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Mechatronics Advanced Course, Spring Term

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Atlas Copco HK Project, Spring

Power Tool Test Rig for Ergonomic Assessment and Optimisation of Tightening Algorithms

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Abstract

Today, a very real problem that often goes under the radar is a global health issue experienced by the assemblers in the assembly industry. This health issue comes in forms of injuries such as hand-arm vibrations syndrome and carpal tunnel syndrome. That is where this project comes into view. This report is a pre-study to a design project with the goal of developing a nutrunner test rig on behalf of the company Atlas Copco, one of the market leaders in the assembly tools industry. With the test rig, they hope to be able to improve and optimise their nutrunners further to decrease the risk of permanent damage for the assemblers. This pre-study has therefore gathered information pertinent to the task at hand, including research regarding the ergonomics of bolt tightening, ways of modelling different operators and ways of realising these models. The research then culminates in a design proposal, which will continue development during the fall of 2021.

Acknowledgements

We would like to thank Atlas Copco for offering this interesting project to us, as well as our contact person Sofia Olsson along with the other stakeholders at Atlas Copco, giving us feedback and support throughout the project. We would also like to thank our supervisor Daniel Frede for supporting and helping us, for example by giving us advice on how to handle different issues. We would also like to thank Claes Tisell and Henrik Sandberg for taking the time helping us by providing professional knowledge and inputs.

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Nomenclature

Physics Constants

ω_{in} Rotational speed of input shaft

ω_{out} Rotational speed of output shaft

θ Angle of the Nutrunner

A_{piston} Area of the pneumatic piston

b Damping Constant

B_m Damping Constant Muscle

c Speed of light in a vacuum inertial system

F_{max} Maximum Muscle Force

F_{mc} Force at which the Muscle Becomes Linear

F_{tc} Force at which the Tendon Becomes Linear

g Gravitational Constant

h Plank Constant

$I_{operator}$ Inertia of the Operator

I_{tool} Inertia of the Nutrunner

k Spring Constant

k_m Spring Constant Muscle

k_t Spring Constant Tendon

k_{me} Shape Parameter Muscle

k_{ml} Linear Spring Constant Muscle

k_{te} Shape Parameter Tendon

k_{tl} Linear Spring Constant Tendon

l Length

l_m Muscle Length

l_{mc} Length at which the muscle becomes linear

l_{mopt} Optimum Muscle Length

l_{ms} Slack Muscle Length

l_{tc} Length at which the tendon becomes linear

l_{ts} Slack Tendon Length

l_t Tendon Length

n Number of moles

P_{c1} Pressure in chamber 1

P_{c2} Pressure in chamber 2

P_{in} Initial pressure

R Ideal gas constant

R_{in} Radius of input transmission component

R_{out} Radius of ourput transmission component

T Temperature

T_{in} Torque carried through input shaft

T_{out} Torque carried through output shaft

v_m Muscle Contraction Velocity

V_{c1} Volume in chamber 1

V_{c2} Volume in chamber 2

V_{in} Initial volume

v_{max} Macimum Muscle Contraction Velocity

w Width of Force-Length Relationship

x Linear Displacement

x_{cyl} Half way point of the pneumatic cylinder

x_{dis} Piston displacement

Abbreviations

KTH Kungliga Tekniska Högskolan
HAVS Hand-arm vibration Syndrome
CTS Carpal Tunnel Syndrome
MSD Musculoskeletal Disorder
IRTT Inline Rotary Torque Transducer
CE Contractile Element
PE Parallel Element
SE Series Element
SSCs Stretch-Shortening Cycles
EMG Electromyography
SOTA State Of The Art
FEA Finite Element Analysis
PMI Project Management Institute
CVT Continuously Variable Transmission

1 Introduction

Atlas Copco's Industrial Technique business area provides industrial power tools and systems, industrial assembly solutions, quality assurance products, software and service through a global network. Atlas Copco's customers oftentimes work in environments which are physically demanding on the human body, such as assembly work. Atlas Copco continuously develops new and better ways to understand and improve the ergonomics for the operators of their handheld tools. Weight, noise and physical design are some important ergonomic parameters, but the main focus in this project are vibrations and reaction forces from the tool during bolt tightening.

The project has several stakeholders at Atlas Copco, both the team working with developing tightening algorithms and developing tightening techniques, as well as the team working with ergonomics. Except from a challenging, multidisciplinary task, Atlas Copco offers the group at Kungliga Tekniska Högskolan (KTH) a lot of engagement and feedback to help the group solve the problem and grow from it!

1.1 Background

1.1.1 Nutrunners

A nutrunner is a tool used in industry for tightening bolts and nuts. There are two main types of tools, classified by the way they are held by the operator: pistol grip and right angle grip. See figure 1 for a comparison of the two grip types.

Figure 1: Examples of a pistol grip nutrunner [1] (left) and a right-angle nutrunner [2] (right).

A right angle nutrunner is preferred for tightenings that require high torque. Compared to a pistol grip nutrunner, it has three times lower vibrations during a tightening and the risk for white finger syndrome is more than three times smaller.[3]

1.1.2 Tightening Techniques

There are many methods for tightening bolts. The most common ones are torque control, angle control, and yield control. Torque control measures the tightening torque and stops tightening when a predetermined torque is reached. This is very simple but as the friction in the joint can vary widely, the tension in the bolt is difficult to predict. In angle control the nut is turned to a specified angle. The angle can be difficult to measure precisely and the required tightening angle can vary between different joints. Yield control requires measurement of both torque and angle. By monitoring both the torque and the nut angle, it is possible to detect when the yield point of the bolt is reached.[4]

Atlas Copco uses the previously mentioned tightening strategies, but with extra functionality and more advanced control. This report considers three different Atlas Copco tightening strategies: *Turbo Tight*®, *Quick Step*, and *TensorPulse*. *Turbo Tight*® calculates the rotational energy in the tool (motor rotor and gearbox) and predicts when the bolt will be fully tightened. The goal is to let the rotational energy inside the tool power the last bit of the tightening process, and thus achieve a very small reaction force to the operator. *Quick Step* has two phases. In the first phase the bolt is tightened to a first torque, and in the second phase the speed is reduced and the final torque is reached. This decreases the effects of joint relaxation, but it is not as fast as *Turbo Tight*® and has a larger reaction force [5]. *TensorPulse* tightens the bolt using pulses. The pulses lowers the impulse acting on the operator, but give rise to vibrations.

1.1.3 Ergonomics of Bolt Tightening Tools

When a nutrunner is tightening a bolt or a nut, a reaction torque occurs as an effect on the tool and the operator. As a consequence, the operator needs to apply an equal and opposite force at the tool handle to counteract the reaction torque. This can be seen in figure 2.[6] Apart from the reaction torque and the reaction force, the operator is also exposed to vibrations when using a bolt tightening tool. All these types of exposures contribute to risk factors of the operator's health.

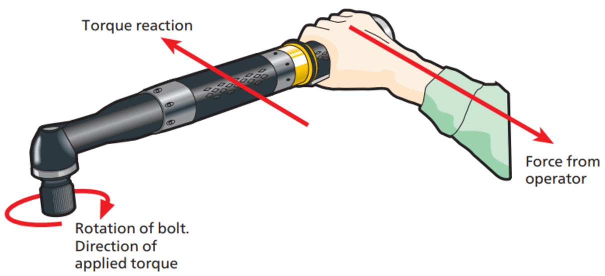

Figure 2: Reaction torque on the operator and resulting reaction force from the operator. [7]

When evaluating the ergonomics of a nutrunner the vibrations on the operator are of interest. When the operator is using the tool, vibrations are planted through the fingers, hands and arms. These types of vibrations are also called hand-arm vibrations. Hand-arm vibrations are today regarded to be significantly related to serious disorders from workplaces.[8] Hand-arm vibration Syndrome (HAVS) is a condition of several serious symptoms for damage caused by vibrations. The most common symptoms are tingling and numbness in the fingers, and loss of strengths in the hands and white fingers. White fingers is both painful and very serious.[9] Another term for white fingers is Raynaud's phenomenon, which is a disorder of decreased blood circulation in the fingers or toes. If white fingers are not discovered in an early phase, it can result in permanent limitation of blood circulation in the fingers.[10]

Today there are strong regulations and strict standards for both manufacturing new vibrating hand-tools and how these tools should be used from a health perspective. The EU has a legal act called The Directive, which among other things lists the minimum standards of acceptable vibrations exposures. The Directive selected the the action value of vibration exposure to be 2.5 m/s^2 and the limit value of vibration exposure to 5 m/s^2, see figure 3. These values are average A-weighted values over an 8h working day. From experiences the right angle nutrunners at Atlas Copco are considered to expose the operator to low vibrations.[8] However, it is still important to have vibrations in mind when evaluating ergonomics in nutrunners.

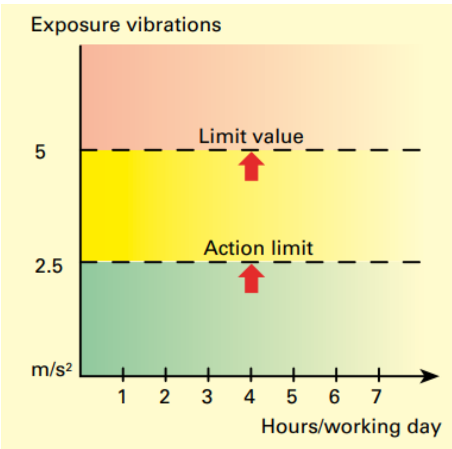

Figure 3: Action value and Limit value for vibrations exposure.[8]

Another common hand condition from working with powered hand tools like nutrunners is Carpal Tunnel Syndrome (CTS). The syndrome is caused by damage or compresses of the median nerve, which is the major nerve through the hand. The median nerve gets compressed when the operator is working with a bent wrist.[11] Some symptoms of CTS are pain, numbness, and tingling through the hands and arms.[12] In figure 4, the placement of the median nerve, as well as the areas of the hand that affect damages on the nerve are illustrated.

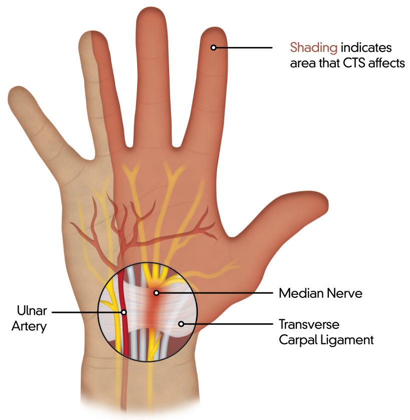

Figure 4: Illustration of damages on the median nerve and Carpal Tunnel Syndrome. [13]

Since the syndrome occurs when the wrist is twisted and exposed to intense loads, the design of the handle is of great importance. To minimise the strain of the operator, the design of the tool handle should enable the operator to keep the wrist straight. Another design factor of the tool that affects the load on the wrist is the weight. If a powered hand tool is too heavy for the operator it can result in injuries on the wrists and also decreased precision ability. This can in turn affect the quality of the operation.[6] Another aspect that also affects the strain of the hands and wrists is the position of the operator's centre of body mass. If the operator is standing in a stable position and using a two-hand grip on the nutrunner the operator can receive reaction forces better and it can decrease the level of tension in muscles. When an operator is using a nutrunner with only one hand, the operator's capability of handling the torque is significantly reduced.[11] Figure 5 demonstrates an operator holding the tool with a two-hand grip. Lastly, some other design factors that affect ergonomics of a nutrunner are: grip shape, centre of gravity of the tool, noise and lubrication between components.[11]

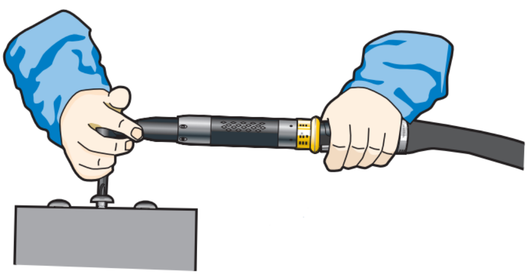

Figure 5: Operator holding a nutrunner with a two-hand grip. [7]

Besides from damaging the median nerve, another health risk when using a nutrunner is crushing or injuring the fingers. If the reaction torque on the operator is larger than 60 Nm, a reaction force bar is recommended to minimise the risk of injuries.[11]

Lastly, an important part when evaluating the ergonomics of an operator's work situation is to take the repetitive motions into account. Very common ergonomic risk factors in general are high repetition rate of monotonic motions and lack of recovery between uses of the tool. When the muscles of an operator get exhausted from repetitive motions it can lead to upper Musculoskeletal Disorder (MSD). MSDs include a wide range of injuries on parts like joints, ligaments, muscles and nerves. When the operator is using the nutrunner, the operator needs to increase the grip force in the hand and to activate the muscle groups in the forearm during the rundown phase of the bolt tightening. This in order to resist the resulting reaction torque from the nutrunner.[14] After many repetitions of using a nutrunner these muscle groups can get exhausted and potentially injured if the muscles do not recover.

1.1.4 The Value of Improving Ergonomics in Tools

Including ergonomics in the process of developing and evaluating products used by operators has an enormous value in several aspects. Firstly, it can secure the operator's safety, physical health and mental well-being. By improving the ergonomics in nutrunners, the risks of the serious conditions like hand-arm vibrations syndrome, CTS and musculoskeletal disorders can be minimised. Secondly, when the operator feels safe and comfortable when using the tools, the job satisfaction can be improved. A great job satisfaction can in turn also increase job motivation and result in great productivity and quality of the operations. This can in turn create great benefits for the employees at companies.[6] Atlas Copco has throughout the years seen that there is a great economic benefit of including ergonomics in product development.

Atlas Copco has a long and rich history of including ergonomics in development of hand-held powered tools. In the 1950's, Atlas Copco began to apply ergonomics in the development of a drill. Medical experts were a part of the designing process and gave their opinions on different types of grips. The resulting product was a great success and became very popular on the market.[11] Today, ergonomics is a strong part of Atlas Copco's brand and is an important factor for their successful tools. Even though the ergonomics of the existing nutrunners has been improved throughout the years, there is still a strong strive at Atlas Copco for continued improvement of the tool. In the end these improvements would help and relieve the operator.

1.2 Project Description

The vibration and reaction force levels the operator is subjected to depend on several factors, for example the tool type, the pre-programmed tightening algorithm, the joint type and the operator himself. The operator's impact on the vibration and reaction force levels varies uncontrollably from test to test, as it depends on how much the operator resists the movements of the tool. In order to perfect Atlas Copco's ergonomic measurement methods and develop their tightening algorithms, the company is in need of a controlled test environment with a repetitive model of an operator. As mentioned in 1.1.4, there is a large incentive to develop ergonomic tools. It has large benefits for the operator and the assembly line as a whole, and that is why Atlas Copco is in need of a test rig for reliable ergonomic assessment.

Atlas Copco is currently in possession of a simple nutrunner test rig, see figure 6. It has only passive components and is therefore limited in its functionality. The goal is to construct a test rig that can more accurately simulate a human operator.

Figure 6: One of the current test rigs at Atlas Copco.

1.2.1 Goals and Impacts of the Project

The goal of this project is to develop an operator-independent test rig for ergonomic assessment and optimisation of new tightening algorithms. The test rig will provide a systematic, repetitive test method to output comparable test results and optimised parameters for tightening algorithms.

