Design of a Disconnect Clutch and Actuator for a 48V P4 Electric Drive Transmission

Oscar Blomqvist and Matthis Schneider

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MASTER THESIS

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Oscar Blomqvist and Matthis Schneider

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Department of Design Sciences
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Abstract

The vehicle industry is making great strides in electric and hybrid technologies, representing a significant step towards reducing emissions from personal vehicles.

In order to make the transition to new hybrid technologies as convincing as possible, well-functioning and smooth transitions between traditional combustion engines and electric motors are required. So-called mild hybrids enable a cost-effective method to electrify existing drivetrains, as a smaller electric motor can be used separately from the combustion engine drivetrain.

Due to the reduced size of the electric motor, it is not able to rotate at the high top speeds of many modern personal vehicles. This necessitates a means to disconnect the electric motor from the wheels. This disconnect device is conceptualized in this report.

The aim of this thesis is to develop the disconnect system, which incorporates both the mechanical configuration and the actuator providing force transfer. A requirement is the use of a dog clutch.

The product development process of the report follows the methodology produced by K. T. Ulrich and S. D. Eppinger, resulting in seven promising concepts of which two were chosen for detailed design. A total of 34 concepts were screened. Despite being removed, some concepts show additional potential and are recorded in a concept library. Placements of the disconnect system, as well as actuator technologies were evaluated.

The two chosen concepts, concept WY - *First Pinion* and V - *GenVI Hydro Sleeve*, present two different methods for disconnection placed at different positions along the drivetrain. This enabled both effective work with one concept for each thesis student, and freedom of choice for BorgWarner during evaluation.

The thesis continued into the phase of detailed design but further work is required before the concepts can be verified and finished.

Keywords: Driveline disconnect, hybrid technology, P4-architecture, electric rear axle, product development.

Sammanfattning

Hybrida fordon står för en allt större sektor av fordonsbranschen, och är ett viktigt steg för att minska utsläppen från privata fordon. För att göra övergången till nya hybridbilar så friktionsfri som möjligt behövs smidiga övergångar mellan traditionella förbränningsmotorer och allt kraftigare elmotorer. Så kallade milda hybrider möjliggör en kostnadseffektiv metod att elektrifiera befintliga drivlinor, då en mindre elektrisk motor kan användas fristående från förbränningsmotorns drivlina.

På grund av den förminskade storleken på den elektriska motorn kan denna inte rotera i högre hastigheter som moderna personbilar lätt uppnår. Därmed måste den elektriska motorn kopplas lös från hjulen, och denna frånkoppling har konceptutvecklats i denna rapport.

Målet med detta examensarbete är att utveckla frånkopplingsenheten, som innefattar både den mekaniska utformningen och aktuatorn som står för kraftöverföringen. En så kallad "dog clutch" ska användas i mekanismen. Produktutvecklingsprocessen i rapporten följer metodiken framtagen av K. T. Ulrich och S. D. Eppinger, som resulterade i sju lovande koncept varav två valdes för slutgiltig detaljdesign. Även utvärdering av frånkopplingens placering längs drivlinan, samt aktuatorteknologier genomfördes.

De två valda koncepten, koncept WY - *First Pinion* och V - *GenVI Hydro Sleeve*, presenterar två olika metoder för frånkoppling placerade på olika drivlinepositioner. Tack vare detta möjliggjordes både effektivt arbete med ett koncept per examensarbetare, men även valfrihet för BorgWarner vid utvärdering. WY-konceptet utnyttjar en solenoid och är placerad närmast elmotorn. V-konceptet använder en hydraulikpump och är placerad längst ut på drivaxeln. Tack vare de olika placeringarna kan två olika koncept med för och nackdelar väljas.

Examensarbetet gick vidare till detaljerad design av koncepten, men ytterligare arbete krävs för verifiering och färdigställning av systemen.

Nyckelord: drivlinefrånkoppling, hybridteknologi, P4-arkitektur, elektrisk bakaxel, produktutveckling.

Acknowledgments

We would like to express gratitude to our supervisor at BorgWarner, Fredrik Larsson, for his support and many interesting discussions, as well as our supervisor at LTH, Per-Erik Andersson, for his continuous feedback and motivation. Furthermore, we are thankful to our colleague Sven Kalkan, for invaluable help and input during the project.

Our friends and colleagues John Huzell and Johan Göransson helped make our time in Landskrona fun and welcoming, for which we are very grateful. A huge thank you also goes out to our friends Olov Günther-Hanssen and Gunnar Granlund, for making the most of new, unfamiliar routines with us.

The COVID-19 health crisis began during the early spring of 2020 and continued through the extent of our thesis. This presented new challenges and unfortunately impacted parts of our project. We are thankful to BorgWarner for providing us with the time and means to continue with the project during the crisis.

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List of Acronyms

ASC Active Short Circuit

AWD All Wheel Drive

CAD Computer Aided Design

DC Disconnect Clutch

ECU Engine Control Unit

EM Electric Motor

EMF Electromotive Force

FEA Finite Element Analysis

FWD Front Wheel Drive

HEV Hybrid Electric Vehicle

ICE Internal Combustion Engine

MHEV Mild Hybrid Electric Vehicle

PDU Power Drive Unit

PTU Power Transfer Unit

RBS Regenerative Breaking System

RDM Rear Drive Module

RDU Rear Drive Unit

RPM Rounds Per Minute

RWD Rear Wheel Drive

Chapter 1

Introduction

This chapter deals with the background, objectives, assumptions and limitations of the development of a disconnect device for the electric drive of a hybrid vehicle. Furthermore, it presents an overview of the methodology used during the project.

1.1 Team Background

The design team consists of Matthis Schneider and Oscar Blomqvist who are two students studying Mechanical Engineering at the Faculty of Engineering at Lund University. Both are specializing in product development where they have studied courses in mechanical design and material science.

1.2 Background

These days more and more pressure is coming from governments and the public to reduce the emissions of road vehicle exhausts. Awareness of a "greener" future has started among people. In 2019, the European Parliament and Council adopted regulation *2019/631*. This means among other things that from 2025 on, manufactures have to meet the requirements specified, for example, a large percentage of the emission must be reduced [1]. This is important since the transportation sector contributes to about 10% to 30% of total greenhouse gas emissions worldwide. At the same time, it is predicted that ownership of light-duty vehicles will increase by about 100% to over 2 billion in the year 2050 [2].

To be able to meet these requirements, vehicle manufactures must rethink how the car will be driven. Traditional internal combustion engines (ICE) have over many years been optimized to reduce the emission and fuel consumption, but there seem to be technological and economical limits for these types of vehicles to have a future. [2]

Hybrid electric vehicles (HEVs) and pure electric vehicles (EVs) are responsible for an enormous shift in the automotive industry to fulfill these requirements. Not only can HEVs and EVs improve the fuel economy but also improve the overall performance of the car. To do so, all the different subsystems must be optimized. Meeting these requirements while at the same time fulfill customer's wishes and demands regarding performance, cost, emissions and safety is a hard challenge, but a necessary step towards a sustainable future for society.

