	161	70	$6,2 \times 10^5$

One leg:

Table 11: Mass and geometries of one Hexapod Leg.

Mass (kg)	Hight min (mm)	Hight max (mm)	Outer Dia (mm)
4,1	654	964	91

Hexapod:

Table 12: Mass and geometries of entire Hexapod, excluding mass of placeholder plates.

Mass (kg)	Hight min (mm)	Hight max (mm)	Width (mm)	Length (mm)
24,6	734	1050	820	710

Note: Mass here does not account the mass of base and top plate. Which could be an increase of about 70kg if made of solid aluminum as it is currently. Dimensions are accounting the base plates with a thickness of 50mm. These are just placeholder values and have not been evaluated at all.

8 Discussion & Conclusion

This chapters discusses what design considerations are left to be investigated, and what future planes can be made for the Hexapod.

8.1 Discussion

8.1.1 **Accuracy**

High accuracy is one of the most important parameters of the Hexapod. Many assumptions and general calculations were done to estimate the effects various designs would have on the accuracy of the system. It is worth noting that, these calculations were made for one leg, and the accuracy requirement of +-0.1mm for the whole Hexapod, was assumed to translate to 0.1mm accuracy for each leg. In reality, it might not be true, as some errors can be additive. Furthermore, there were many parts left out of the analysis, the rigidity of the carbon fiber tubes and all the interfaces in between, rigidity of the motor, joints (other than the shafts) and the base and top plate. Thus, further analysis, calculations, simulations and testing needs to be performed to establish what the actual accuracy of the Hexapod is.

Cognibotics AB specializes in compensating for inaccuracies in robots. Inaccuracies caused by elasto-kinematics (movement of points on the robot due to elasticity of the parts), can be compensated using their software, however certain inaccuracies, such as the backlash of brakes cannot be compensated. Identifying these faults and minimizing them is crucial for further improvements of the hardware design of the Hexapod.

8.1.2 **Motor**

An interesting discussion is whether the frameless motor is a good concept in general. Before this development, it was argued with other co-workers and the consensus was that it is not worth it. A frameless motor generally means a lot higher costs, especially in low volume production, it was estimated to be a 50-100% cost increase. It is more work, and it entails a lot higher risks. Designing a frameless motor without any prior experiences in motor development can often lead to issues with bearing fitting, temperature management, expansion of materials and general wear and tear. Furthermore, there are issues with how it translates to the assembly of the motor, which would have to be performed properly not only by the designer, but by people on the production line. Lastly the motor would have to go through a lot of testing to get proper UL (ETL) and CE certificates which are required for it to be sold on the market. On the other hand, buying a widely available motor from known manufacturers, puts the responsibility of its design on them. Thus they, who specialize in the subject, must ensure these aspects are met.

So, when do you choose a frameless motor?

Generally, this is chosen for very specific scenarios which cannot be achieved by the commercially available motors. For example:

- Very constrained weight and volume requirements.
- A lot of moving components which do hinder the use of belts or cog drive.
- Difficulties regulating the temperature.
- Very specific hardware requirements (brakes/bearings/encoder/sensors etc.).
- A combination of the above

While the Hexapod might have weight and volume constraints it is difficult to tell at this point whether a commercial motor will not be good enough. Through simulations and testing you can establish where the fault and issues of the motor lie. By determining those, you can then investigate if other available motors would solve these problems or if a specifically customized frameless motor is necessary. Without any tangible test results that highlight certain issues, it is not worth the costs and risks of a frameless motor design.

However, the reason for this concept to have been developed at this early stage of the project, is due to it just being that, a concept. It is of interest to the company (Cognibotics AB), to investigate what a frameless motor could look like and what necessary parts would be required. What their weight, size and rigidity would be. So that if the decision is made to create one, a baseline concept would already be made, upon which it can be further developed to meet any new requirements.

8.1.3 **Force regulated sensors**

Sensors for force regulation is a necessary step for future developments of this project. Due to time constraints and uncertainties at the company it was not implemented in this version of the concept. There were prospects to mount the sensors on the top plate (which is not within the scope of the project) or mounting them on the actuator. Final decision on this matter has not been made yet and will have to be further reviewed.

8.1.4 Simulations

As a necessary next step in this project, the CAD static simulations must be performed, to see what stress is present on each involved component. This will create a base for further improvements, like shapes (cut-outs, filets, thickness, and sizes to minimize weight and relieve stress concentrations) and choice of materials and their grades (for example aluminum or steel grade).

Simulations should also be performed to explore the Hexapods movement capabilities. To establish a workspace and investigate the clearances between the legs so that they do not collide and have safety margins. This would have to be performed along with the design of the top and base plate.