1.2.2 Deliverables and Most Important Issues

This report is written during the spring term 2021. The project will continue in the fall of 2021, when the group will focus on constructing the test rig. The main goal with this report is to present information about the problem and related fields, i.e.

a state of the art analysis. The state of the art analysis will give the project group, as well as the reader of this report, a knowledge foundation to better understand the problem and make choices during the design process.

This spring term report will also present the design concepts with a concept evaluation and a final concept that will be constructed during the fall. The plan for the fall is presented, this includes a GANTT schedule, required components with a cost analysis, and a risk analysis.

1.2.3 Project Organisation

The project group consists of nine students. During the state of the art analysis the project group was split into subgroups of three. When designing the concepts the group was also split into groups of three, with one member from each state of the art group in each concept group. This resulted in each concept group having knowledge from the whole state of the art analysis, while the group combinations were all new.

The group had meetings several times per week to discuss the current state of the project. Because of the Covid-19 pandemic, meetings were held online. Meetings within the group were handled through Discord [15] or Zoom [16]. To keep in touch with the coach and the stakeholders, weekly meetings were held with the coach Daniel Frede through Zoom, and meetings every other week with the stakeholders at Atlas Copco through Microsoft Teams [17].

1.3 Requirements

By close communication with the stakeholders, the stakeholder requirements have continuously been refined and updated during the course of this project. The requirements are divided into different subcategories such as Operator Model, Physical Requirements, Stakeholder Suggestions, Measurement, Wishes, Safety, and Costs.

1.3.1 Stakeholder Requirements

Operator Model:

- Simulate a human operator accurately (compared to Atlas Copco test data)
- Enable repetitive and accurate test results (of reaction forces and accelerations)
- Being able to test new tightening strategies (redundancy, not just one tightening technique)

Physical Requirements:

- Be robust enough to operate during heavy vibrations

- An Inline Rotary Torque Transducer (IRTT) must fit between the tool and the bolt
- Be robust enough to withstand extreme cases (reaction torques up to 100 Nm)
- Fit a beam test joint (a special part supplied by Atlas Copco)
- Adjust for height difference from bolt tightening
- Work for different tightening strategies (Quickstep, TurboTight and TensorPulse)
- Be able to send data to a Dewe 43
- Work with three different angle tools: STR series ETV STR61-50-10, ETV STR61-70-13, ETV STR61-100-13
- 60° rotary displacement

Stakeholder Suggestions:

- Have a lifespan of >10 000 test runs
- Movable by two people
- Short list of components and/or low manufacturing complexity
- Easy maintenance (in terms of sustainability)
- Independent installation of tool (ease of use)

Measurement:

- Accurately measure reaction torque on nut
- Accurately measure vibrations of the tool
- Accurately measure reaction force to the operator

Wishes:

- Work with pistol grip tools
- Simulate different types of operators (different model parameters, at least as continued work)
- Different operator positions can be tested, horizontally and vertically

Safety:

- Protect against injuries, e.g. finger injuries, while running the test rig
- Have an emergency system that stops all operations
- Noise levels below harmful threshold

Costs:

- Enable component and manufacturing costs within budget

1.4 Delimitations

After some initial research, the project group decided to focus on the main requirements and put some of the wishes aside to reduce complexity of the construction. The pistol grip tools have been chosen to not take into account when designing the rig. The main focus has instead been put on the different angle tools, and that the rig should have the ability to change operator characteristics such as inertia, stiffness etc.

The operator model used in this project is similar to a spring-mass-damper system combined with an actuator in order to have the ability to actively control the system. Other operator models, such as the Hill model are discussed, but are not realised in this project.

1.5 Interviews

During the State Of The Art (SOTA) review, which is explored further in section 2, a few interviews were also conducted. These were carried out to gain knowledge and to utilise experience already developed by experts within the fields relevant to the project.

Claes Tisell, lecturer in Systems and Component Design, and Henrik Sandberg, professor in Decision and Control Systems, were interviewed, and a brief summary of these conversations are provided below. Also, a session was held with our stakeholder Sofia Olsson, where the project members for example were allowed to test a nutrunner. A summary of this is also provided below.

1.5.1 Interview with Claes Tisell

During the meeting the realisation of the operator model was discussed. It was also discussed how the data provided by Atlas Copco, showing the reaction force on an operator during nut tightening, can be used to determine the parameters for the operator model. Claes provided the group with contact information to researchers at KTH specialising in biomechanics, active damping, ergonomics, as well as electric and mechanical components.

1.5.2 Interview with Henrik Sandberg

During the meeting it was discussed how an electric motor could be controlled to simulate a mass-spring-damper system. It was also discussed which type of controller that should be used. Henrik provided the group with educational material written by Bo Bernharsson and Karl Johan Åström about active damping. He also recommended research done by the Robotics Lab at Lunds University that might be relevant for this project.

1.5.3 Bolt Tightening with Sofia Olsson

Early on it was discovered that the project group would benefit greatly in the concept phase if some hands-on experience with the tools in question could be acquired. Therefore, a session was scheduled with Sofia Olsson from Atlas Copco to let each group member try a couple of different tightening techniques with a right angle nutrunner. In figure 7 one of the project members is demonstrating the two hand grip most commonly used during tightening with a right angle nut-runner.

Figure 7: Project members tested to use a nutrunner from Atlas Copco.

During this session Sofia also provided, in addition to the chance of testing a nutrunner, a lecture regarding the testing procedures at Atlas Copco. What software they use, what the results are used for etc. One of the key points from the lecture was Dewesoft, a software for processing data. This was extra fruitful as this is one of the interfaces that the test rig will need compatibility with.

1.6 Reader's Guide / Report Disposition

The report is divided into four chapters and follows the pattern:

- Introduction
- State of the art studies
- Concept design
- Future work

In the chapter State of the art the reader will be presented to all research performed during this project. The chapter includes studies of:

- Operator models
- Damping system and realisation of damping systems
- Actuators
- Transmission
- Variable mass
- Sensors
- Bearings
- Existing test rigs

In the following chapter Concept design three different concepts, which is based on the State of the art, is presented. The three different concepts differs from one another so in the end of the chapter a concept evaluation is presented which explains the reasoning behind choosing one of them. In the last chapter Future work the project plan for the fall is presented, as well as a cost analysis and budgeting and other activities to be performed in the fall.

2 State of the Art

Section 2 covers the preliminary research done, the state of the art analysis, in order to later be able to utilise this information in the concept design.

2.1 Operator Models

This section describes the human operator, different methods of simulating the behaviour of the human operator by using operator models, what considerations have to be made when choosing an operator mode as well as how the parameters for the operator model can be determined. This to find a suitable operator model for use in the test rig.

2.1.1 Skeletal Muscle Anatomy

In order to understand the operator models, it is important to first understand how the human operator behaves. For the purposes of this report, the human hand-arm system will be of interest. There exist three different types of muscles in the human body; skeletal, heart and smooth muscles [18]. For the purpose of this report, only skeletal muscles will be described as they are involved in movements. In figure 8 a skeletal muscle is shown.

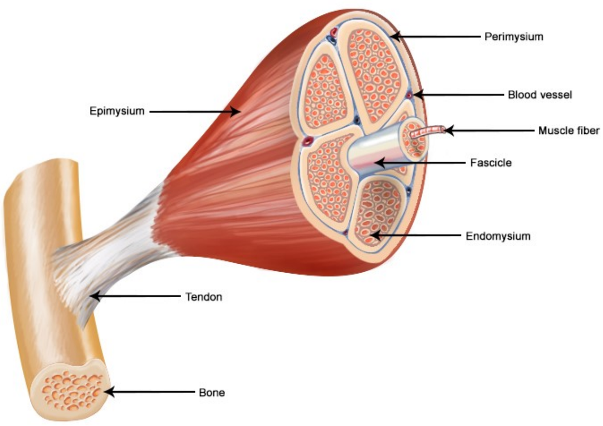

Figure 8: Depiction of a skeletal muscle.[19]

The inside of the skeletal muscles are comprised of bundles of muscle fibres, known as fascicles. A layer of connective tissue, Perimysium, surrounds the fascicles. Inside the fascicles each muscle fiber is in turn separated by connective tissue, referred to as Endomysium. The entire muscle is then enclosed in a connective tissue, Epimysium, that allows the muscle to contract and move while maintaining its structural integrity.[20] The muscles are connected to the skeleton via tendons, that transfers the force from the muscles [18].

To generate a force a neural impulse from the nerve system is sent to the muscle tissue, causing it to contract. The contraction causes the viscoelastic properties of the muscle to change. When the muscle is stimulated at a high frequency, it generates its maximal tension.[18]

2.1.2 Human Reaction Time

The reaction time is defined as the time it takes from a stimuli to when a response is performed. During nutrunner operation a stimuli could be the onset of torque build-up.

A study performed by Oh and Radwin indicated that there is a time delay between torque build-up and when a muscle activity burst occurs, defined as Electromyography (EMG) latency. In the study the muscle EMG activity was measured during nutrunner operation. It was concluded that the EMG latency is greater for softer joints with larger torque build-up time, and that a harder joint with shorter torque build-up time results in a lower EMG latency.[21]

2.1.3 The Use of an Operator Model

An operator model is a mathematical model used in order to describe the behaviour of the human operator. These models can be used in order to estimate the force the human operator applies to the tool during bolt tightening. There are multiple different operator models that use different methods of simulating the human operator. Some models are more biologically based simulating the muscles and tendons while others are at a higher level, lumping the effects of the different muscles and tendons together. When choosing an operator model some aspects to take into consideration are the complexity of the model, the accuracy of the model as well as how the parameters of the model can be determined for different operators. Three of the most common operator models are described below.

2.1.4 Mass-Spring-Damper Model

One of the most commonly used operator models in nutrunner test rigs is based on the mass-spring-damper system shown in figure 9. The model uses a spring, mass and damper to simulate the dynamics of the human hand-arm system. Instead of simulating all muscles, joints and tendons in the human arm, their influences are lumped together into three parameters. These parameters are the spring constant k, damping constant b and mass m. The values of these parameters will vary with different operators. Some factors that have significant impact on the value of the parameters are the gender of the operator as well as horizontal and vertical location of the tool in relation to the operator [22]. There is also a possibility that these parameters will vary during a tightening as different muscle groups are activated [23].

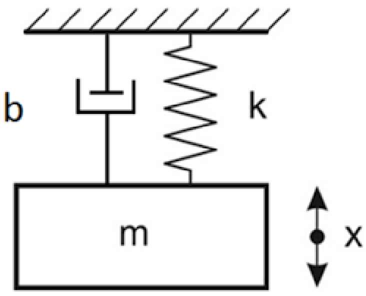

Figure 9: Depiction of the mass-spring-damper model with spring constant k, damper coefficient b and mass m.

The equation describing the dynamics of the mass-spring-damper system from figure 9 is shown in equation (1).

$$F(t) = m\frac{d^2x}{dt} + b\frac{dx}{dt} + kx \tag{1}$$

where m is the mass, F is the force acting on the mass, x is the displacement, b is the damping constant, and k is the spring constant.

From the mass-spring-damper system in figure 9 the operator model can be simulated according to figure 10. Where θ is the angle of the tool and l is the length between the axis of rotation of the tool to the point where the operator grips the handle.

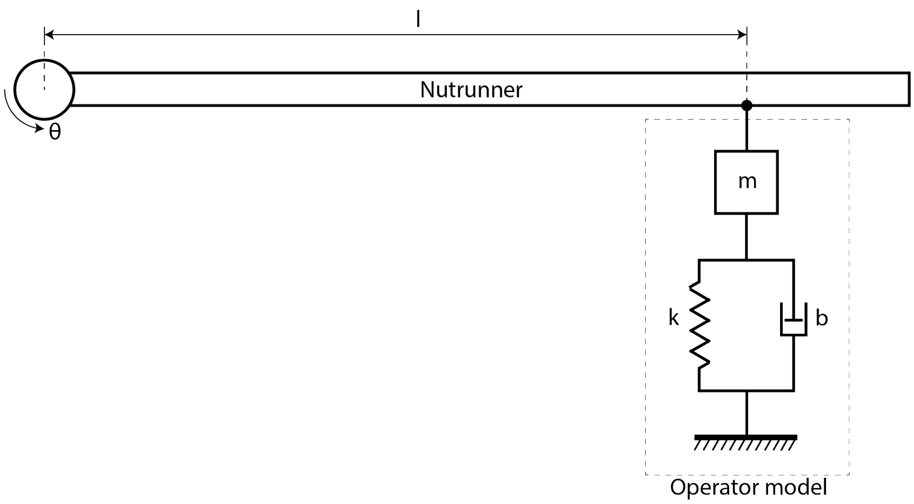

Figure 10: Depiction of the spring-mass-damper operator model with spring constant k, damper coefficient b, mass m, length l and angle θ.

The dynamic equation of the operator model can then be derived according to equation (2).

$$F(t) = \frac{I_{tool} + I_{operator}}{l} \cdot \frac{d^2\theta}{dt} + bl\frac{d\theta}{dt} + kl\theta \tag{2}$$

where F is the force acting on the tool handle, I_{tool} and $I_{operator}$ are the inertia of the tool and operator respectively, l is the length between the axis of rotation of the tool to the point where the operator grips the handle, θ is the angle of the tool, b is the damping constant and k is the spring constant.

By lumping the parameters together the complexity of the model is reduced as only the value of these three parameters has to be determined. At the same time, this results in the model ignoring physiological phenomenons though. However, even though the model is ignoring physiological phenomenons, it has been concluded that the operator model produces satisfactory results for ergonomic assessment.[22]

In order for the operator model to accurately estimate the dynamics of the human operator the accuracy of the parameter identification is important. There are several ways to identify the parameters. One way is to identify the parameters from studying the time domain response of the angle displacement and reaction force when a human operator is operating the tool [24]. Another way of identifying the parameters is by using the grey-box model, where equations can be combined with data to determine the parameters and generate a complete model [25].

2.1.5 Hill Model

The Hill model is commonly used in biomechanics to simulate the behaviour of human muscles. The model simulates the active and passive muscle forces and is considered an macroscopic muscle model since it simulates the muscle forces on an organ level [26]. The output of the Hill model is a one dimensional force and the input is the muscle length, muscle contraction velocity and an activation signal, similar to the neural impulse. The parameters for the Hill model are biologically based and can be determined from experiments.

The Hill model consists of three components as can be seen in figure 11: the Contractile Element (CE) that generates a force from an activation signal, the Parallel Element (PE) that has viscoelastic properties simulating the connective tissue in a muscle and the Series Element (SE) that simulates the effects of the tendons and aponeuroses [27].

Since the Hill model only simulates one muscle, the complexity of the operator model is increased greatly compared to the mass-spring-damper model. This is because the effect of multiple muscles and joints have to be combined in order to simulate the

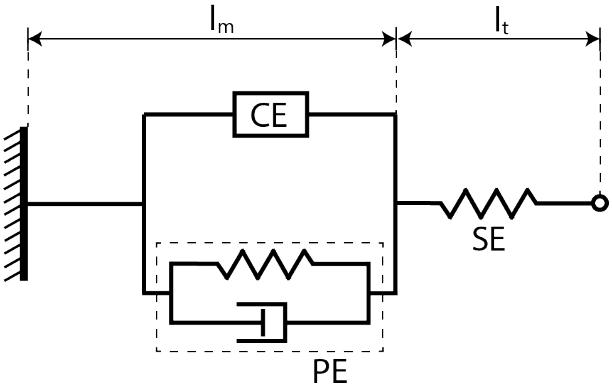

Figure 11: Depiction of the Hill model with muscle length l_m and tendon length l_t.

hand-arm system. This has been done previously where a human arm was modelled in software with a three-link planar manipulator, figure 12, activated by the Hill model [28]. However the Hill model has not yet been implemented physically in a nutrunner test rig.