1.3 Mission Statement

The objective of this thesis was to design a novel concept of a disconnect clutch for a hybrid vehicle. This was done by using methods for concept generation and concept selection and then iterative development of the product(s) using CAD-software and calculations. Heavy effort has been put into the actuator choice and mechanic connection to transmit the power to the dog clutch.

1.4 Assumptions and Limitations

At the start of the thesis, it was assumed that the concept would be a general concept that could be implemented at a later stage to specific car models. Some constraints and dimension factors were provided that is usually required from customers but in general, the design space was very open.

It was assumed that a dog clutch was to be used. No specific model was provided and it was estimated that different types of dog clutch could be used. Another assumption was that the electronic motor could be spun up to enough speed for engaging.

The system should follow all relevant standards, such as ISO 26262, as well as receive an ASIL rating. This rating will only be evaluated if time permits during the project and explicit standard compliance checks of all concepts are regarded as further work. The system requirements derived in collaboration with BorgWarner are considered sufficient safety requirements during the concept stages.

1.5 Method

1.5.1 Product Development Process

The development of the disconnect device, henceforth known as the *Disconnect for 48V P4*, will follow the general methodology described by Ulrich and Eppinger, which is an extensive product development method undergoing several different Phases before arriving at a final concept. The process can be outlined as follows. [3]

Phase 0 - Product Planning - The actual product development process is launched, then market objectives and opportunity identification are assessed. This Phase produces the *mission statement*. The existing portfolio of projects is also looked into.

Different hybrid architectures and disconnect systems were examined, the general aim of the project was set and the first literature studies performed.

Phase 1 - Concept Development - During this Phase, the needs of the target market are identified, product concepts are developed and one or multiple concepts are chosen for continued development.

The majority of the project time frame was spent in this Phase. Needs and metrics were established and concepts were generated and evaluated during multiple steps. Ultimately, the first selection of concepts was chosen for further scoring. Of these, two were promising and were selected as final concepts, receiving further development. Phase 2 was begun before the first concept selection, as e.g. the subsystems and target specifications required more work.

Phase 2 - System-Level Design - In this Phase, the product architecture is defined, subsystems are concretized and key components receive rough designs. Initial thoughts on production and final assemblies are processed here as well. This Phase generally produces geometric layout of the product, specified subsystems and preliminary flows for the production process.

As mentioned above, Phase 2 was during some periods of the project worked on in parallel with Phase 1. Subsystems and system functions were established during this Phase, but the division of e.g. subsystems was very dependent on some of the concepts.

Phase 3 - Detail Design - Now the complete specification of the product geometry, materials and tolerances is set. The production system is prepared for the product with tooling and process plans. *Control documentation* is written for every part of the product.

The project began to move into Phase 3 but did not see any major developments here. Manufacturing was a focus from Phase 2 (design for manufacturing), and standard components - bearings, springs, specific actuators - were examined and a general selection of these was made.

Phase 4 - Testing and Refinement - Multiple preproduction versions of the product are constructed and evaluated. *Alpha prototypes* are also built with parts using the same geometry and tolerances intended during production. Alpha prototypes are built to verify that the product will work as intended and will satisfy customer needs. *Beta prototypes* are then constructed using parts created with the proper processes but not assembled with the intended methods. These prototypes are intended to show reliability and performance.

Phase 5 - Production Ramp-Up - The product is now constructed using the intended production systems. The process is ramped up to prepare and teach the workforce to build the product. The built products are carefully evaluated and any remaining problems must be worked out. The product is at some point launched, and a *postlaunch project review* occurs.

This project will arrive at and enter Phase 3, detail design, but will not develop the concepts further from there. The methodology according to [3] has been used as a tool and adjusted to fit the project. The chosen methodology is as follows.

Chapter 3 - System Specification - Target specifications as mandated by BorgWarner are received and used to complete the full specifications. The customer needs are established and interpreted to metrics. Phase 1 begins.

Chapter 4 - System Functions and Subsystem Division - Phase 2 begins here, in tandem with Phase 1.

Chapter 5 - Architecture Scoring - The best placement of the disconnect device along the drivetrain is determined.

Chapter 7 - Concept Generation - The different concepts are invented with the guidance of system specification and customer needs. The concepts can be a simple sketch or in some cases a CAD-model.

Chapter 8 - Concept Screening - Where a first initial screening of the generated concepts are made in order to erase concepts that are not feasible.

Chapter 9 - Actuator Scoring - Since different types of actuators were used and could be placed on different concepts, a separate scoring was performed to establish the best ones.

Chapter 10 - Concept Scoring and Concept Selection - Where the top level concepts chosen from the *Concept Screening* were scored against each other to obtain the best concept. From the concept scoring the concept to go further with are chosen. Reflection of the process were made in order to choose the right one. This is known as a decisive point in the development process. Phase 1 ends, Phase 2 nears the end.

Chapter 11 - Detailed Design - Phase 3 begins in tandem with Phase 2. In the detailed design Phase, improvements of chosen design were made regarding to functionality, feasibility and manufacturing. Relevant calculations and simulations were made in both assembly level and component level to support these.

All of the scoring and screening procedures in this report have been performed by the degree workers, but have been discussed and confirmed by the supervisors at BorgWarner. Deviations from this will be mentioned in the applicable method sections.

Some methodology by Pahl G. and Beitz, W. et.al. was also used during the project. Similar steps were recommended during the design process, with some differences. This was used to get a second opinion in different cases where more consideration had to be made on the process. [4]

Chapter 2

Theory

This chapter explains the theoretical knowledge needed to understand the report. It briefly explains the history of hybrid vehicles and proceeds with different types of modern hybrid vehicles including the electric motor and other adjacent components. Furthermore for understanding the disconnect device the chapter explains the structure of drivetrains, the dog clutch used and actuator variants that were looked into the project. It ends with presenting some similar systems already on the market.