8.1.5 Materials

As stated above the choice of materials will depend on component stress, environmental parameters, rigidity, weight, and aesthetics. As for now only use of aluminum, steel and carbon fiber tubes was considered, but other techno-plastics, for example POM-C (Delrin) or similar, are of interests too. Use of plastic bushings instead of bearings in joints or motor could be considered as the most advanced available types have similar parameters to the chosen bearings while smaller in size.

8.1.6 Sustainability

Sustainability is a very important design criteria which has not been completely addressed in this report. For example, the current choice of material has not been evaluated from an ecological point of view, which might bring novel insights into the future designs. Though, some considerations such as serviceability have been accounted for, which improves the lifetime of the various components increasing the products sustainability.

Furthermore, the general application of the Hexapod aim to enhance the sustainability of the production industry. It does this through eliminating the need to create new fixtures and tooling for each new product variation or assembly operation. This minimizes the raw materials used and greatly reduces the storage space needed, which lowers the cost and decreases the ecological footprint, Meaning the development of this product itself leads to a more sustainable industry.

8.1.7 Other design considerations

Other than the above-mentioned points of interest, there are many small design changes that must be investigated and applied properly.

Those are selection of standard fasteners, mounting solution for joints to top and base plate, cables, and cable management as well as sensors and sensor mounts. Verifying the fitting tolerance grades for all components and applying them in the design.

Once all changes are made, drawings, assembly instructions and BOM (bill of materials) should be created.

8.1.8 Cost analysis

Most of the parts have not been analysed from an economical perspective. Some offers have been received but have not been analysed or compared in this thesis. This is due to a budget not being specified and the lack of response from certain manufacturing parties. In future iterations of the project, this needs to be completed.

8.1.9 Methodology

The methodology presented in this thesis (chapter 2) was more of a plan, which in practice was not followed to the letter. There were a lot of deviations due to many factors such as time constraints, lack of data or lack of experience. As an example, investigating the serviceability was not performed on the actuator in phase one. It is noticeable that the serviceability was only addressed in later iteration of the actuator. This experience, lead to planning for the serviceability of the joints and motor already in the earlier phase. Similarly, the development of the frameless motor happened quicker and more efficiently due to better planning and deeper understanding of the problem/requirements, which led to less unnecessary research, smoother implementation and minimized the need to re verify the iterations.

Another deviation of the plan was when iterating the process. The first iteration went through each phase more thoroughly, while the following iterations would skip several steps especially parts of the research. For example, several bearing arrangements were studied, and initially the locating/non-locating was chosen, however this was later changed, and the research wasn’t performed again as the data was already collected on this topic.

Lastly, all phases did not have the same weight, or importance. The research phase and the implementation phase were the focus of this thesis. The verification phase played a key part in iterating the concept, however, since the simulations could not be performed, more analytical improvements could not be made. The review stage also experienced some difficulties, mostly due to time constraints. Organizing proper meetings at the company could take weeks as everyone was busy with their own projects. The actuator went through a more proper review which can be seen in chapter 4.3.1. However, when the joint and frameless motor was reaching this phase there was a hectic situation with other projects at Cognibotics. So, the review happened more informally, through digital information exchanges, which affected the quality of the evaluation. Nevertheless, phases three and four will see greater use when further iterations of the Hexapod are performed, and while the research phase is larger at the start of such a project, it will find lesser spotlights in the future.

8.2 Conclusion

A concept has been developed for the three parts (Actuator, Joints, Motor) which make up the leg and the dynamic part of the Hexapod. The necessary components have been researched and chosen based on the requirements. Interface to hold and connect these components together have been designed. Assembly/disassembly and serviceability of the three parts has been considered.

However, the Hexapod is far from being complete. To start with, a proper base and top plate needs to be designed. These plates make up about 50-70% of the mass of the Hexapod, their dimensions and shapes define how the legs will be attached and what possible workspace the robot can have, and their material and geometry will define the rigidity and thus also the accuracy of the entire Hexapod.

As discusses in 8.1.2, a frameless motor concept is not ideal at the early stages of the project, thus designing a belt or cog driven commercial motor mount is required.

Once all these parts, and the more detailed pieces like cables, cutouts, as well as material choices and weight/size optimization has been completed, simulations of the robots should be performed. Both on induvial parts, and on the Hexapod as a whole. This will shed light on new issues which can and should be tackled before the creation of the first prototype.

Following that, a prototype should be created and thoroughly tested and applied to the use-case specified by the potential consumers. This will lead back to many new iterations of both research, implementation, and verification of these concepts.

Even without any specific requirements/use-case, this version of the Hexapod could be produced, used in early testing and eventually as a model exposition. The concept is, with minor additions, complete and should work to perform basic movement which can be used to attract potential investors and costumers. As the Hexapod leg concept was completed, we consider the thesis goal achieved.

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