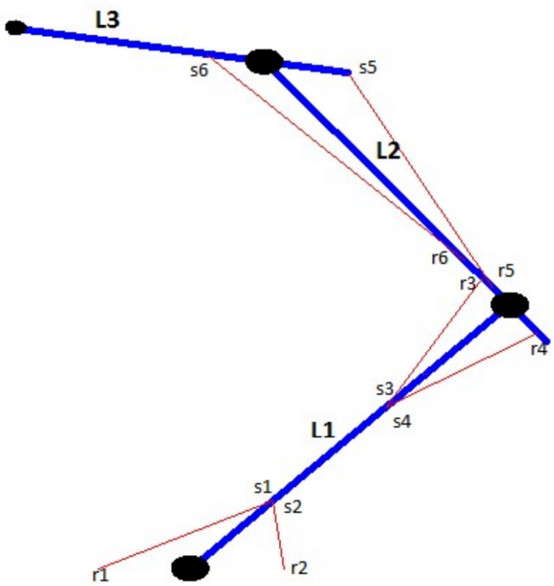

Figure 12: Three-link arm model, blue represents links, red represents muscles and black represents joints.[28]

One of the benefits of the Hill model, when compared to the mass-spring damper model only consisting of passive components, is that it has the ability of reproducing the force from active muscle contraction. This achieved by the CE element, the force generated is given by equation (3).

$$F_{CE} = F_{max}a(t)f(l_m)g(v_m) \tag{3}$$

where F_{max} is the maximum possible force, $f(l_m)$ is the force-length relationship of the muscle and $g(v_m)$ is the force-velocity relationship of the muscle.

The force-length relationship is given by equation (4).

$$f(l_m) = 1 - \left(\frac{\frac{l_m}{l_{mopt}} - 1}{w} \right)^2 \tag{4}$$

where l_m is the muscle length, l_{mopt} is the optimum muscle length and w is the width of the force-length relationship.

The force-velocity relationship is given by equation (5).

$$g(v_m) = \frac{1 - \frac{v_m}{v_{max}}}{1 + 4\frac{v_m}{v_{max}}} \tag{5}$$

where v_m is the muscle contraction velocity and v_{max} is the maximum contraction velocity.

The PE in the Hill model represents the connective tissue that runs in parallel to the muscle fibers and consists of a spring and damper [18]. The force generated by the spring effect shown in equation (6) can be seen as partially exponential until it reaches maximum stiffness, then it behaves as a linear spring [27].

$$F_p(l_m) = \begin{cases} 0, & l_m < l_{ms} \\ \frac{k_{ml}}{k_{me}}(e^{k_{me}(l_m - l_{ms})} - 1), & l_{ms} \leq l_m \leq l_{mc} \\ F_{mc} + k_m(l_m - l_{mc}), & l_{mc} < l_m \end{cases} \tag{6}$$

where l_m is the muscle length, l_{ms} is the slack muscle length, l_{mc} and F_{mc} are the length and force when the spring effect changes from exponential to linear respective, k_m and k_{ml} are spring constants and k_{me} is a shape parameter.

The contribution of the damping effect is given by equation (7).

$$F_d(v_m) = B_m v_m \tag{7}$$

where B_m is a damping constant and v_m is the muscle contraction velocity.

The force generated by the PE is then given by the sum of the spring force and damping force according to equation (8).

$$F_{PE} = F_p + F_d \tag{8}$$

In order to represent the force transmitted by a tendon the Hill model uses an SE. The force transmitted by the SE is shown in equation (9). Similar to the PE the force transmitted can also be seen as partially exponential until the maximum tendon stiffness is reached, then it behaves as a linear spring[27].

$$F_{SE}(l_t) = \begin{cases} 0, & l_t < l_{ts} \\ \frac{k_{tl}}{k_{te}}(e^{k_{te}(l_t - l_{ts})} - 1), & l_{ts} \leq l_t \leq l_{tc} \\ F_{tc} + k_t(l_t - l_{tc}), & l_{tc} < l_t \end{cases} \tag{9}$$

where l_t is the actual tendon length, l_{ts} is the slack tendon length, l_{tc} and F_{tc} are the length and force when the spring effect changes from exponential to linear respective, k_t and k_{tl} are spring constants and k_{te} is a shape parameter.

2.1.6 Huxley Model

The Huxley model is said to be a model on microscopic level and is more detailed than the Hill model [18]. Compared to the Hill model the Huxley model contains a relationship between metabolic and mechanical behaviour, which is missing in the Hill model [29]. This leads to a more complex model and increases the computational power needed.

Even though the relationship between metabolic and mechanical behaviour is missing in the Hill model, a study has shown that the two models are equally valid for predicting mechanical behaviour. The study evaluated how accurately the Hill and Huxley model could predict the mechanical muscle behaviour during Stretch-Shortening Cycles (SSCs). This was done by comparing the predictions of the two models to SSCs performed on rats.[29]

As the Huxley model is more complex, thus requiring higher computational power in combination with the conclusion that the predictions from the Hill and Huxley model are equally valid. It was decided that the Huxley model would not be investigated in more detail.

2.2 Damping Systems

A damping system is an assembly that through its set-up and system model reduces the vibrations of the system it is integrated into. However, depending on the mentioned set-up, a damping system can achieve tasks beyond only reducing the vibrations. The three main set-ups of a damping system are the passive, the semi-active and the active set-up. These are described further in this chapter.

2.2.1 Passive Damping

A passive damping system depends only upon components that have time-independent characteristics. Therefore, the agency of a passive system is quite limited. One of the main use cases of a passive system is therefore simple vibration reductions. This is not to say that a passive system is useless. On the contrary, its simplicity often makes it the preferred solution for such use cases [30].

2.2.2 Semi-Active Damping

When the limited agency of a passive damping system is hindering, the semi-active set-up is the next option. The key feature that differs between the passive system and the semi-active system is the ability to change the damping coefficient. The coefficient is changed using a small source of power, and with that in different ways change the amount of power dissipated inside the damper.

2.2.3 Active Damping

The third and most sophisticated variant of damping systems used in the industry is the active damping system. It has, in addition to the existing damping system, a control structure that can in real-time adjust the oscillating behaviour of the system by either modifying the damping coefficient or by generating additional force which cancels out the undesired motion and helps the system to follow a desired trajectory.

2.3 Realisation of Damping Systems

When it comes to realising the different damping setups described in section 2.2, various options and components are available. This section delves into a selection of these.

2.3.1 Passive Damping - Mass-Spring-Damper

The least complicated of the three set-ups, the passive damping system, can be as simple as rubber feet on a vibrations prone machine in a workshop. However, when they are used in a more demanding setting, they are often comprised of a spring, a damper and a mass [30]. What type of spring and damper that are implemented varies from use case to use case. The only requirement for the system to be classified as passive, is that the components chosen for the implementation are to have time-independent characteristics. A typical example would be a standard spring. However, also fluid induced spring and damping effects can be employed. An example would be a passive hydraulic cylinder.

It is this fact, that each of the components are passive, that results in the limited agency of the passive damping system mentioned in subsection 2.2.1. A passive component will only dissipate energy, never induce it. This means that the passive damping system is subject to the incoming vibrations entirely, without the power to output any control [31].

2.3.2 Semi-Active Damping - Solenoid-/Servo Valve

There is a couple of different techniques for making hydraulic semi-active dampers but they are all based on the same fundamental idea. Since hydraulic oil is incompressible the sliding piston inside the hydraulic cylinder must allow hydraulic oil to flow through it as it slides. By changing the volume flow through the piston the power dissipated during action is manipulated. The servo-/solenoid valve technique is based on valves that opens and closes additional ducts for the hydraulic oil to flow through. The valves can be actuated either by a solenoid or a small servo which both is located inside the damper. These dampers often only have two different damping modes, when the valve is open and when the valve is closed, as illustrated in figure 13. They do also not all have the ability to be continuously variable. The difference between the two valve techniques is that the servo valve offers much faster and accurate response, but they are also much more complicated in design leading to higher production costs.[32]

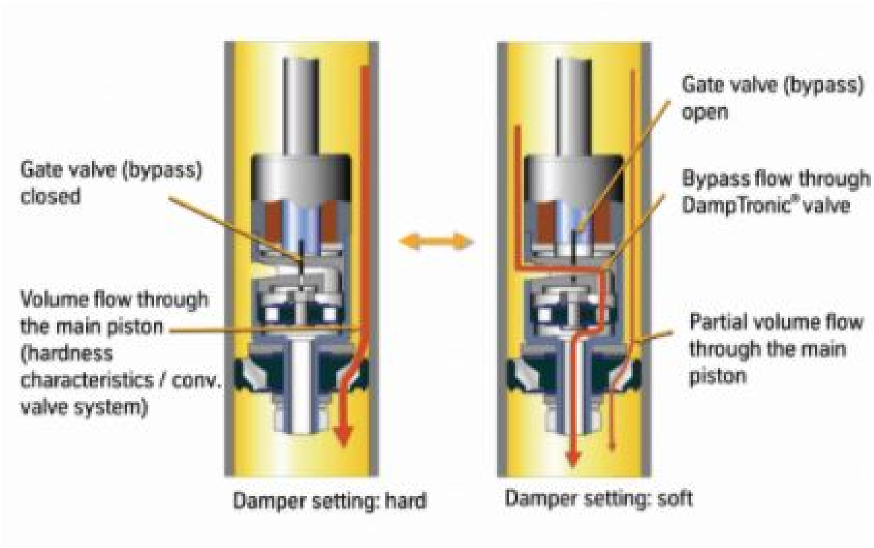

Figure 13: Fundamental functionality of a valve based damper.[33]

2.3.3 Semi-Active Damping - the Fluids

The volume flow through the main piston can also be changed by manipulate the characteristics of the hydraulic fluid. To accomplish this either magneto- or electrorheological fluids can be used. The working principle for this setup is to change the viscosity of the fluid by applying an electric or magnetic field. This results in a change of the volume flow since the ducts have constant area. When a magnetic or electric field is applied the micron-sized polarisable particles suspended in the fluid will align with the flux lines in a chain-like structure and hence almost instantly solidify as illustrated in figure 14. This solidification of the fluid will change the volume flow through the piston leading to a change in dissipated energy.

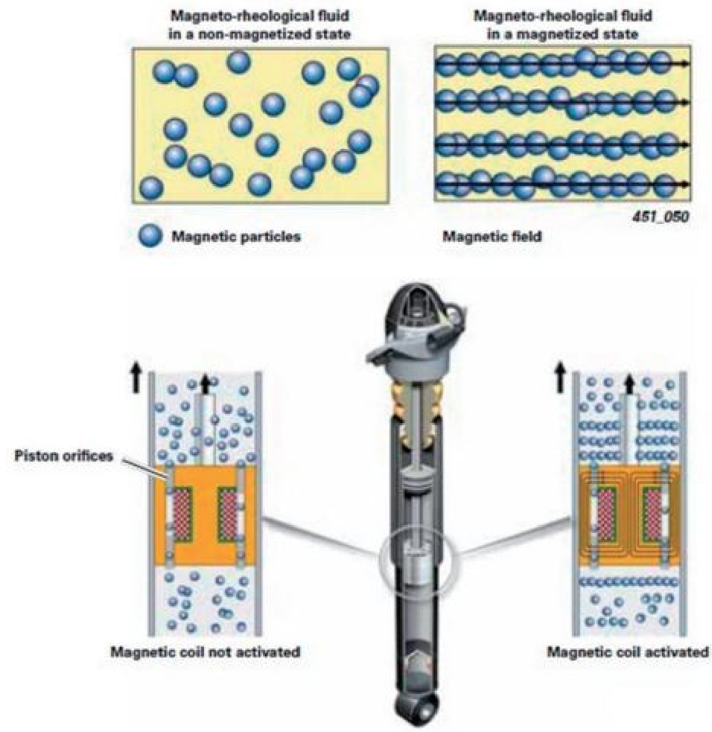

Figure 14: Fundamental functionality of an magneto-rheological damper.[32]

The two fluid techniques work in similar way but they do differ in characteristics as can be seen in table 1, which make the two alternatives suitable for different applications.

Table 1: Comparison between MR and ER fluids. [34]

Compared items	MR fluids	ER fluids
Yield strengths range (kPa)	up to 100	3–5
Operating temperature range	−45° to + 150°	−15 to + 150°
Voltage source requirement	12–24 V	2–5 kV
Current requirements	1–2 A	1–10 MA

MR: magnetorheological; ER: electrorheological.

2.3.4 Semi-Active Damping - Pneumatic Cylinders

Pneumatic cylinders can be used to realise a system with a wide range of different spring coefficient. By utilising the ideal gas law, analysis can be done on different types of pneumatic cylinders to ensure a linear characteristic of desired spring rates. This is done by positioning the piston of a double-acting cylinder at its center point, X_{cyl} with equal pressure on both sides, see figure 15.

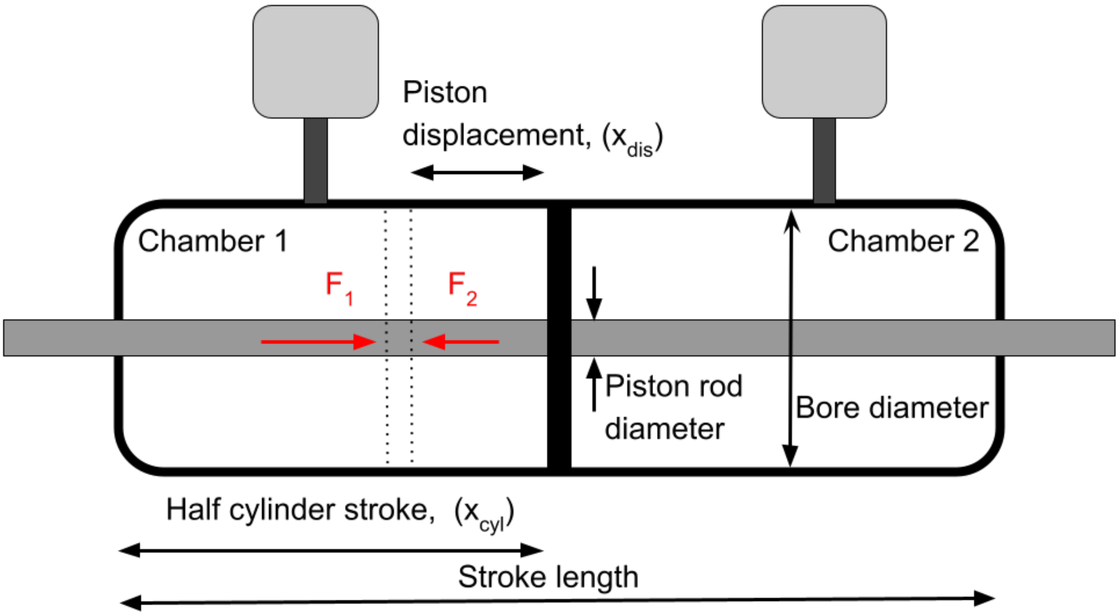

Figure 15: Schematic of double rod pneumatic cylinder.