2.1 The Hybridization of Automobiles

Modern vehicles with internal combustion engines (ICE) give the driver good performance and long ranges without refueling due to the high energy density of petroleum fuels. The downside with this type of vehicle is the poor fuel economy and most important, pollution effect on the environment. The main reason for the poor fuel economy is that internal combustion engines have low efficiencies due to thermal and parasitic losses and it not always operating at its most efficient characteristics. Vehicles powered by electric motors have some advantages that ICE vehicles do not have. These are high energy efficiency and zero environmental pollution while driving. Drawbacks with electric vehicles (EV) is the range and the currently high purchasing cost. Due to the drawbacks and advantages of both ICEs and EVs, a large portion of the vehicle fleet now consists of Hybrid vehicles, which combine the advantages of both these powertrains. For a short to medium time frame ahead, hybrid powertrains are recognized as one of the more effective ways to reduce CO_2 emissions for the automotive industry. It gives the industry a long term means proceeding toward the zero-emission goal. [5]

2.1.1 History of Hybrid Vehicles

The first hybrid car can be derived back to the golden age of electric cars. In the second half of the nineteenth century, the vehicle market exploded with new electrical vehicle inventions. One common problem the fully electrical vehicles had, like today, was the range of the batteries. One solution to this problem was the hybrid vehicle. The first gasoline-electric car was shown at the Paris Exposition in 1900. This was developed by Ferdinand Porsche and was called "System Mixte Lohner Porsche", see figure 2.1. The Mixte was popular on the market and over 300 were produced. The nineteenth-century hybrid vehicle also included regenerative braking, further explained in section 2.3.2. The hybrid vehicles were said to be the best of two things: quiet while driving in the city and unlimited range available while exploring the countryside. Unfortunately, there were some drawbacks. One big issue was that the control of the electric motor was not mastered. Another one was that the costs were high compared to competitive products. This, combined with the competitive and relatively cheap mass-produced gasoline car from Henry Ford in year 1904, may be the reason why the hybrid car disappeared during the first world war, not being considered seriously again before the early 1970s. [6]

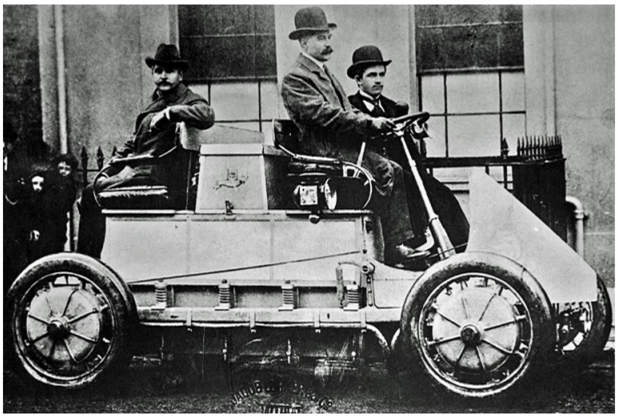


Figure 2.1: System Mixte Lohner Porsche. [7]

In 1973, during the Arabic oil crisis, the gasoline prices rose and people started to demand vehicles with less fuel consumption. The vehicle companies therefore started to invest large sums and efforts to develop hybrid technology. Despite this, no hybrid vehicles were produced that had better performance and price than the gasoline-based market. The first mass-produced and successful hybrid car of modern times was the Prius produced by Toyota, launched in year 1997. Further on hybrid vehicles started to appear such as Honda Insight and Chevrolet Volt. [8]

2.2 Hybrid Architectures

2.2.1 Micro, Mild or Full Hybrid?

Many different variants and types of hybrid drive trains exist today. These can be categorized by the degree of hybridization *micro-*, *mild-* and *full hybrid*. A micro-hybrid uses a small electric motor to help the ICE shut down and restart during idling speed, and is used in many drivetrains today. A mild hybrid uses an electric motor to assist and help the ICE, letting it shut down during coasting or braking. It can usually assist with regenerative braking at some level. It cannot drive the car by itself, only enable coasting. Full hybrids have both a primary energy source, such as ICE or fuel cells, but can also run in pure electric mode. [6]

2.2.2 Parallel Versus Series Hybrids

Hybrids can also be divided into three types of powertrain structure: *Series hybrid*, *power split* and *parallel hybrid*. In series hybrid vehicles the electric motor is directly connected to the transmission system. The car runs on the electric motor which is powered by the combustion engine, working as a generator. The advantage being that the combustion engine always can run at the most efficient level, enabling one of the most efficient hybrid drivetrain solutions on the market today. Series hybrids are not dependent on a heavy battery pack since the ICE works as a range extender. Unfortunately, it is not very popular amongst car manufacturers due to big design changes in the drivetrain and all surrounding interfaces. [6]

Power-split hybrids are by definition a parallel hybrid. These consist of two motor generators. One is used for charging and the other one to provide torque. The Toyota Prius is a typical example of a power-split drivetrain. The parallel hybrid system adds both the ICE drivetrain and electrical drivetrain together to generate mechanical torque. It can run in pure electric mode, mostly utilized in low speeds like urban areas where the ICE is not very efficient. When the ICE runs more efficiently, e.g. a highway, the EM can be used to provide additional power. From a manufacturer's perspective, a advantage with the parallel hybrid is that it can be directly added to current ICE vehicles without a large number of design changes. A downside is that the ICE is usually oversized and therefore raises fuel consumption and emissions compared to other hybrid solutions. [9]

2.2.3 Parallel Hybrids

Parallel hybrids are further divided into different setups. These setups are denoted P0-P4 and can be seen in figure 2.2. P0 uses an electric motor directly at the engine. It acts as a belt starter generator. P1 uses an electric motor at the crankshaft between the engine and the clutch 1 (C1 in figure 2.2). It can supply torque assist to the engine and recuperate energy during braking. It does not allow for pure electric drive due to the placement, and reduced size and performance of the electric motor. P2 places the electric motor at the input of the gearbox. Because of the extra clutch 0 (C0 in figure 2.2), the engine can be decoupled while driving purely electric or regenerating energy from braking and thus allow for higher efficiency compared to the P1 layout. P3 places the electric motor on the output of the gearbox. In this position it is not able to use the variable gear ratio of the transmission. Shifting the gears into neutral while driving electric can still achieve a higher efficiency than both P0 and P1. P4 places the electric motor at the free axle (in most cases the rear axle), where it can act completely independent or in tandem with the ICE via a prop shaft. P4 is the system used in the design of this thesis and will be explained further in section 2.3, where some terminology will also be elaborated. [10]

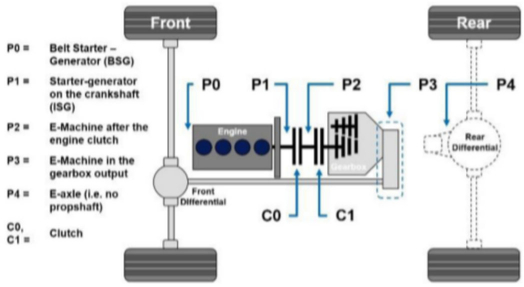


Figure 2.2: Hybrid architectures along the drivetrain. [11]

2.3 The 48V P4 System

The P4 architecture means the car is utilizing an *e-axle* at the rear. The front axle is conventionally driven by an ICE and the rear axle by an electric motor, see figure 2.2. The most important distinction from an AWD car, apart from the obvious electric motor, is often the absence of a propeller shaft, which would connect the rear axle with the transfer case of the ICE. [10]

2.3.1 Working Principle

As the 48V electric motor does not have sufficient power to propel the car at higher speeds, it is mainly utilized during starting (E-launch), as a torque assist, for active coasting and when recuperating braking energy [10]. The 48V electric motor may only have around 10% of the power the ICE provides [5]. A large advantage of having a P4 hybrid with a disconnect functionality is the ability to remove parts of the drivetrain during normal driving. As the 48V electric motor is relatively restricted in use during higher speeds, it is better to simply remove the losses of the rear drivetrain, which saves energy. Hence the lower energy usage of the ICE when using both motors does not offset the friction in the longer drivetrain when cruising.