Since the effective area of the piston and volume in the camber are identical on both sides, the piston will find itself in a resting position. If assuming that the piston has a perfect seal, so that no gas can leak from one chamber to the other, the ideal gas equation can be used to calculate the number of moles n in both chambers where R is the ideal gas constant, T is the temperature and P_{in} and V_{in} are the initial pressure and volume in both chambers.

$$n = \frac{P_{in}V_{in}}{RT} \tag{10}$$

If an external force is applied to either side of the rod, the piston will move a certain distance. When this happens, the volume and pressure in the two chambers will not be equal anymore. One of the chambers will get a larger volume and therefore a decrease in pressure and the opposite will happen in the other chamber. This chain of events can be explained by equation 11, 12, 13 and 14.

$$V_{c1} = A_{piston}(x_{cyl} - x_{dis}) \tag{11}$$

$$P_{c1} = \frac{nRT}{V_{c1}} \tag{12}$$

$$V_{c2} = A_{piston}(x_{cyl} + x_{dis}) \tag{13}$$

$$P_{c2} = \frac{nRT}{V_{c2}} \tag{14}$$

Since the effective piston area will remain constant while the piston is moving, the net force on the piston will increase. This will result in a linear relationship between the force and displacement of the piston which is equivalent to Hooke's law.

$$k = \frac{F_{piston}}{X} \tag{15}$$

2.3.5 Active Damping - Closed Loop Feedback Control

One commonly used setup of active damping consists of various sensors and actuators which are connected through a closed-loop feedback structure [31]. The sensors can pick up information from the system such as system voltage, velocity, current, torque, force, acceleration, positioning, etc. The generated data will then be compared to a pre-defined reference value which results in an error signal. This error term will be feed into a control unit which will modify the error term with the help of a mathematical algorithm. The output signal of the control unit will then be passed on to a "plant" which in this case is an actuator. When the actuator receives the updated control output, it will adjust its performance so that the added energy can counter the system vibrations and oscillations, see figure 16. This type of control setup is a reactive system since it reacts to the system output.

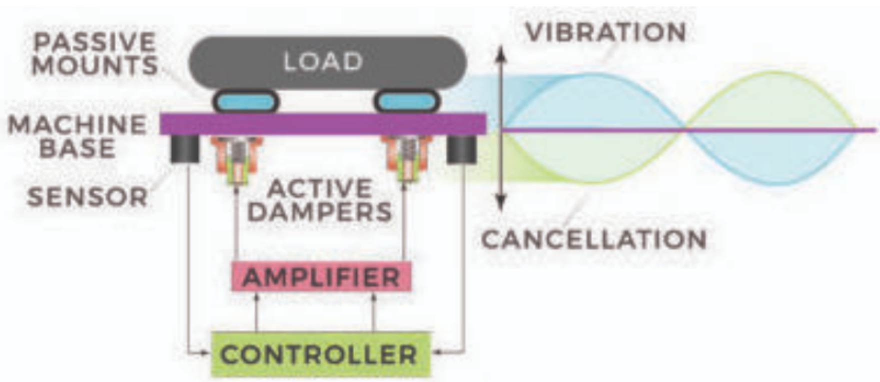

Figure 16: Active damping in a closed-loop system.[31]

2.3.6 Active Damping - Feedforward Control

Another possible setup for an active damping system is by trying to achieve a proactive damping response. While the feedback control structure acts on passed information in order to generate inverse forces to cancel the system vibrations, a feedforward control tries to calculate the expected vibrations and proactively generate appropriate control outputs to counter the system oscillations.

2.4 Actuators

In order to realise an active system, described in section 2.3, some kind of actuator is needed. This section presents some feasible actuators that can be used to provide force/torque to the this particular system.

2.4.1 DC-Motor

A DC motor converts direct current electrical energy into mechanical energy. There are two commonly used configurations of the DC motor. These are the brushed DC motor and the brushless DC motor. As the name implies, the brushed DC motor uses internal mechanical commutation (brushes) to generate torque. The brushless motor on the other hand requires some kind of electronic driver to convert DC current into AC power. The construction of the brushless motor is not the same as a regular DC motor, it is more like a synchronous motor but with no need for synchronisation with an external power supply. Advantages and disadvantages with each motor type can be seen in table 2 bellow [35].

Table 2: Brushed and Brushless Motor: Advantages and Disadvantages [35]

	Brushed motor	**Brushless motor**
Lifetime	Short	Long
Speed and Acceleration	Medium	High
Efficiency	Medium	High
Electrical Noise	Noisy	Quiet
Acoustic Noise & Torque Ripple	Poor	Medium or good
Cost	Lowest	Medium

2.4.2 Servo Motor

A servo motor is used for precise control of angular position, velocity or acceleration [36]. The system consists of a rotary or linear actuator which is combined in a closed loop with a sensor for position feedback. Due to this control circuit, the servo motor can rotate with great precision. This makes servo motors a great component in applications such as Robotics.

2.4.3 Stepper Motor

A stepper motor is an electric motor that divides the shaft rotation into a number of discrete angle steps [37]. These steps makes it possible to control the motor position without having a feedback position sensor, the position is known by simply counting the number of steps that have been performed. In order to run the shaft of stepper motor, the motor coils need to be energised in a specific sequence. This is done using a controller which often consists of a transistor bridge, a pre-driver and a microcontroller unit. Since stepper motors move in precise repeatable steps, they are often used in applications that require precise positioning, such as 3D printers and robot arms.

Advantages:

- Does not require a position sensor
- Does not require a complex control algorithm to work properly
- High torque at low speed
- Good for holding position
- Tends to have a long lifespan

Disadvantages:

- Can miss a step if the load torque is too high
- Always uses the maximum current
- Low torque and becomes noisy at high speed

2.4.4 Hydraulic Cylinders

Another way to generate a force is to use a hydraulic cylinder. A hydraulic cylinder is a mechanical actuator that can apply linear motion and force, by transferring the pressure from a fluid to the point of operation [38]. Hydraulic cylinders are used in both industrial and mobile applications. Compared to pneumatic, mechanical or electric systems, hydraulics are simple, more durable and very powerful. In order to control a hydraulic cylinder the system requires at least a pump, some sort of fluid and a fluid reservoir. The force, F, that the cylinder can provide is limited by the pressure of the pump and is determined from Pascal's law

$$F = pA, \tag{16}$$

where p is the pressure acting on the cylinder piston area, A.

2.5 Transmission

Transmission lines are utilised in a large variety of assemblies where power has to be transferred from one shaft to another. In this chapter a few relevant transmission technologies are described.

2.5.1 Pulley Based Continuously Variable Transmission

A CVT, is a transmission that seamlessly changes gear-ratio within a set span of gear-ratios, without discrete jumps.

The pulley based CVT utilises a V-belt connecting two variable width pulley assemblies of two cones pointing at each other. The variability in width comes from one of the cones, in each of the pulleys, being movable in the axial direction, moving towards or from the static cone [39]. In figure 17 two width setups of a pulley based CVT, resulting in different gear-ratios, can be seen.

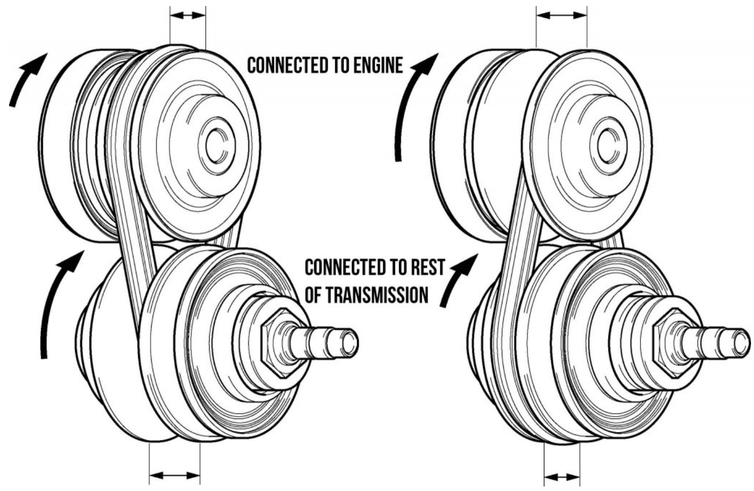

Figure 17: Pulley based CVT.[40]

To move the movable cone axially there are different methods to use. The one explored in the article by [39], called SAPA CVT or Single Acting Pulley Actuator Continuously Variable Transmission seen in figure 18, utilises a servomotor as actuator to move the cone. The cone moving assembly is fixed to the axis, but only axially and not by rotation. This means the pulley is free to rotate independently from the cone moving assembly, achieving control of the gear-ratio of the system.

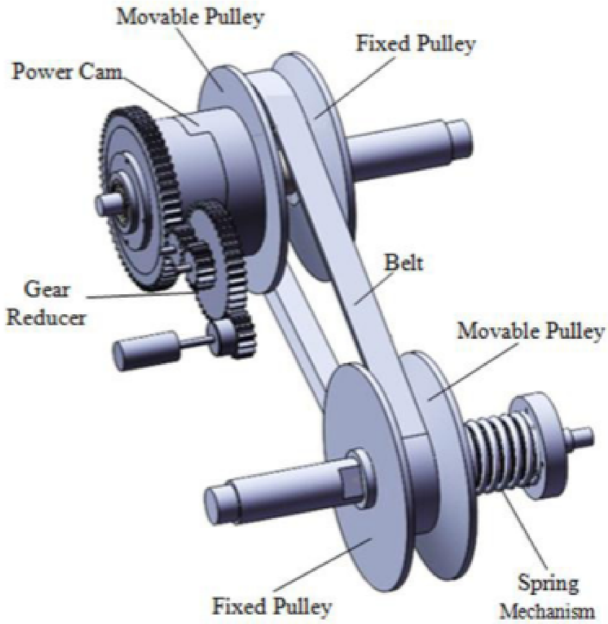

Figure 18: Single Acting Pulley Actuator Continuously Variable Transmission system.[39]

If the belt has a fixed length and rotates without slip, the relationship between the running speed, running radii and transferred torque of each of the pulleys is

$$\frac{\omega_{in}}{\omega_{out}} = \frac{R_{out}}{R_{in}} = \frac{T_{in}}{T_{out}} \tag{17}$$

where ω_{in} is the speed of the the driving pulley, ω_{out} is the speed of the driven pulley, R_{in} is the running radii of the belt on the driving pulley's side, R_{out} is the running radii of the belt on the driven pulley's side, T_{in} is the transferred torque on the driving pulley's side and T_{out} is the transferred torque on the driven pulley's side. [41]

As with every friction based (belt) transmission, this system is prone to slipping if not configured and utilised correctly. In systems with a lot of jerk, this is extra important to take into consideration. [42]

2.5.2 Toroidal Continuously Variable Transmission

A toroidal CVT is also a friction based transmission. However, the toroidal CVT does not depend upon a drive belt and pulleys for its continuous change of gear ratio. Instead, one output and one input disk are connected via a set of power-rollers, as illustrated in figure 19.

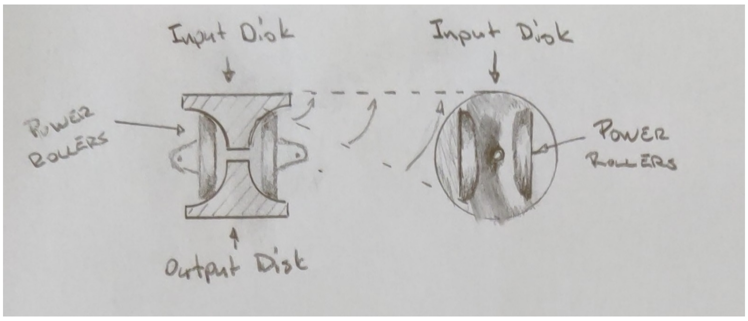

Figure 19: Toroidal Continuous Variable Transmission with pivoting displacement.

In order for a toroidal CVT to change gear ratio, the power-rollers can either pivot or move up and down to alter their contact points to the input and output disks. If they are arranged in a pivoting fashion as seen in figure 19, the contact points to the interface disks change as a direct result of that pivoting displacement. This would result in a wider contact circle for one of the interface disks and a narrower contact circle for the other interface disk, achieving a changed gear ratio. If the power-rollers instead are arranged in a linear displacement fashion, as illustrated in 20, the contact point will change vertically, resulting in a changed force vector direction, which in turn will pivot the power-rollers. This would result in the same type of contact circle relationship change between the two interface disks, achieving a change in gear ratio.

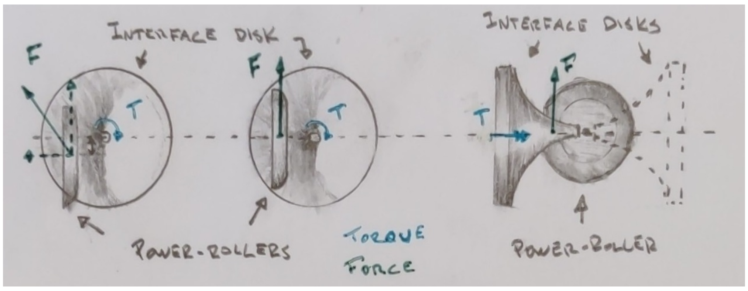

Figure 20: Toroidal Continuous Variable Transmission with vertical displacement.

I.e. the controlling displacement varies between the two different methods, but the basic principle with changing the gear ratio via a change in contact circle relationships stays the same.

2.5.3 Belt Drives

The working principle of a belt drives is to transmit energy from a driving shaft to a driven shaft and this is done by either a friction based interface or mechanical interface between the belt and the pulley mounted on the shaft. The two shafts can be widely separated and does not need to be parallel. There is mainly three types of belts, as can be seen in figure 21:

- Flat belts

- V-belts
- Synchronous belts

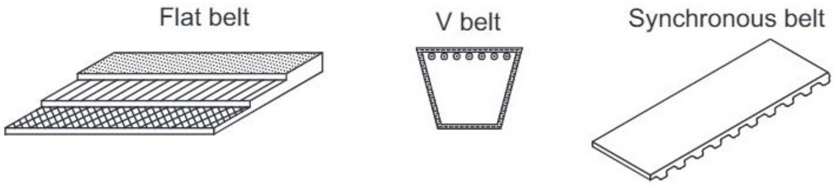

Figure 21: Cross sections of the different belts.[43]

Flat belts rely on friction to transmit the energy from the pulley to the belt and then to the other pulley. The belts are made of reinforced rubberised fabric to give them high strength and much friction in the interface with the pulley. Due to the wide contact line with the pulley, compared to other belts, and high friction there is no need for high tension in the belt, leading to lower loads on the shaft axles and bearing. Another advantage which comes with the geometry is that it has nearly 99% efficiency [44]. This is because of the thin profile that gives lower bending losses. A drawback with the flat belts is that the pulleys needs to be precisely planar aligned to prevent the belt from slipping off.

V-belts are, just as flat belts, friction based. Compared to the flat belt, the v-belt can transfer higher torque due to the bigger cross section area , seen in figure 21, and because the v-shape makes the belt wedge down into the groove in the pulley leading to higher friction. The v-shape makes it also possible for the belt to self align with the pulley so the planar alignment is not as crucial as with the flat belts.[43]

Synchronous belts rely on a mechanical interface to transmit the torque from the pulley to the belt. The mechanical interface is made up out of teeth on both the belt and the pulleys and hence ensures synchronisation between the shafts without any slipping. This is the reason why synchronous belts are used in application where high accuracy in timing is required, for example driving the camshafts in an engine.

In the table 3 below a comparison is presented between the different belts.

Table 3: Table for comparison of belts performance.[43]

Parameter	Flat belt drive	V belt drive	Synchronous belt drive
Optimum efficiency	98%	80%	98%
Maximum speed (m/s)	70	30	50
Minimum pulley diameter (mm)	40	67	16
Maximum speed ratio	20	7	9
Optimum tension ratio	2.5	5	–

2.6 Variable Inertia

In systems there are often needs to regulate the inertia, both rotational and linear. This section delves into some technologies that allow for such control.

2.6.1 Inerter

An inerter is one of the components used to model physical systems within control theory. In a model of a system, the inertia needed can be achieved by just increasing the simulated mass. However, in a real life system it could be desirable to achieve a greater inertia than the mass of an assembly can account for on its own. This can be realised with a flywheel whose rotational motion is tied to the linear motion of the assembly, meaning that if the assembly is to move linearly, the flywheel needs to be rotated. An example of this can be seen in figure 22 where the rack pushes the assembly forward linearly at the same time as it rotates the flywheel through a gear set.

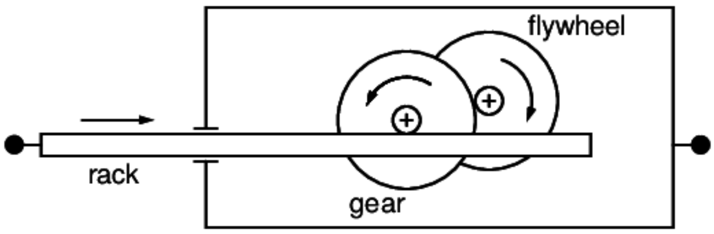

Figure 22: Realisation of a simple inerter in a real life system [45].

This transfers the moment of inertia of the flywheel to the linear inertia of the assembly, and as moment of inertia is more versatile (different configuration with the same mass can achieve different moments of inertia) this gives more control over the linear inertia to the system designer.

2.6.2 Radially Moving Mass

Another way of achieving a variable inertia is just to consider the basic theory behind rotational inertia,

$$I = mr^2, \tag{18}$$

where I is the rotational inertia, m is the point mass at the rotational radius r, one can see that the inertia is variable by simply moving the point mass radially [46].

2.7 Sensors

This section covers theory about all sensors that are necessary for this project. To be able to choose the right sensors, different options will be discussed. The sensors will be used to measure force, torque and motion. This section will also cover theory about Dewe 43A and Dewesoft that will be used to analyze the data from the sensors.

2.7.1 Dewe 43A and Dewesoft

The Dewe 43A that can be seen in figure 23 is a product designed by Dewesoft to allow the user to collect data with high precision and that is synchronised with time. The company also provides a free software called DEWESoft X3 that can be be used to visualise and analyse the data. The Dewe 43A has 8 analogue input channels that can measure voltages with 24 bit resolution. It also has 8 encoder inputs and 2 CAN ports.

An important feature with the Dewe 43A is that it can be used to amplify signals. This is important since the signals from all the sensors can be very small.

Another aspect with the Dewe 43A is that it is not designed to provide data in real time. Because of this it is not suitable to use it when implementing real time applications like active damping. For that a microcontroller need to be used instead to receive and transmit signals. In that case extra amplifiers need to be used to scale the signals so it corresponds to the microcontrollers reference voltage.[47]

Figure 23: Dewe 43A board.[48]

2.7.2 Inline Rotary Torque Transducer

An inline rotary torque transducer (IRTT) is used to measure torque on a rotating shaft. In this project the sensor will measure the torque between the tool and the bolt.

To accomplish this an inline transducer sleeve will be used to hold the IRTT. Figure 24 below shows a 3D printed inline transducer sleeve that was build for and designed by Adam Klotblixt and Sofia Olsson. A similar component will be used in this project.

The motor torque could also be measured and used to calculate the torque on the bolt, but since there is friction in the gears it would not be as accurate as directly measuring the torque between the tool and the bolt. Also it is useful to measure the inline torque since it is related to the reaction force on the operator that also is researched in this project.[49]

Figure 24: Inline transducer sleeve designed by Adam Klotblixt and Sofia Olsson.[49]

2.7.3 Accelerometer

An accelerometer is a device that can measure vibrations or acceleration of a motion. A commonly used type of accelerometer is piezoelectric which produces an electrical charge that is proportional to the force applied to it. Since the acceleration is also proportional to the force and the mass of the piezoelectric material is known, the acceleration can be calculated.[50]

Another type is the mechanical accelerometer. The basic principle is the same as a passenger sitting in a car, but instead it uses a mass connected to a spring. When it accelerates, the deformation of the spring directly corresponds to the acceleration.

A third type is called capacitive accelerometer. Similar to the mechanical accelerometer it has a mass connected to a spring, but there is also a capacitor that consists of two plates. When the mass moves it changes the distance between the plates. This also changes the capacitance which can be measured and used to calculate the acceleration.[51]

2.7.4 Gyroscope

A Gyroscope is a device that is mounted to a frame and is able to sense the angular velocity if the frame is rotating. Two examples of different types of gyroscopes are mechanical and optical gyroscopes.

Mechanical gyroscopes relies on the gyroscopic effect which means that a spinning mass tends to keep its angular position in relative to an inertial frame. The spinning mass is often attached to gimbals that can rotate freely. The gimbals are also attached to an inertial frame which often consists of a type of vehicle, but in this project it is a nutrunner. The angular velocity can then be calculated since it is the product of the spinning mass and the torque acting on it.

Another type is the optical gyroscope. It does instead rely on the Sagnac effect which is a phenomena that has been derived from the general relativity theory. It states that two counter-propagating beams propagating in a ring structure will change phase relative to each other if the ring structure is rotating. This principle can be used in gyroscopes to calculate the angular velocity.[52]

2.7.5 Force Sensor

To be able to measure the reaction force from the operator a force sensor is needed. Force cannot be measured directly, but there are different methods for measuring it indirectly for example by deformation. These different types of methods will be described below.

Load cells is a force transducer and converts a force/pressure to an electric signal. Examples of different types of load cells are pneumatic, inductive and hydraulic load cells and they are typically used to measure weight.

Another type of force sensor is a strain gauge. Depending on the load the resistance of the sensor varies. This affects the output signal and can be used to calculate the applied force. Sometimes several stain gauges are combined in the same sensor to allow multi axis measurement.[53]

The third type of force sensor is called force-sensing resistors and is a type of piezo resistive sensing technology. It consist of semi-conductive material/ink that is sandwiched between two different substrates that are separated. When a force is applied the ink will deform which also causes the resistance to drop. In this way the resistance becomes inversely proportional to the applied force.[54]

2.7.6 Rotary Encoder

Rotary encoders are often used in industrial machines to track the position of a rotor shaft. The encoders can generate digital information about position and motion.

There are different types of rotary encoders and some of the most important types will be described below.

Encoders are often divided into two types: incremental and absolute encoders. Incremental encoders are often used in low budget applications and or when only the relative position is needed. A typical application for an incremental encoder is with an AC motor. Absolute encoders returns the exact position and is typically used with servo motors.

Absolute optical rotary encoder that can be seen in figure 25 uses a disc with a pattern. Together with LEDs and photosensors it can generate an unique value for each position on the disc. To accomplish this Gray code is often used.

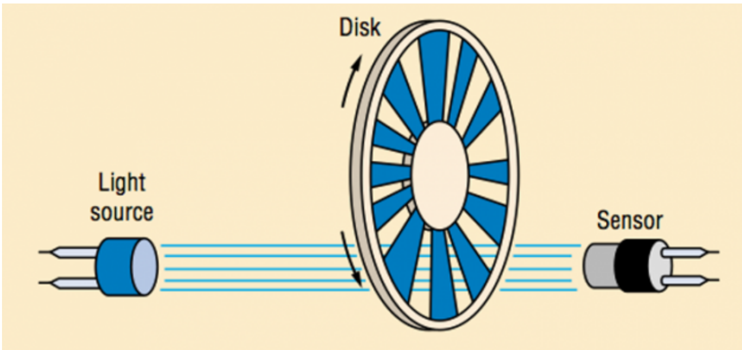

Figure 25: Ilustration of an optical rotary encoder.[55]

Another type of encoder is rotary magnetic encoder. It is generally less fragile and more robust to vibrations and high temperatures than the optical rotary encoder. It often uses a magnetic disc together with hall effect sensors to calculate the rotary motion. When the disc rotates it causes changes in the magnetic filed which is registered as pulses by the hall effect sensors. A magnetic rotary encoder can be seen in figure 26.[56]

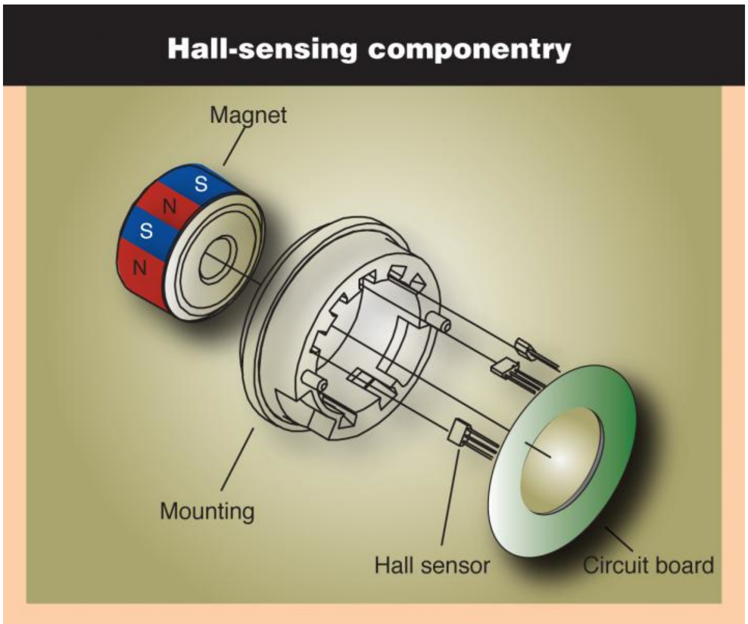

Figure 26: Magnetic rotary encoder.[57]

2.8 Bearings

Bearings are used to reduce friction in both rotating and linear motions. This section presents common types of bearings and their characteristics.

2.8.1 Rotary motion

The most common ball bearing type is the deep groove ball bearing. It has grooves in the races for the balls to roll in, see figure 27. Deep groove bearings have low friction, noise and vibrations, and as such they work well in high speed applications. As they are very common they are cheap and accessible.[58]

Angular contact ball bearings are similar to deep groove bearings, but can tolerate higher axial loads because of larger contact area in the axial direction, see figure 27.[58]

Thrust bearings can sustain high axial loads, but no radial load. The thrust bearing can seen in figure 27.[58]

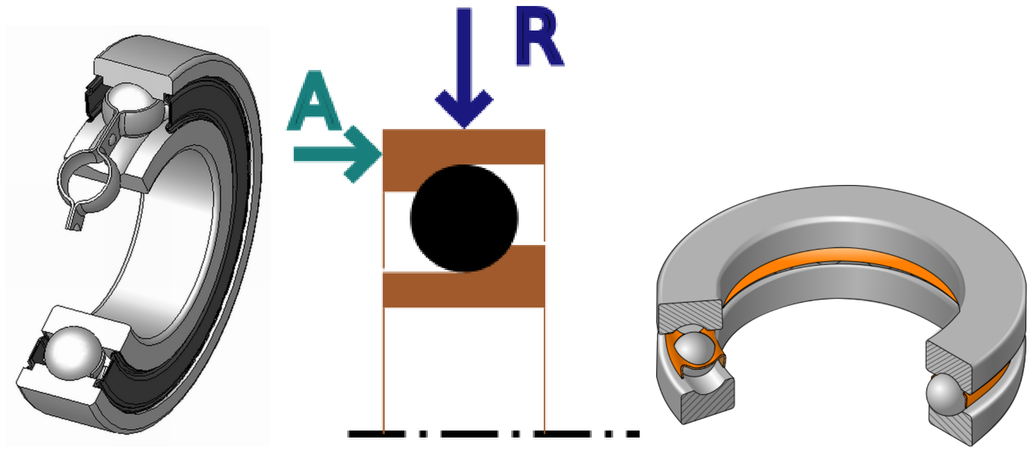

Figure 27: Different ball bearing types. Left: a deep groove ball bearing[59], middle: section view of an angular contact bearing[60], right: a thrust ball bearing[61].

Roller bearings are bearings with rollers (cylinders) instead of balls. As the rollers have much larger contact area compared to a ball, they can sustain substantially higher loads compared to ball bearings. However they have low axial load capacities.[58]

There are many more types of bearings, and it can be hard to choose which bearing type is best suited for a specific application. One can use the comparison table presented in figure 28 to better get an understanding of when different bearing types are used.

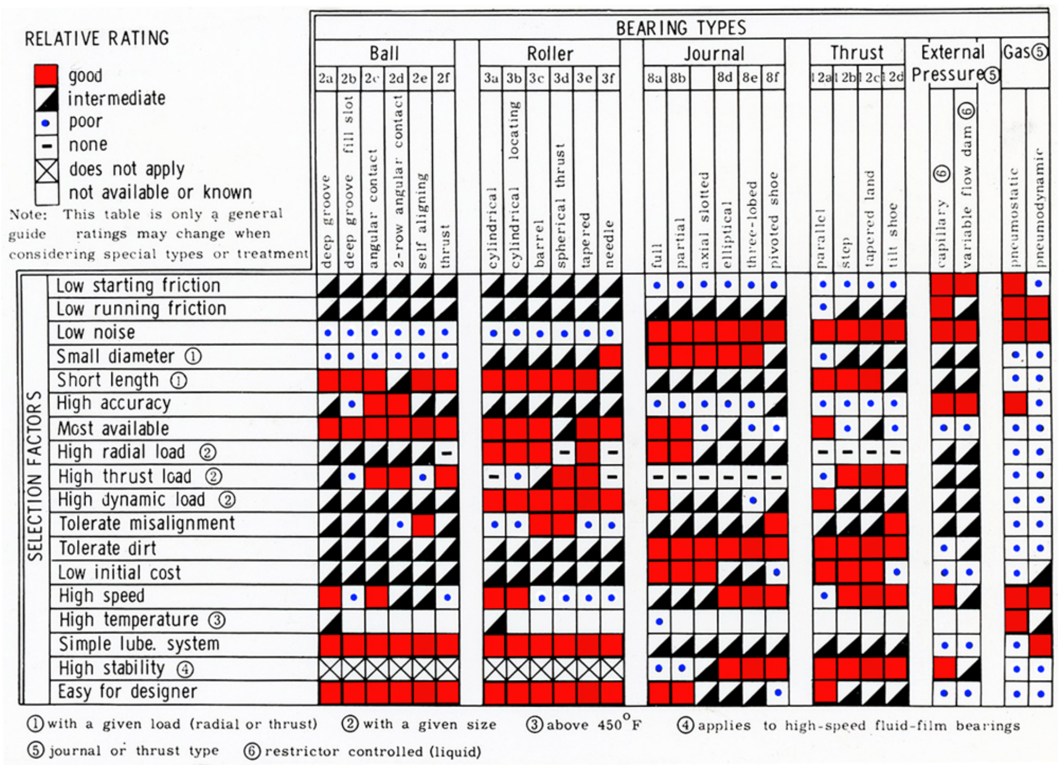

Figure 28: Comparison table for different types of bearings.[62]

2.8.2 Linear motion

There are mainly two types of linear motion products: linear bushings and linear guides. The two can be seen in figure 29.