E-launch - The E-launch functionality is for very low speeds with frequent stops, where the car is maneuvering parking lots or congested traffic in urban areas. The required power for the EM is very low and as long as the battery has enough charge the ICE can stay turned off. [10]

Torque Assist - Torque assist helps the ICE when accelerating, and the bulk of the power is not supplied by the EM. Torque assist is preferable at low to medium speed accelerations, as higher speeds (over a threshold speed) can damage the EM. Torque assist can also be used to stabilize the car. For example during dangerous under-steering when the yaw moment over the rear axle is not sufficient. [10]

Active Coasting - Active coasting means the gas pedal is under a certain threshold, and the car needs very little energy to stay at a certain speed. The EM can in this case assist in keeping the speed constant, while the ICE is disengaged to remove its motor braking effect from the driveline. Thus, the resistance in the driveline is kept at a minimum. If the gas pedal is pushed down further, the ICE is started, connected and begins supplying torque to raise the speed. [10]

2.3.2 Regenerative Braking

Regenerative braking systems (RBS), i.e. recuperating braking energy, means the EM acts as a generator, directly supplying the battery with power. There will be a motor braking effect by the EM over the rear axle, but this is usually not sufficient to stop the car completely and not within time frames. The regenerative braking can only be applied on the axle driven by the motor and the mechanical brakes will at all times assist in the total retardation of the vehicle. [12]

According to UN/ECE regulation ECE13 two types of regenerative braking are allowed, type A, *parallel RBS* and type B, *serial RBS*. Type A is not part of the service brake system, while type B is. This means that type A acts as a type of motor braking while the gas pedal is released, and type B acts in tandem with the mechanical brakes when the brake pedal is pushed. [13]

The recoverable energy will always be lower than the total kinetic energy during deceleration, as evident by losses in e.g. the electric motor. [14]

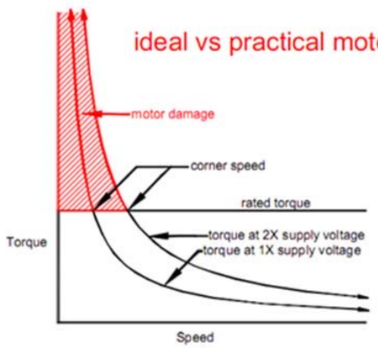
2.4 Electromagnetic Motors

The principle of operation for electric motors is relatively simple, where magnetic currents are manipulated to result in magnetic fields that rotate the output axle due to produced magnetic forces. There are different types of EMs, e.g. DC (direct current) motors, induction motors, asynchronous motors, etc. Their working principle will not be explained further in this thesis. There are some situations where extra attention must be given to the properties of the EM. There is a hard threshold in speed where the EM in a 48V hybrid can receive damage due to excessive heat, often between 70-140 km/h [15]. This is where the disconnect action will have to occur.

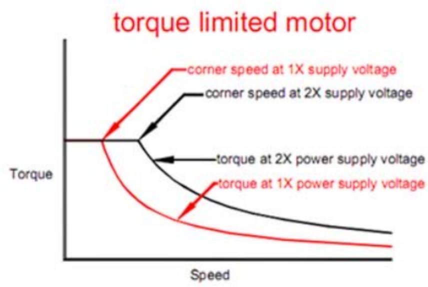
If the EM detects an internal failure, different strategies are depending on the current speed. The strategies for handling an internal failure are separated by a threshold, the *corner speed*.

2.4.1 Corner Speed

In an ideal electric motor, the torque generated varies inversely with rotational speed. Torque is proportional to current which means that a low speed gives high torque and high current which is shown figure 2.3a [16]. High current risks destroying the motor since the copper windings in the motor generate heat. The speed that determines where dangerous levels are met is called the corner speed [17]. To prevent this the drive of the motor has to be limited to avoid reaching these high currents. For performance, this means that below the corner speed the torque will be constant. Over the corner speed, it will start to drop inversely with the speed, this due to increasing *back-electromotive force* (back-EMF). Past the corner speed, the power is constant. The torque/speed with constant torque below corner speed can be seen in figure 2.3b. It is worth mentioning that different parameters determine the corner speed e.g. how efficient the cooling system of the motor is. [16].



(a) Ideal torque versus speed in a electric motor [18]



(b) Constant torque (below corner speed) at low speeds in a real motor. [19]

Figure 2.3: Speed versus torque in an electric motor.

2.4.2 Active Short Circuit - ASC

If the EM speed is above the corner speed, the safe state during a detected internal failure is *active short circuit* (ASC). To avoid the EM pushing large currents backward through the electrical system the motor alternating current lines are short-circuited. One difficulty with ASC is a resulting torque over the EM, see figure 2.4. The torque is high for low speeds and decreases when the speed increases. This extra torque makes the disconnection of drivetrain components more complex, as the EM has to be disconnected during ASC. [20]

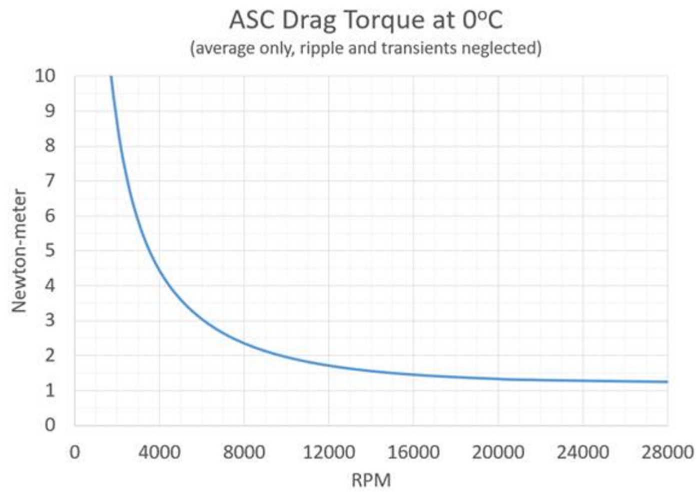


Figure 2.4: Typical resulting torque over an EM during ASC. [21]

2.5 Automotive All Wheel Drivelines

Drivelines for commercial cars have seen many improvements and developments since the advent of more modern four-wheel drives. The transfer units, which distribute torque to the front and rear axles, can change the torque ratio seamlessly during driving. An example of this is the Audi Quattro self-locking mid differential.

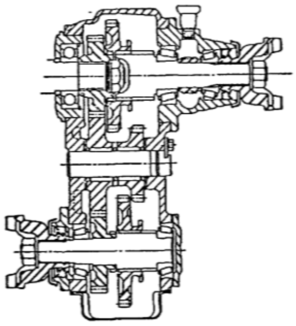
Being able to at will remove parts of the driveline which are not momentarily being used has seen many uses in commercial cars. The famous Jeep used during World War 2 (Willys MB), see figure 2.5a, used a simple form of disconnect in the transfer case to easily disconnect the front axle for better performance on flat, straight ground [22]. The transfer case in question is shown in figure 2.5b.

2.6 The Dog Clutch

An ordinary clutch for engaging or disengaging the engine to the drivetrain during gearshifting uses frictional plates that transmit torque proportional to the frictional forces. Frictional clutches thus have the benefit of a smooth driveline disconnection. These clutches are subject to frictional losses, i.e lower efficiency and disc wear. If the torque transfer over the clutch doesn't need to be controlled directly a dog clutch can be used instead. [25]

In this thesis, a dog clutch will be used for the disconnect system. A dog clutch is a mechanical connection that facilitates torque transfer without any friction or slip

(a) The famous Willys MB, commonly known as the Jeep, produced 1940 to 1945. [23]



(b) Transfer case of the Willys MB, using a sliding gear layout shifted by a fork. [24]

Figure 2.5: The Willys MB and its transfer case.

but instead with interference. These clutches are often used for applications where there is no need for intermediate torque transfer. Frictional losses can still be present during engagement if for example a synchronization mechanism is incorporated. Its function is to synchronize the speed of the connecting axles before clutch engagement to facilitate a smooth and safe connection. [25]

2.6.1 Working Principle

The dog clutch is generally composed of two or three parts: Two sets of dog teeth and some type of axial or radial mechanism to connect the teeth. This mechanism can be a sleeve that is axially moved or one of the dog teeth axles being able to axially slide in and out of connection with the other. This creates interference that transmits the torque. When choosing a dog clutch it is important to be aware of the disengagement force. This is derived in section 11.1.1.

2.6.2 Synchronization

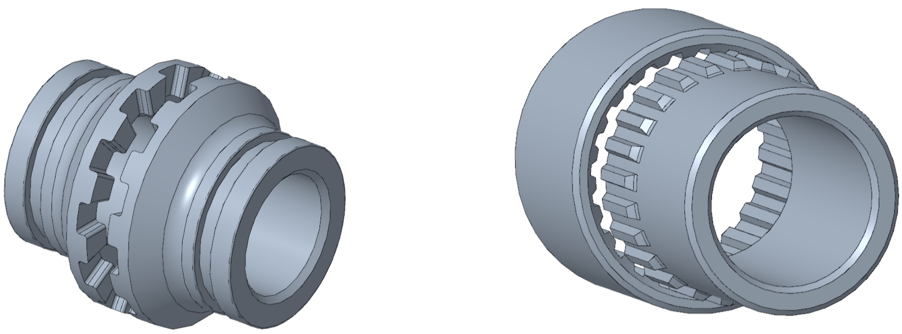
The synchronization can be done by e.g. a synchromesh or as in this case: Using the electric motor on the hybrid vehicle to spin the axle to the required speed [26]. By gathering data from e.g. wheel speed sensors the EM ECU can supply data for the correct speed. However, the control sequence of synchronizing the EM is out of scope for this thesis and it will be assumed that the EM can sufficiently synchronize with the axle. Some information about the synchronizing will be given below.

When synchronizing, there must always be a small difference in rotational speed between the input and output shaft. In theory, a speed difference of zero could potentially hinder a connection completely, if the clutch is out of phase. An increase in speed difference can increase the chances of a successful connection, as well as lower the time for a full connection [27]. Unfortunately, higher speed differences result in noise and harshness during the connection (colloquially referred to as *klonk* in Swedish). Klonk can be explained as the relationship between the stiffness of the system and the amount of energy having to be absorbed by it. Higher amounts of energy lead to more difficulties during connection, while a stiff system is favourable as it can absorb more energy. Due to security aspects, it is wise to make sure the dog cannot connect at high differences in rpm, as unintentional connections at high speeds should be avoided. Naturally, the number of teeth in the dog clutch has an impact on the optimal speed difference. The gearing between the EM and the DC system also plays a role in klonk, as the EM must reduce its speed momentarily during connection.

The difference is usually $\Delta n = 0$ to 30 rpm, where n is rotational speed, according to our supervisor. This should be kept in mind when the dog clutch design is finalized, as for example the number and size of teeth (which both affect the synchronization) are relatively interchangeable.

2.6.3 Dog Clutch Variants

There will be two variants of dog clutch examined during this project, type A and type B. The variants are shown in figure 2.6. Type A is a very common dog tooth geometry, where one of the halves is pushed axially into contact with the other. Type B, also called *sliding tooth collar clutch*, is similar to a spline but with more possibility for movement when the teeth are in contact. Type B has the advantage of more of teeth in contact at the same time. It also requires smaller sliding distance when engaging.



(a) Example of a dog clutch, type A.

(b) Example of a dog clutch, type B (sliding tooth collar clutch).

Figure 2.6: Two examples of a dog clutch.

2.7 Disconnect Architectures

Five distinct rear-axle disconnect architectures will be evaluated in this thesis. The positions range from placement at the wheel hubs to directly at the electrical motor, which varies the type of challenges during the system design. The following five architectures are widely used and examined for this project:

- Propeller shaft DC
- Internal differential DC
- Wheel Hub DC
- Single side axle DC
- Twin side axle DC

See table 2.1 for the respective placements of the architectures.

2.7.1 Propeller Shaft Disconnect

By placing the disconnect between the electrical motor and the differential, the internal pinions of the differential are stopped during normal, straight cruising. Balancing of the components can be a challenge, as the rotational speeds are the highest directly at the output of the electrical motor.

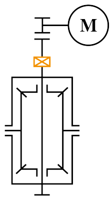
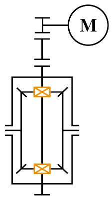
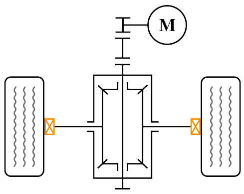
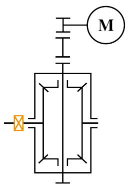
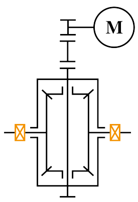
2.7.2 Internal Differential Disconnect

The purpose is to decouple some component inside of the differential. An internal disconnect in the differential gives a very compact design with some of the same benefits as the propeller shaft DC. There are still problems to be solved, especially packaging as the disconnect must be placed within an already cramped and restricted system. There are also difficulties with introducing new components into a already sophisticated and thoroughly designed system, i.e. the final drive of an AWD vehicle.