Linear bushings are relatively cheap and easy to use. They are self-aligning, meaning they will not bind up if the rails are not completely straight.[63] They run along a round shaft and can spin on the shaft. It is therefore necessary to have to shafts in parallel, in order to restrain the carriage to one degree of freedom. The maintenance requirements are low. Accuracy and tolerance requirements are not high, and the shafts can bend and result in a large deflection.[64]

Linear guides are more expensive and require flat surfaces for installation. Because the rails are mounted to a surface, they can support much larger forces compared to linear bushings. The rails have grooves that allow for more contact surface with the balls. A linear guide with a width of 25 mm can support up to 1000 kg. The guides are very accurate and have errors in the µm-range.[64] Good maintenance and mounting is required for smooth operation.[65][66]

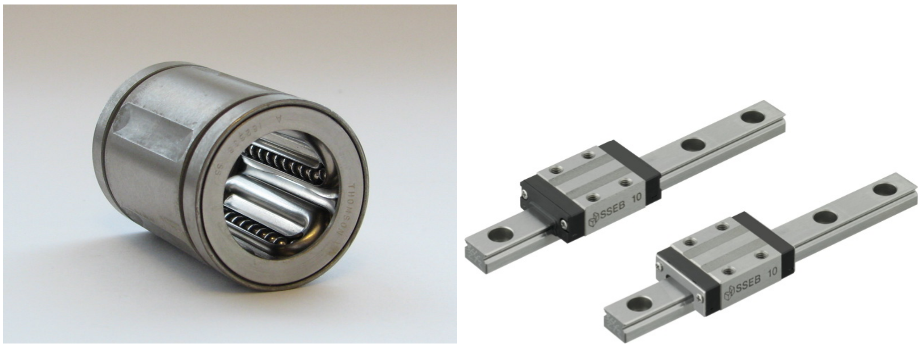

Figure 29: A linear bushing [67] (left) and a linear guide [64] (right).

2.9 Existing Test Rigs

S. Mukherji constructed a test rig for a right angle nutrunner (manufactured by company Stanley, but the tool is similar to Atlas Copco's).[23] The opertor is modelled as a spring-mass-damper system. A pneumatic cylinder was used as a semi-active spring. The mass could be varied using small removable weights in a cart. The test rig could handle torques of 60 Nm at the nut and the tests were highly repeatable. However, the rig did not contain a damping element, and therefore had some oscillations. The rig is shown in figure 30.

H. Ay et al. constructed a similar rig to Mukherji.[68] The rig also had a pneumatic cylinder used as a semi-active spring, and a variable mass using weights. The damping was not adjustable, but the inherent damping of the pneumatic cylinder was found to be close to the human arm model. The system was automated and could with high accuracy simulate a human operator.

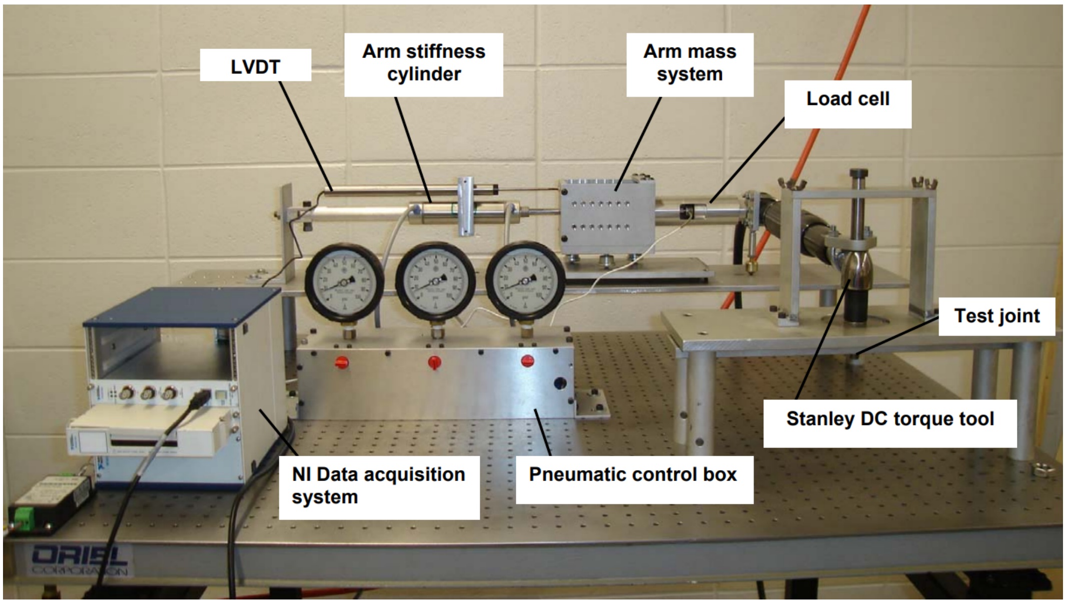

Figure 30: Pneumatic test rig constructed by S. Mukherji.[23, p. 53]

3 Concept Design

All the information presented in the previous chapter, State of the Art, was resorted to in the following phase of the project, the conceptualisation. Three concepts were developed and are described below.

3.1 Concept 1

Concept one revolves around the central idea of the test rig following the tool. This is done by fastening the tool to a rotating beam with a rotational axis aligned with that of the tool, as seen in the side view of the rig in figure 31. The fastening point at the handle of the tool represents an operator hand holding the tool firmly, and will therefore also include a force sensor in order to record the reaction forces experienced by the simulated operator. On top of the rotating beam a stepper motor is attached with a lead screw that can move a weight radially with the central axis of rotation. At the head of the tool, only a cup representing an operators resting hand holds the tool in place.

This cup is in turn connected to a shaft that also holds the rotating beam. Hence, it is also rotationally aligned with the tool. Centrally on this shaft, a belt drive connects it to an electric motor, as can be seen in the front view of the rig in figure 31. Further up the shaft a torsional spring is attached. This torsional spring is grounded in the rig's frame, resulting in the shaft being grounded through the spring.

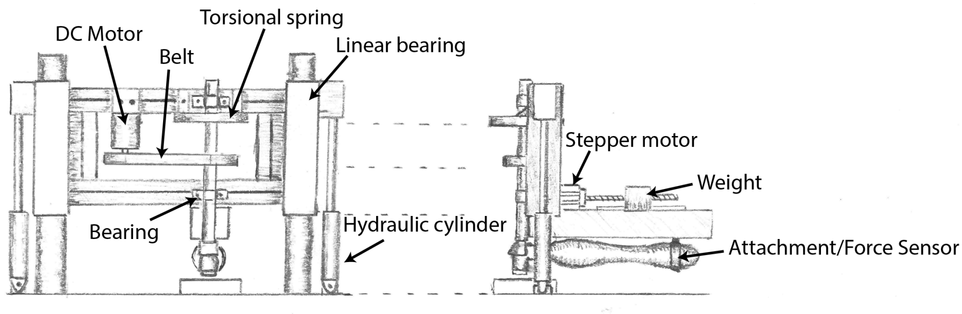

Figure 31: Concept 1 drawing with front view, left, and side view, right.

The frame itself is set on two linear bearings that are both firmly connected to an independent work surface, which can also be observed in the front view of figure 31. These linear bearings would mean that the entire weight of the moving assembly (the electric motor, the belt, the torsional spring, the shaft, the cup, the rotating beam and all their common frame) would rest on the tool. However, also attached to the frame of the movable assembly, parallel to the linear bearings, are two helical screws with stepper motors holding it in place.

All these concept choices are in response of the different requirements from the stakeholders, presented in subsection 1.3.1. These responses, and short evaluations of the different options at hand are presented below where the different subsystems are presented.

3.1.1 General Model

One of the first decisions that was made regarded the model to be used to simulate the human operator. As mentioned in the SOTA in section 2.1 there are two main models used to simulate human arms. The mass-spring-damper model and the Hill model. And even though the Hill model is somewhat more precise, the fact that it is very difficult to realise in an actual implementation, resulted in the choice of the mass-spring-damper model.

The mass-spring-damper model has a very simple realisation available, the passive implementation described in 2.3.1. However, as mentioned, this implementation has a very limited agency with only being able to dissipate energy. This would mean that an entire tightening of a nut would have the same reaction from the rig. It is quite intuitive to imagine an actual human operator not reacting this way. They would first take some time to react to the started tightening to then tense up their muscles, and in doing so, change the characteristics of their arm system. This implies this is something desirable to achieve with the test rig, to fulfil the stakeholder requirement of being able to accurately simulate human operators.

To change the operator model actively during a tightening requires an entirely active damping assembly. With that determined there are still a couple of different options presented in section 2.3. Of these, the one that seems the most versatile, and easy to implement, would be the active damping with an electric motor from section 2.3.5 and 2.4. This option also goes well in line with the model chosen, granted the slight modification presented in figure 32.

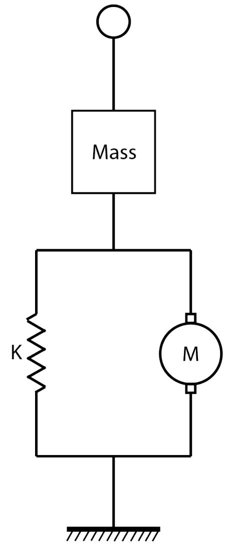

Figure 32: Model used in concept 1 to try and simulate a human operator. Realised with an electric motor (M) and rotational spring (K).

In this realisation, the electric motor would be controlled to simulate a damper and spring, assisted by the passive spring also in place. This active component of the system would allow for a change in simulated operator characteristics during a tightening to achieve a greater accuracy.

3.1.2 Belt Transmission and Torsional Spring

To transfer the control output from the electric motor to the shaft, some sort of transmission was required. Some obvious choices were gears, chains and belt drives. The choice made here was made mostly based on the inertia each transmission would add to the rotating assembly. This as the total inertia of the rotating assembly is constrained by an actual human operator's arm's inertia. And as it is not quite clear how large the rest of the rotating assembly's inertia would be, it was desirable not to add too many high inertia components. Therefore the choice of a belt drive was made quite quickly, as this would add only a small inertia to the subsystem.

One concern with a belt drive was whether or not the its elasticity would be a problem. However, if this would arise as a problem in the future, the option of switching to a chain drive, or just compensating for the elasticity within the control of the electric motor, was assuring enough to add the belt drive as transmission in concept 1.

3.1.3 Stepper Motor with Movable Mass

One of the key aspects of the test rig being designed is that it should allow for different operators being simulated. This incurs the requirement that the inertia of the test rig should be variable, as different operators would have differently massive arms holding the tool. This could be achieved in a couple different ways, as was discovered during the SOTA. One would be to utilise an inerter paired with a continuously variable transmission, as can be read more about in section 2.

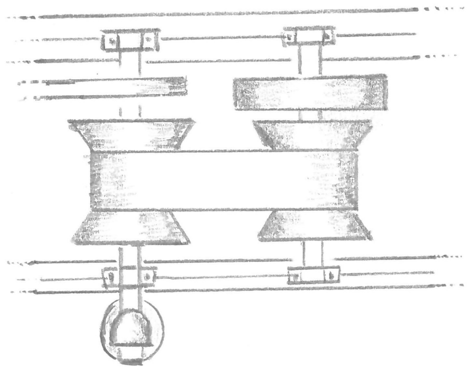

Figure 33: Part of concept 1 with the addition of a belt based continuously variable transmission and reaction wheel as inerter connected to the central shaft.

One could imagine, for example, a belt based CVT connecting an inerter to the shaft, similarly to how to electric motor is connected to the shaft as can be seen in figure 33. However, this was decided against as the much simpler solution of pushing a mass in and out radially, and in doing so changing the inertia of the rotating assembly, was thought of. One drawback of choosing the linearly moving mass is that the range of inertias the rig can reach is somewhat narrower. A CVT can have very high gearing ratios, allowing the inerter to have a very great effect on the inertia of the rotating assembly, while the linearly moving mass only increases the inertia by a factor of the difference in rotational radius squared.

Both these options also fulfilled the requirement of the test rig should be adaptable for different tools of different lengths. If the tool handle fastening is radially further out on the rotating beam, the simulated operator's arm's inertia would be greater, which is something both these solutions would fulfil.

3.1.4 Linear Bushings and Stepper Motors

One of the more complicated requirements to find a reasonable solution for was the specification that the rig should compensate for the vertical movement of the tool as it tightens a nut. In concept 1 this was solved by mounting the entire rotating assembly on a linearly moving frame that is attached via linear bushings to rods that allow for motion vertically. This choice was weighed against going for linear rails as presented in section 2.8.2. However, the linear rails solution was deemed somewhat less robust, which is a very import factor in the test rig as it will experience extreme cases of nut tightenings with torques.

To control this vertical movement, both hydraulic cylinders and electric stepper motors with helical screws were considered. However, to not add too much mechanical complexity, and to avoid the introduction of hydraulics into the system, the choice was made to continue with stepper motors and helical screws.

Another advantage of using stepper motors with helical screws was that they can achieve a larger span of motion than its counterpart.

3.2 Concept 2

The basic idea of the second concept is that the test rig would visually mimic a human operator. The concept is mainly to control and absorb reaction forces with the help of a mass-spring-damper system, in which a pneumatic cylinder is used to actively regulate the various forces that arise when tightening the screw. A mass-spring-damper system is used due to its simplicity, while still being sufficient to achieve the goal of the project (read more about it in 2.3.1). The tool is mounted in a bracket on a kind of carriage that runs along a circular rail, to make it easier for the tool to move sideways. The mass-spring-damper system is also mounted onto the same carriage. The bracket in which the tool is fixed is in turn mounted on a linear, vertical rail. This allows the tool to move in a vertical direction when tightening and loosening screws in beam joints. A principled drawing of concept 2 can be seen in figure 34 and a closer look of the mounting of the tool to the test rig can be seen in figure 35.

According to project requirements, the test rig must take into account that the tool moves relatively much vertically as well as horizontally. An adjustable spring constant on the mass-spring-damper system can also enable the test rig to simulate different human operators, and therefore it is also important that the response time of the system on the test rig can correspond to the reaction time of a human operator. It is also important that the test rig can repeat tests with specific parameters and then get results that can be analysed and compared against other tests performed on the rig. Another requirement for the test rig is that it must be able to handle 100 Nm (extreme case). All these requirements have been taken into account when this concept has been developed, but with varying degrees of success, for example, it can be difficult to know whether the concept so early in the process will withstand 100 Nm (in addition to the fact that the test rig will be robust).

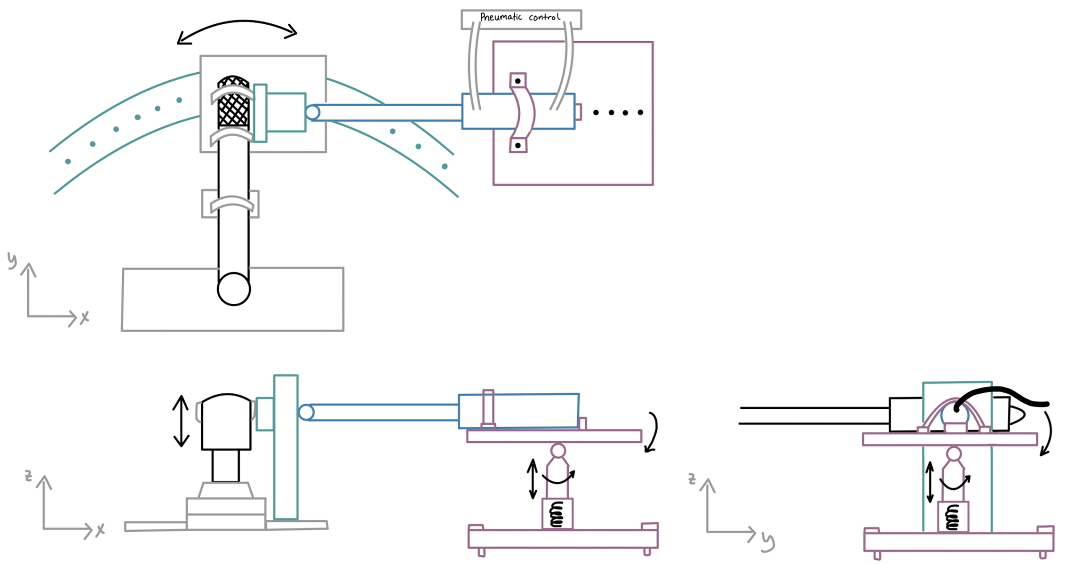

Figure 34: Concept 2 drawing with front view (xz-plane), side view (yz-plane) and top view (xy-plane).

The pneumatic cylinder can (if chosen, set and regulated correctly) make the system act relatively similar to a human arm. The idea is that the pneumatic cylinder should have an equivalent to an adjustable damping constant, which would make the test rig semi-active as well as being able to preset parameters, and then let the cylinder regulate accordingly. This idea was a result of the subsection 2.2 of the SOTA about damping systems, with special focus on active damping from the subsection 2.2.3.

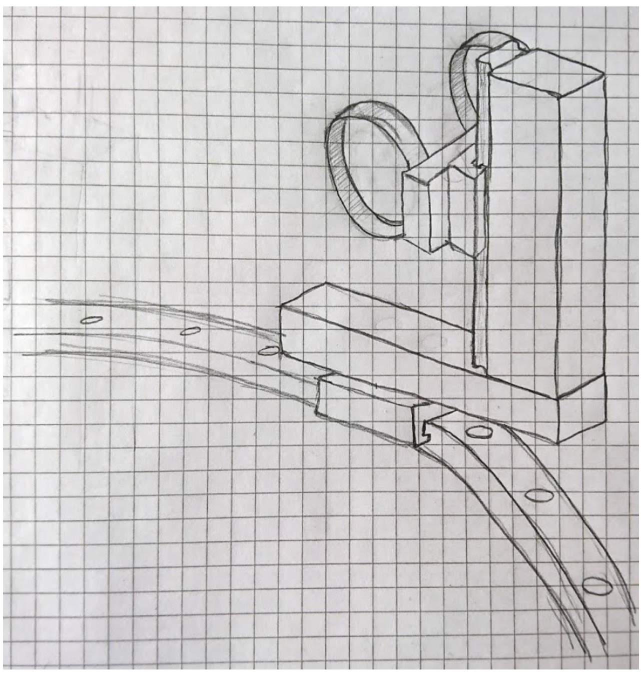

Figure 35: the attachment of the tool in concept 2.

An active system would certainly be the most favourable damping system, but with this solution (with a pneumatic cylinder) it did not seem very feasible. The problem is that a pneumatic cylinder becomes quite complex if it is to be controlled by active regulation. It is also required that the pneumatic cylinder can react quickly enough, to ensure that the system truly acts as supposed. To use a pneumatic system can be difficult as the rise-time for the step response of a pneumatic control system may be too slow for its purpose in this project, and therefore a concept with a pneumatic cylinder might only be feasible as a semi-active system.

3.3 Concept 3

The starting point when concept three was developed was to look at the already existing operator test benches discussed in section 2.9. They do all have a similar fundamental functionality which have proven to be sufficiently accurate when simulating a operator during a tightening procedure. Hence the thought was to get inspiration from those rigs and incorporate new state of the art technology so that the test bench would meet the requirements set by the stakeholders.

The key feature of the concept is the belt drive system actuated by an electric motor. In the figure 36 below the belt drive system includes the motor, gearbox, toothed belt and the carrier. The electric motor was thought to be modelled so that the dynamics of the system mimics the dynamics of a passive mass spring damper system as described in 2.3.5. This allows for full adjustment of the system parameters and also to fine tune the system to be as accurate as possible as a real operator, just as one of the requirements states. Since the requirements also states that the test rig must be able to test extreme cases with torques up to 100 Nm the position of the contact point between the nutrunner and actuator is vital for the motor and its performance. The further away from the axis of ration the smaller force is needed to prevent the motion. That is why the belt system is connected as far back at the nutrunner as possible, just like a human operator would hold the tool.

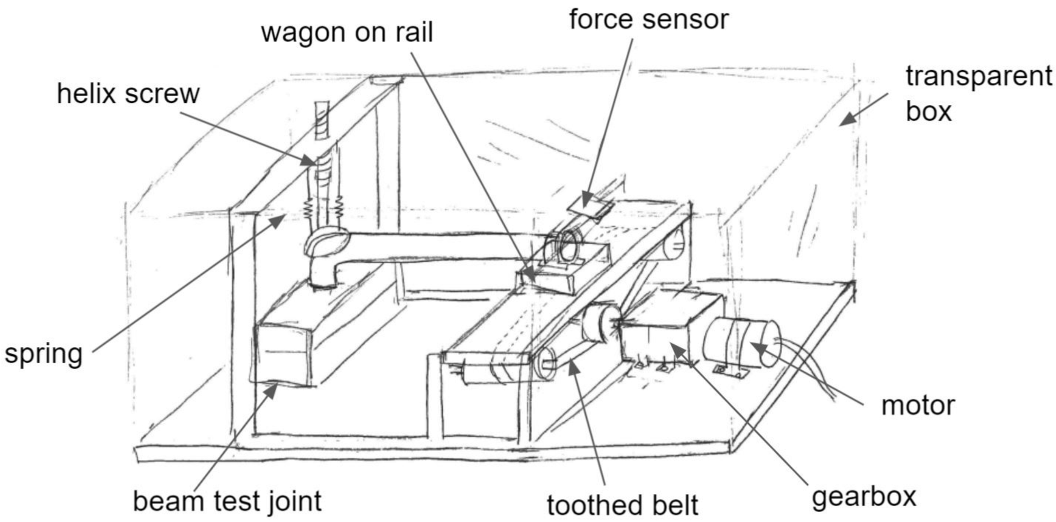

Figure 36: Overview of concept 3 with all important features marked out.

One of the benefits of using a belt, in this configuration, to transfer the rotational motion of the motor into linear motion of the tool is that the placement of the motor is flexible. The motor can, instead of being mounted on the side, be placed underneath. This fosters a more compact design which is both easier to be moved around and easier to encapsulate with safety glass to prevent injuries.

A linear rail guide was chosen to be used as a support structure where the actuation and force sensing takes place. By mounting the tool to this linear rail guide via a carrier the the nutrunner can move friction less from side to side. The motion is, however, quite limited since the nutrunner handle travels in a circular path. This was solved by having a slot in the carrier and a pin on the tool handle bracket allowing the tool to move in both x-direction and y-direction as pictured in figure 37 below. This solution does not allow the nutrunner to turn 90 degrees but more in the range 10 to 15 degrees. This is not fully in line with the requirements, but the decision was made to do so because a wider motion range will make the construction more complex.

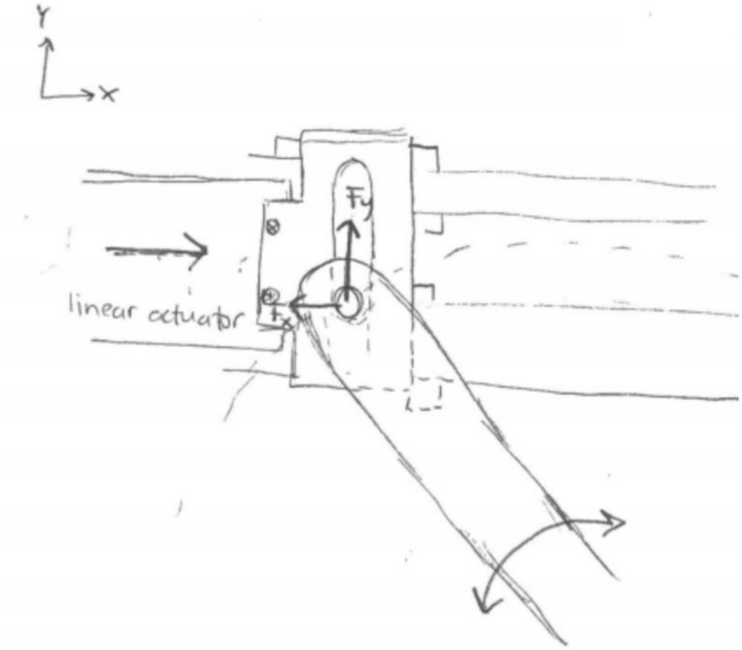

Figure 37: Detailed view of the tool handle and the carrier.

As pictured in figure 36 there is a helix screw and springs pushing down the head of the tool onto the socket and beam test joint. The intention of the helix screw is to stabilise the tool to prevent unwanted vibrations to arise, and the helix screw can be adjusted up and down to adjust for different situations. There is how ever a small gap, so even though the helix screw is fully screwed down the tool head can still move slightly up and down. This is to adjust the height of the tool during a tightening process, and that is also why the springs are added in parallel with the helix screw. The intention with the springs is to keep the tool head under a constant pressure and push down the tool when the nut is tightened, but also allow the tool to rise as the nut is unscrewed. All this motion within the motion gap the helix screw leaves. The motion span is how ever not big enough to support the run down section when the axial motion is much bigger. The motion span is enough for the tightening sequence and that trade-off was made to promote the simplicity of the construction.

In figure 36 the IRTT, mentioned in section 2.7.2, is not pictured but it shall obviously be placed between the nut and the socket. Since no other forces than the downward force and torque will act on the beam test joint, it was thought that a shape-dependent clamping would be sufficient to prevent the beam test joint from spinning when the nut is tightened.

3.4 Evaluation of the Concepts

When the concepts were created, the next phase was to evaluate each concept and select one for further development. First, the concept groups presented their solution to the whole project group and to the stakeholders. After the group had received feedback from the stakeholders, the concepts went into a new small realisation phase where each concept group made simplified calculations and investigated potential components. Later on, the project group listed and compared pros and cons for each concept and then finally, the project group used a weighted evaluation matrix to select one concept. After the concept evaluation, an idea came up of a new concept which was a mix of great parts of the previous concepts. The stakeholders gave great feedback on this mixed concept. At the moment the project group is still in process of selecting one final construction. In order to accomplish this, the project group will need to investigate the realisation of the constructions in more depth.

3.4.1 Pros and Cons for Each Concept

The project group discussed the pros and cons for each concept and these were the main points:

Pros of Concept 1:

+ The height of the rig is movable, which enables the rig to follow the tool down during bolt tightening and to fit different types of beam test joints.
+ The attachment construction makes it easy to change beam test joints and to change the tool.
+ The construction enables the tool to rotate $\approx \pm 90^\circ$ around its rotation axis.

Cons Concept 1:

- To generate torque directly on the tool head requires a strong motor, which can be very expensive, or a high gear ratio, which require a lot of space.
- The control system needs to be precise fast and in order to measure the small angle changes at the tool head and have time to apply enough motor torque.
- The beam that the tool is attached too can be heavy. This can affect the inertia and the results of the tests.
- The concept will require a large gear ratio, which can increase the inertia of the system a lot, as the inertia of the motor is proportional to the gear ratio squared.

Pros Concept 2:

+ Easy to use semi-active system which has been proven to work (section 2.9).
+ With correct settings on the pneumatic cylinder and for small tool angles it can simulate a human arm well.

+ Using a pneumatic cylinder semi-actively is robust for extreme cases, since it does not need be actively controlled and requires no power during test runs.

+ The curved rail enables the tool to rotate several degrees around its rotation axis.

Cons Concept 2:

- Semi-active systems limits the modelling of a human operator. For example, the stiffness and damping can change during rundown which is hard to simulate with pneumatic.

- Large forces will occur on the attachment points on the cylinder, which can affect the robustness.

- Friction forces in the rail might affect the test results, and if the nut is not perfectly centred the linear bearing might bind up.

Pros Concept 3:

+ The placement of the motor enables lower torque requirements on the motor, compared to concept 1. The price of the motor would be lower.

+ A passive system with springs and dampers can be implemented parallel to the active system. This would also decrees the torque requirement on the motor.

Cons Concept 3:

- The tool has limited rotation range since the tool attachment point in the wagon has limited vertical translation space.

- Forces of the components on the attachments point between the tool and the wagon are high, which can affect the robustness.

- As for concept 1, the inertia of the motor will have a large effect on the total system inertia, as there is a high gear ratio.

3.4.2 Stakeholder Feedback

The project group received great feedback on each concept on great aspects and some potential issues from the stakeholders. The stakeholders also gave great input on general points to have in mind for the construction and also some ideas for improving the construction. Here are some of the main feedback points the project group received from the stakeholders:

- The stakeholders liked the different ideas from all the concepts.

- One idea from the stakeholders was to have a permanent passive system parallel to an active system, to decrease the requirements on the active system.

- A suggestion was to use a plate coupling between the passive and the active system and to connect the active system at a specific time.
- One solution might be to regulate the height of the beam test joint and not the rig itself during tightening.
- The tool does not need to start from degree 0 but it can start from example -20 and when rotate to 20 degrees (40 degrees rotation in total).
- In terms of motor requirements, the motor needs to counteract the tools movement by applying a larger torque to stop the tool rotating.
- It was more important to have a robust rig than it would be easy to change tool and to move the rig.
- It was also more important to be able to test different tightening techniques than to test different operators.
- The project group need to have in mind that the rig needs to have space for a tool battery or space for a power cable.
- Concept 1 would probably require a gear ratio since the torque requirement on the motor could be very large. If a gear box would be used it would affect the inertia of the system.
- The group also needs to check that the motor for concept 1 would be fast enough, this with also having the extra inertia in mind.
- For the rails in concept 2 and 3, the project group needs to have the friction in mind.
- A cylinder can effectively damp vibrations.

3.4.3 Evaluation Matrix

An evaluation matrix was used as a method to systematic evaluate each concept against the stakeholder requirements. The evaluation matrix contained the stakeholder requirements where the project group had together put a weight on each requirement of how important the requirement was. The weight number was between 1 to 5, where 5 was extremely important and 1 was desirable but not so important. The project group presented the weights for the stakeholders who agreed on the weights for the requirements. The matrix and the weights for each requirement can be seen in appendix A.

When the matrix was created, the next step was to give each concept points on how well each concept fulfil each requirement. The points were between 1 to 5. If a concept receive 5 points the concept satisfy the requirement really well and if a concept receive 1 point the concept does not satisfy the requirement well. The project members filled the matrix individually to give a wider range of points, since each point was subjectively decided. Appendix A is showing a matrix that was filled by one project member. The total results turned out to be very even between the concepts, but concept 1 got the highest score for most of the project members.