2.7.3 Wheel Hub Disconnect

Disconnecting the driveline directly at the wheel hubs is beneficial for any frictional or geometrical losses in the system, as the final drive and differential is stationary. As the whole driveline between the EM and wheel hubs is at rest, there is an increase in required torque by the EM to begin rotating the axles. This has a low impact on the overall performance. Construction-wise, this solution is more complex. The further out the system architecture moves, the higher the torque and shock loads from the environment and road surfaces. There must also be two separate disconnect systems mounted on the vehicle, although they can be controlled from the same ECU.

Table 2.1 Schematics of the proposed architectures.

Architecture	**Layout**	**Comment**
1. Prop Shaft DC		
2. Internal Diff DC		
3. Wheel Hub DC		
4. Single Axle DC		
5. Twin Axle DC		

2.7.4 Single Side Axle Disconnect

This architecture exploits a feature of the differential, which is that any torque out of the differential must equal the torque input. By disconnecting one of the output axles and stopping the electric motor the majority of the differential will cease to rotate. The still connected output axle will still rotate, as well as the internal pinions which will climb on the opposite output axle gear. This solution solves the problem of having two separate disconnects for each side of the differential. There are however noise issues with this architecture, as the differential pinions will climb at high speeds during normal driving [28].

As the disconnect should always be disconnected during higher speeds, this is almost guaranteed to become an issue at some point and demands good differential design and quality. ZF Friedrichshafen has examined this solution and concluded that the open differential can coast excellently during high speeds as long as there is no load on the differential. It is claimed that there are no noise or vibration issues if executed properly. Drag losses and service life demands are claimed to be met according to performed tests as well. [29]

2.7.5 Twin Side Axle Disconnect

For this architecture, the single side axle DC is simply duplicated for both sides of the differential, which eliminates the noise issues. This will also increase the cost as two mechanisms are needed. As with the hub DC, one ECU could potentially be used to control both systems. As a result of this architecture, the entire final drive, where the moment is the same as the output wheel and differential will be stationary during disconnect. This reduces friction losses and decreases the driveline inertia when accelerating the car.

2.8 Actuators

An actuator is generally described as a part of a machine or system with the function of moving another component or linkage. There are many different types of actuators available, with various working principles. An example of a commonly used solenoid actuator can be seen i figure 2.7. The solenoid uses a magnetic field to apply force. Hydraulic actuators are another example of a commonly used actuator, and work by pumping hydraulic fluid which exerts pressure on some type of cylinder surface. This moves a piston which in turn leads to displacement. Since the concepts chosen for final design (see section 11.2) utilize these types of actuators, they will be described in more detail in the following sections. Additionally, all other types of actuators considered during the project will receive brief mentions. [30]

2.8.1 Bistable and Monostable Actuation

A bistable actuator has two stable equilibrium positions. If the mechanism is actuated to one position it will stay there until the control signal states otherwise. A normal

Figure 2.7: Example of a standard solenoid actuator for linear motion. [31]

'clicking pen' is a good example of a bistable actuator. The upside to this type of actuator is less holding force required for stable operation in both mechanism positions. [32]

The monostable actuator has only one stable equilibrium position. The force is transmitted by an actuator in one direction and the return action is provided by e.g. springs when the actuator power is cut [32]. A door with a closing mechanism can be exemplified as a monostable actuator, as the door will always close independently of which position it is in. If all power is lost the return action can always be guaranteed by the mechanical force from the spring, which is an important advantage.

2.8.2 Actuator Variants

The actuators evaluated during this project have been categorized into five general categories, depending on the primary means of energy usage in the actuator. These categories are: Hydraulic, pneumatic, mechanical, chemical, electric and magnetic. The electric and magnetic categories are merged in this chapter, as both electric motors and magnetic actuators work with very similar principles, making it superfluous to differentiate the methods.

Hydraulic

Hydraulics are characterized by high force density and widespread use in all types of vehicles and machines. The most common fluids to use as a working medium in hydraulic systems today are different types of oil. Issues to solve are the sealing of valves and connections, leakage, a relatively low actuation speed, and possible contamination of hydraulic fluid. The inverse, the hydraulic fluid contaminating or damaging other systems, must also be avoided. The following sections, describing the theory of hydraulic functions and calculations, are derived from [33].

An inherent risk when using hydraulic systems is the large force output of the actuator, which can damage other components. Some types of hydraulic actuator pumps must be spooled up before actuating, which when combined with a low opposing force risks damage as the pump motor cannot spool down fast enough in case of components seizing etc. High-pressure valves can also be included to make sure no components are damaged. The components of a hydraulic system consist of a pump or machine, piston and valves. The pump or machine fill the same roles, but reversed. The pump generates an oil flow while a machine receives it instead. Pistons use a cylinder with incoming oil flow and pressure to move the wanted object. This creates a linear motion. The valves control the direction, speed and pressure of the oil flow.

The most interesting values to obtain from a hydraulic system for this thesis is the force the piston can generate, the time it takes for it to transmit the power and the power required. Equation 2.1-2.5 are used later in chapter 11.

Displacement of a hydraulic pump can be compared to the stroke volume in a car engine. The displacement is the amount of volume covered by one stroke of the piston. Displacement for a hydraulic pump is defined in equation 2.1 and the flow generated by the pump by equation 2.2.

$$D = \frac{q}{n} \tag{2.1}$$

$$q = \epsilon_P \cdot D \cdot n \cdot \eta_{vP} \tag{2.2}$$

Where ϵ is relative modulation for specific case, q is the flow and n rotational speed (rpm). The volymetric efficiency η_{vP} is defined in equation 2.7 rises from leakage of the system. D describes the geometric displacement which can be describes as the size of the hydraulic pump or engine. Its units are $[m^3/\text{rotation}]$. The flow of the hydraulic system is given in the units $[m^3/s]$. By using the flow q, the velocity v $[m/s]$ of the piston can be derived as

$$v = \frac{q}{A} \tag{2.3}$$

Where A is the area of the piston bore. The area is obtained from the bore diameter if a cylindrical piston is used. The force F of the piston is calculated as

$$F = pA \tag{2.4}$$

Where p is the pressure in Pascal, $[N/m^2]$. Power required for actuation $E_{hydraulic}$ is calculated in 2.5 with the help of the flow q and the pressure p needed for actuation.

$$E_{hydraulic} = \frac{p \cdot q}{\eta_{totP}} \tag{2.5}$$

Where η_{totP} is the total efficiency for a pump given by hydraulic mechanic efficiency and the volymetric efficiency respectively for a pump $\eta_{totP} = \eta_{hmP} \cdot \eta_{vP}$

$$\eta_{hmP} = \frac{1}{1 + \frac{1}{\epsilon_P}(0.02 + 4 \cdot 10^5 \cdot \lambda)} \tag{2.6}$$

$$\eta_{vP} = 1 - \frac{1}{\epsilon_P} \cdot 1.5 \cdot 10^{-9} \cdot \frac{1}{\lambda} \tag{2.7}$$

$$\lambda = \frac{\eta \cdot n}{p} \tag{2.8}$$

Where η is dynamic viscosity for oil used $[Ns/m^2]$. This way of determine the efficiency for a specific load case is according to [33] an estimated one and might differ from the real one, that must be tested.