A weighted evaluation matrix can be a great tool to evaluate the concepts against the requirements but weights needs to be based on justified arguments. Since the results were close it could be interesting to try another scale for the concept points. The project group will investigate this idea of trying a new scale and see if it could affect the result.

3.4.4 Combining Concepts Into a New Concept

The group, and Atlas Copco, saw benefits and drawbacks with every concept. To get the best of everything, a new concept was proposed with parts from all concepts; the rotating beam from concept 1, the circular and linear rails from concept 2, and the belt drive system from concept 3. The concept is shown in figure 38.

The aluminium structure can rotate with the tool. It is fastened to a circular rail to keep it centred around the nut. A load cell measures the force between the tool handle and the operator. When the tool rotates it drives a servo. The servo motor has a control system to emulate the operator.

At the moment, the project group is still in the process of selecting one construction for the test rig. The project group will need to investigate the realisation of the each construction in more detail and do more calculations in order to decide one final construction.

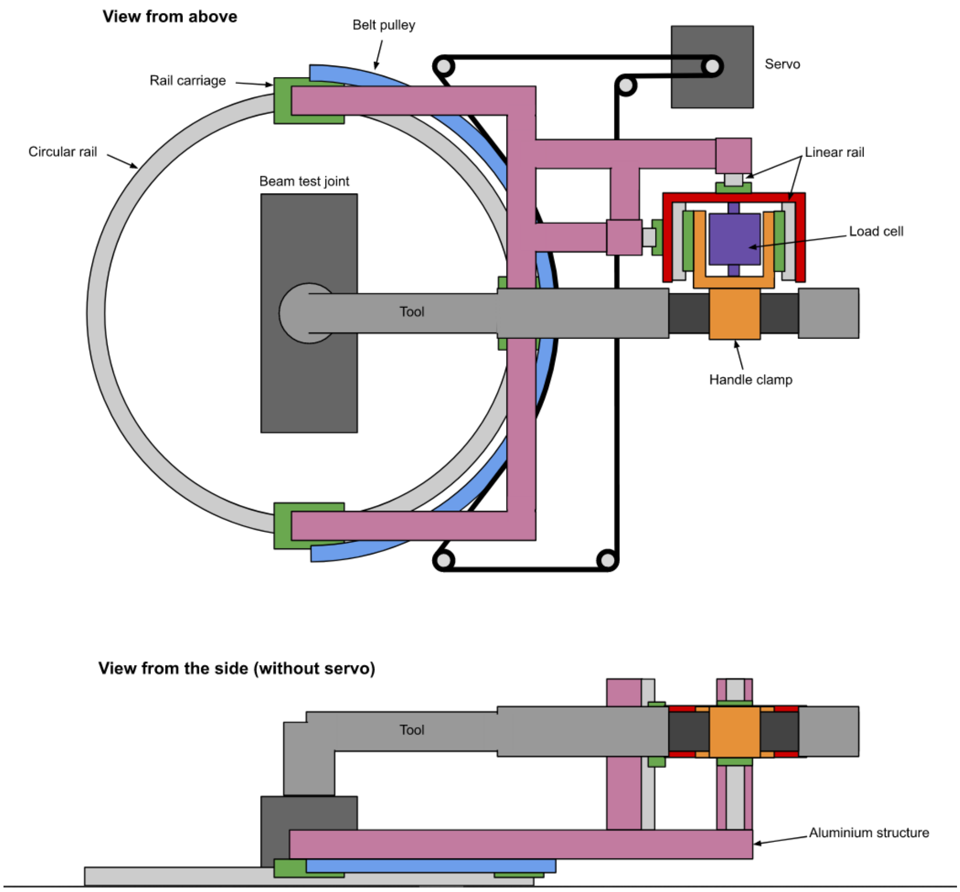

Figure 38: A new concept combining ideas from each old concept.

4 Future Work

This chapter presents the project tasks and the project plan and for the fall. It also consists of a risk analysis that presents the most crucial aspects of this project.

4.1 Assignments

Concepts of relevance will be divided into subsystems, which then will be simulated. The simulations of the subsystems will be compared to analysed test data provided by the stakeholder. This will guide the group with key parameters required to determine which parts that will be ordered for the prototype (motors, belts, beams etc.). Initially, simplified simulations will be made in Simulink. These simulations will tell the group what is feasible and what is not. With help of this, a detailed construction of each subsystem can be made.

The detailed constructions will be drawn in CAD and verifying simulations using tools such as Simulink and Finite Element Analysis (FEA) can be used to verify the constructions before they are built. By building the system using simulation tools, software can be tested even though there is no physical product. Building the final product using subsystems allows for flexibility. If the final simulations show that one subsystem do not fulfil the requirements, the subsystem can easily be changed without changing the whole construction of the product.

The major remaining tasks for the project is presented bellow:

- Divide concepts of relevance into subsystems
- Analyse test data provided by the stakeholder
- Do simulations on each subsystem
- Make up a budget for the project
- Make up a parts list
- Assemble the whole construction in simulation environment
- Write control algorithms
- Implement and build software systems
- Do verifying simulations
- Order parts
- Build the prototype and verify functionality

4.2 Project Plan for the Fall Term

When continuing the project in the fall, the first priority is to recap, refine and decide on a final design concept. The goal is for all project members to be updated on the work that was done during the spring term as well as get a unified view regarding the future direction of the project. The next phase will be to create a final sketch which will be converted into a detailed CAD model of the whole system. Simultaneously, additional research will be done to decide what kind of electric components, mechanical parts, and control structure will be needed in order to realise the project. Ideally, most of this work will be done during spring or summer but the final deadline will be set to the middle of September to have time to purchase the remaining components. The next phase will be a development phase in which the project group will be divided into three subgroups. Each subgroup will focus on one of three areas which are mechanical development, electronic and software development, and control development. This development phase will last from the end of September to the middle of November and then transition to a final phase that includes integration of all the different subsystems. When the integration of all subsystems is complete, the test rig should be ready for validation and verification. A detailed description of the project plan can be seen in Appendix B which includes a preliminary GANTT chart for the fall term.

4.3 Risk Analysis

A risk analysis was conducted to identify and evaluate potential risk factors that could negatively impact the project. This was done by applying a similar risk management method that is presented by the Project Management Institute (PMI) [69]. It begins with risk identification which means that every single risk factor is divided into four different categories;

- **External:** Supplier, Political, Pandemic, etc
- **Organisational:** Resources, Project dependencies, etc
- **Project management:** Planning, Schedule, Communication
- **Technical:** Requirements, Performance, Quality

Every risk factor is then evaluated with respect to the probability of occurrence and the consequential impact it might have on the project. This is done with the help of a qualitative evaluation matrix that facilitates an objective assessment of all risk factors, see figure 39.

		Consequence			
		Marginal 1	Moderate 2	Critical 3	Catastrophic 4
Probability	Improbable 1	1	2	3	4
	Unlikely 2	2	4	6	8
	Possible 3	3	6	9	12
	Likely 4	4	8	12	16

Figure 39: A risk qualitative evaluation matrix.

The matrix provides a risk measurement value which is the product of the probability of occurrence and the consequential impact. PMI refers to this value as a measurement of potential risk exposure. The risk with the highest risk exposure score is also the one that is going to need most attention from the project group. A list of potential risk factors for this project can be seen in appendix C.

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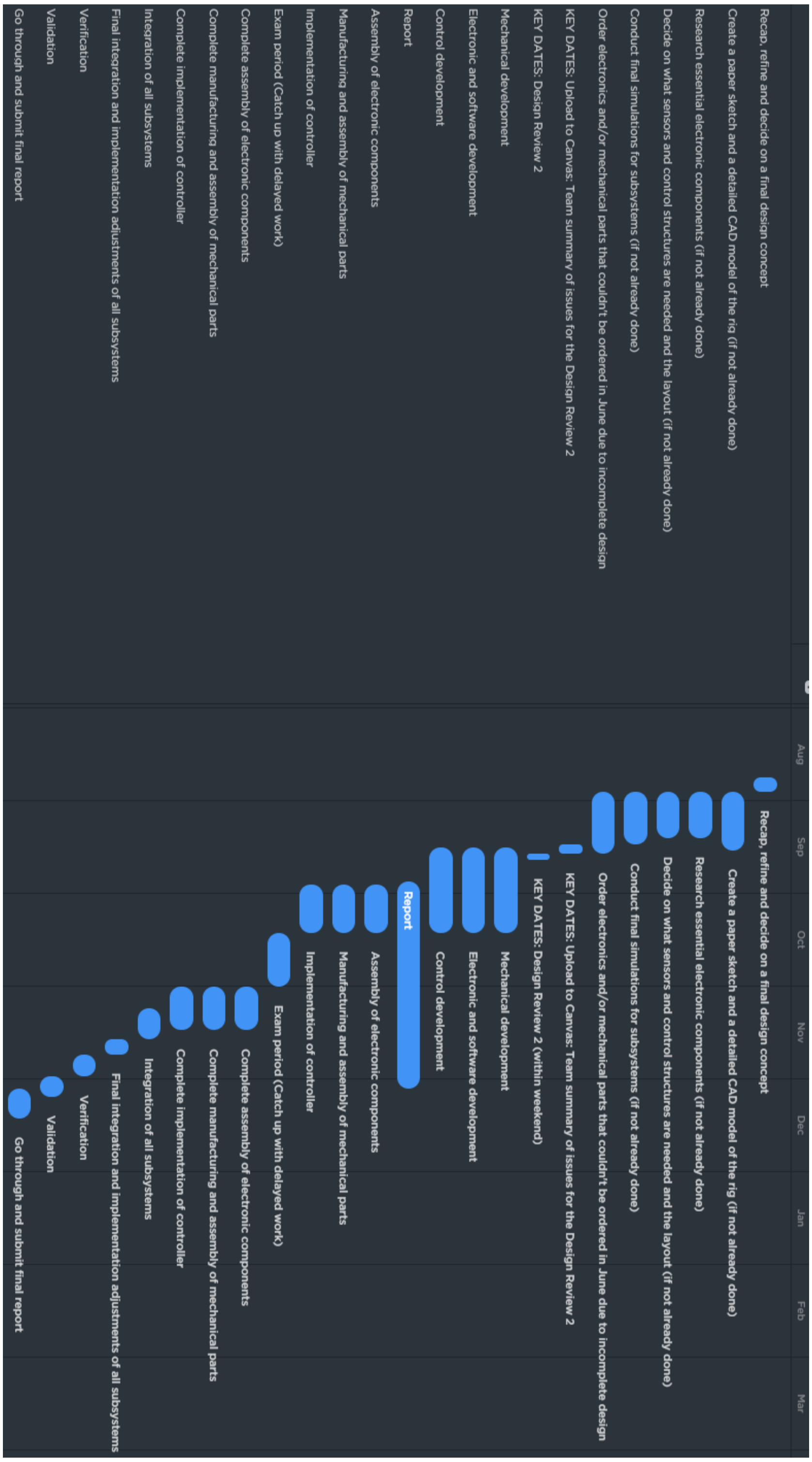
A Weighted Evaluation Matrix

Evaluation Matrix

Requirement Category		Stakeholder requirements	Weight	Concept 1		Concept 2		Concept 3	
		The test rig should...		Concept Point	Score	Concept Point	Score	Concept Point	Score
Operator Model	1	Simulate an human operator accurately (compared to Atlas Copco test data)	4	4	16	3	12	3	12
	2	Enable repetitive and accurate test results (of reaction forces and accelerations)	5	4	20	4	20	4	20
	3	Being able to test new tightening strategies (redundancy, not just one tightening technique)	2	4	8	3	6	2	4
Phyisical Requirements	4	Be robust enough to operate during heavy vibrations	4	3	12	3	12	3	12
	5	An IRTT must fit between the tool and the bolt	5	3	15	4	20	4	20
	6	Be robust enough to withstand exteme cases (reaction torques up to 100Nm)	3	3	9	3	9	3	9
	7	Fit a beam test joint.	5	3	15	4	20	4	20
	8	Adjust for height difference from bolt tightening	4	4	16	3	12	2	8
	9	Work for different tightening strategies (Quickstep, turbotight, tensorpulse?)	3	5	15	3	9	2	6
	10	Be able to send data to a Dewe43	5	4	20	3	15	3	15
	11	Work with threee different angle tools: STR series (ETV STR61-50-10, ETV STR61-70-13, ETV STR61-100-13)	2	3	6	4	8	4	8
	12	60 degree rotary displacement	3	5	15	4	12	3	9
Our Suggestions	13	Have a lifespan of >10 000 uses	3	3	9	3	9	3	9
	14	Movable by two people	4	2	8	2	8	3	12
	15	Short list of components/low manufacturing complexity	3	2	6	3	9	4	12
	16	Easy maintenance (in terms of sustainability)	3	3	9	3	9	3	9
	17	Independant installation of tool (ease of use)	4	3	12	3	12	4	16
Measurment	18	Accurately measure reaction torques on nut	3	3	9	3	9	3	9
	19	Accurately measure vibrations	2	2	4	3	6	3	6
	20	Accerately measure reaction force	5	5	25	3	15	3	15
Wishes	21	Work with pistol grip tools	1	2	2	2	2	2	2
	22	Simulate different types of operators (different model parameters at least as continued work)	2	4	8	3	6	4	8
	23	Different operator positions can be tested, horizontally and vertically	1	1	1	1	1	1	1
Safety:	24	Protect against finger injuries	4	3	12	4	16	4	16
	25	Have an emergency system that stops all operations	5	3	15	3	15	3	15
	26	Noise levels below harmful threshold (noise level under XX db)	4	3	12	4	16	3	12
Costs:	27	Enable component and manufacturing costs within budget	3	3	9	4	12	4	12
					0		0		0
				Total Score:	308	Total Score:	300	Total Score:	297

B GANTT chart - Fall Term



C Risk Assessment

Risk Assessment					
Risk Categories	**Risk Description**	**Risk Severity**	**Risk Probability**	**Risk Level**	**Strategy**
External Risk Factors	School not accessible due to Covid-19	Catastrophic	Unlikely	High	Set up digital communication channels, e.g. Zoom. Download necessary software programs to private computers, e.g. Solid Edge and Matlab.
	Stakeholders change the requirement for the project	Critical	Improbable	Middle	Accept the Risk
	Long delivery time for important components	Critical	Likely	Very High	Try to order the majority of the components before the autumn semester begins. Check with Atlas Copco if they already have unused components that we can use. Alternatively, check if they have agreements with relevant subcontractors, which can speed up deliveries
	Components are out of stock	Critical	Unlikely	Middle	Contact the manufacturer to see when it is possible to order the desired components. Research alternative components if the preferred one are out of stock for a longer time
Organizational	Project members are absent due to illness e.g. Covid-19	Moderate	Possible	Middle	Create subgroups with at least two project members in each group so that the project is not affected by one person being unavailable
Project Management	The project members have different views on how the project should be realized	Critical	Improbable	Middle	Apply an agile framework with continuous meetings in order to make sure that everyone are working towards the same goal
	Not keeping up with the project plan due to late deliveries within the project	Critical	Possible	High	Prioritize tasks that may be a risk for becoming a bottleneck. Apply an agile framework with short sprints to detect bottlenecks
Technical	The test rig does not meet the stakeholder requirements during the validation phase	Marginal	Unlikely	Low	Inform the stakeholders about the capacity of the test rig