Please note all calculations have been made with the assumption that the pressure of the rod side is negligible, i.e atmospheric pressure is present. Incompressible flow is also neglected.

Pneumatic

Pneumatic systems are widely used in consumer cars, where vacuum pumps can be mounted on the engine camshaft, or vacuum is supplied directly by the intake of the (gasoline) engine. Pneumatics are also often used in industrial production lines. The relative ease of air pressure transportation (no spill or danger of contamination) make it ideal to connect many pneumatic actuators to a central air supply. The actuators can be operated where other types, such as electric or hydraulic actuators, are unfit due to only air being used to build up pressure. Pressures up to 4 bar can be used with high speeds and long stroke lengths. Calculations for pneumatic actuators are similar to those of hydraulics, but the compressibility of air must be taken into account. One major downside with pneumatics is the air supply being a high-pressure container which can cause accidents if not properly maintained. [33]

Mechanical

Mechanical systems utilizing springs and potential energy are often used for fail-safe mechanisms when the availability of electricity or fuel cannot be guaranteed. Pretension must however be used and reset. This is often accomplished with some type of motor or manual force from an operator resetting the fail-safe. Therefore, only a very small part of the system uses direct energy in the form of e.g. electricity, as the main function of the system is made possible by the collected potential energy.

Chemical
Actuators converting chemical energy into movement can use a variety of fuels. An ICE can be said to follow the same principle, as chemical energy stored in the fuel is converted into linear energy when the piston is pushed by an explosion. Nail guns commonly use chemical propellants which drive a nail with the resulting explosion. These tools can be very powerful and require a license and safety training. [34]

Ramset and *Hilti* are two well-known PAT (Powder Actuated Tool) manufacturers.

Electric and Magnetic
Electric motors have great controllability and are usually relatively simple in design, and are described further in section 2.4.

The stepper motor is a type of electrical motor which utilizes currents in a number of separate, radially arranged winding pairs to manipulate a permanent magnet or soft iron core. This enables moving it between different specified angles. This can be used to both rotate the rotor as a traditional EM, but more importantly to specify the rotational position with high precision. [35]

When being held, the stepper motor has a *holding torque*. The holding torque is often higher than the rotational torque. Additionally, there is a *detent torque*, which represents the inner resistance against rotation present in the motor. The detent torque is always static. [36]

Electromagnets operate with Maxwell forces. Maxwell describes the forces at the boundary layers at materials with different permeabilities. Electromagnets consist of a fixed element that consists of a magnetic core, a body, or a yoke together with a coil. An armature is also present, which acts as the movable part. Electromagnets holding open doors are a common example. [37]

Solenoids are a different type of magnetic actuator. Similar to the electromagnet it utilizes the magnetic field to apply force. It consists of a coil that generates a controlled magnetic field when current is applied. By definition, the length of the coil is substantially longer than the diameter. Inside the coil, a rod made out of ferritic material can be placed and will create a linear motion when current is applied. A solenoid of this type can only be actuated one way and has to be reset with a mechanical spring, for example. This makes it a good candidate for a monostable actuator. [35]

The force of a solenoid can be calculated by equation 2.9. The magnetic constant μ_0 is $4\pi \cdot 10^{-7}$, $F_{solenoid}$ is the force in Newtons, N is the number of turns, I is current, A is the area in $[m^2]$, and g is the length of the gap between the coil and iron. [38]

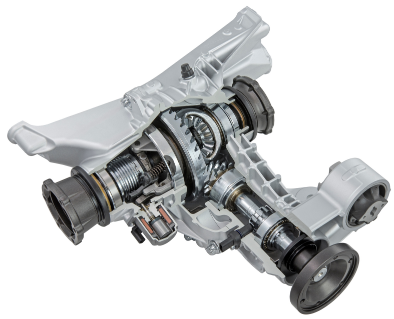
$$F_{solenoid} = (NI)^2 \frac{\mu_0 A}{2g^2} \tag{2.9}$$

It is made evident by equation 2.9 that the gap, g, between the solenoid and the piece of metal must be minimized to obtain a high force.

2.9 Disconnect Systems on the Market

There are several disconnect systems already on the market, also systems manufactured by BorgWarner such as the 48V eAWD (figure 2.8a). These systems are normally used for disconnecting parts of AWD ICE drivelines, with the main objective of lowering the total energy usage of the vehicle. Synchronization can be a bigger issue with these types of systems, as they lack an electric motor that can be spooled to any (within limits) rpm to facilitate the dog clutch connection. Most ICE systems use friction clutches or incorporate a small electric motor to help with the synchronization. A common solution is to use both a PTU (Power Take-off Unit, at the beginning of propeller shaft) and a RDU (Rear Drive Unit, at the end of propeller shaft) with different layouts of friction- or dog clutches, which can facilitate synchronization.

(a) The BorgWarner 48V P4 eAWD system. [39]



(b) The Magna Flex4 disconnect system. [40]

Figure 2.8: Complete disconnect systems on the market today.

Magna supplies e.g. BMW (XDrive) and Audi (quattro) with disconnect systems, *Actimax*, *Ultimax* and *Flex4* (see figure 2.8b). [41]

GKN Automotive sells the *ActiveConnect* disconnect module for use with AWD drivelines [42]. American Axle & Manufacturing sells the *EcoTrac*, a RDM [43].

Two large driveline manufacturers are also ZF Friedrichshafen and Aisin Seiki, which currently do not supply car manufacturers with pure e-axle disconnect systems. ZF manufacture a disconnect system similar to the Audi quattro-architecture [44].

The modes of operation for versions of the two latter systems are very similar, with the power for the disconnect action coming directly from a drive axle and solenoid (magnetic actuator) pin moving the sleeve axially by a groove or thread. [29]

2.10 Automotive Safety Standards

To verify the safety of the system, an ASIL rating (*Automotive Safety Integrity Level*) should be made. This rating is a way to gauge the safety of a vehicle system, following the ISO 26262 standard. In simple terms, ASIL is an assessment to evaluate the severity, exposure and controllability of potential risks in the system. The resulted rating is then a measure of hazards, making it possible to set realistic safety goals. [45]

Chapter 3

System Specifications

Before a product is to be developed, the team needs to know what the customer needs are. This chapter explains how these needs have been gathered and interpreted into metrics for developing the disconnect device.

3.1 Customer Needs

The customer needs are the first part of setting any specifications. These needs were chosen both through discussions with supervisors and determination of crucial disconnect system functions, such as needs for safety, etc. As this project does not finalize the system as a whole, some of the needs will not be addressed here but may be applicable for future work with e.g. control systems and manufacturing process specifics. The identified customer needs are shown in table 3.1. The importance (*Imp.*) of the needs is also specified after how the needs should be ranked. Important needs receive a *3*, and less important a *1*.

Some of these identified needs can be commented on directly. Need number 7, *The DC is easy to install & maintain*, is not relevant in all aspects, as similar products usually are planned as sealed-for-life. Maintaining the product may only imply ease to replace the part in the case of damage, etc. Need number 4, *The DC is safe*, could refer to a multitude of factors, such as maximum stresses for high torque operations, or fulfilment of relevant safety standards. This is discussed further in the next section.

3.2 Metrics

Ulrich & Eppinger let the individual customer needs become translated into metrics. Sometimes multiple metrics are required for the same need. These metrics are dependent on the need reflected and must be practical for the development team to use. Need number 4 about safety can be interpreted in different ways, and in this case, it should probably include an ASIL rating, following the standard ISO 26262

Table 3.1 Identified customer needs.

No.	Need	Comment	Imp.
1	The DC works with a dog clutch		3
2	The DC follows P4 hybrid architecture	Available DC positions	3
3	The DC exerts sufficient force		3
4	The DC is safe		3
5	The DC verifies the actuator position		2
6	The DC engages/disengages quickly		3
7	The DC is easy to install & maintain		1
8	The DC lasts many cycles		2
9	The DC does not break during extreme use		2
10	The DC is powered by the car		2
11	The DC does not use excessive power	Max. power draw & cont. use	3
12	The DC is lightweight		2
13	The DC has a low manufacturing cost		1

[45]. The maximum stresses would then be handled by need number 9. Need number 1, *The DC works with a dog clutch* has the metric *List*, which means that there are a set number of designs of dog clutches, and the concept should preferably work for any number of them. Metrics using the *Bin.* unit are binary, i.e. the unit is a simple 'yes' or 'no'. [3]

The metric *Frequency* refers to the frequency of the actuator position sensor data. *System torque* and the variable U_{tot} is dependent on where the disconnect system is placed along the drivetrain. Need number 4, *The DC system is safe*, is as mentioned above difficult to quantify. It can be interpreted in different ways, and in this case, it should probably include an ASIL rating, following the standard ISO 26262 [45]. The maximum stresses would then be handled by need number 9, but do we show that all stresses are below the elastic limits of metal linkages in the system? What happens when the product reaches a fatigued state? The decision here was made do some simple stress calculations of the most critical system components at a later stage if time permits, as well as the planned FE-analyses. By setting two metrics here, the customer need is simplified. Please note the very first concept screenings will utilize more general approximations of, for example, which linkages or constructions are stronger. A benchmarking of similar products would also be made easier by the list of metrics. The proposed metrics are shown in table 3.2. [3]

3.3 The Needs-Metrics Matrix

Using the needs and metrics, figure 3.1 can be constructed, which intuitively shows which needs correspond to which metrics. Some needs use the same metrics, which is unavoidable and not an issue.

Table 3.2 Identified metrics.

No.	Need No.	Metric	Imp.	Unit	Optimal	Acceptable
1	1	Assembles with dog clutch	3	List	All	None
2	2	Mounts on P4 hybrid	3	Bin.	Y	Y
3	3	System torque	3	Nm	$>M_{tot}U_{tot}$	$M_{tot}U_{tot}$
4	4	Complies with ISO standards	3	Bin.	Y	Y
5	4,9	Max. stress	3	MPa	$>1.2\sigma_y$	$1.2\sigma_y$
6	4	ASIL rating	3	ASIL	-	-
7	5	Frequency	2	Hz	> 2	> 1
8	6	Engage/disengagement time	3	ms	< 100	< 500
9	7	Installation time	1	s	-	-
10	7	Access & disassembly time	1	s	-	-
11	7,13	Cost	1	kr	-	-
12	8	Engagement cycles	2	n	$> 10^6$	10^6
13	10	Voltage	2	V	< 12	12
14	11	Max. power	3	W	< 200	200
15	12	Weight	3	kg	-	-

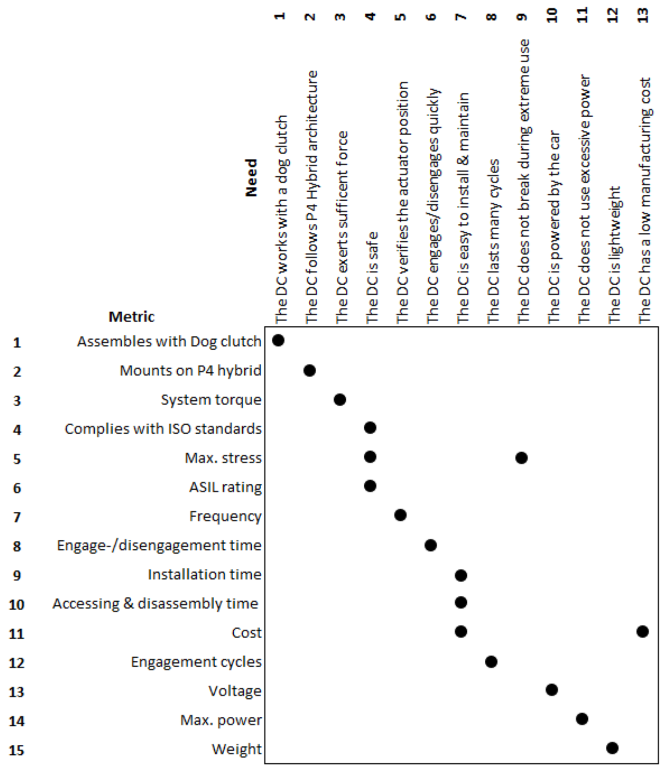


Figure 3.1: The Needs-Metrics Matrix, which shows the relationships between the identified needs and the derived metrics.

3.4 Target Specification

In order to develop a concept that fulfills all potential specifications by customers and during manufacturing, system specifications are set. These aim to further focus the scope of the concepts, and set goals or targets during the following concept development. The target specification follows the method proposed by [3], which lets them originate from the customer needs. After translating the customer needs into metrics, an important target specification can be written. Other methods to create a target specifications are shown by [35].

In order to limit the scope of this project, only a select number of metrics that are directly relevant to the concept generation receive target specifications. It has no use setting careful target specifications for e.g. need 15, *Has a low manufacturing cost*, as no accurate predictions can be made at this stage. These less critical needs have therefore not received any target metrics. It is, however, still a focus to minimize these factors as much as possible. This applies to the weight metric, number 15, as well. The target metrics are shown in table 3.2, and some are discussed further in section 3.5. The metrics have received two target values, one *Optimal* and one *Acceptable*. The acceptable values show that the general targets have been reached by the system, but that the corresponding needs can be further improved by reaching the optimal values.

3.5 System Requirements

The system requirements for the P4 48V disconnect system are summarized as:

1. Reduce mechanical losses as much as possible when system is inactive
2. Disconnect at threshold speed to avoid EM damage
3. Disconnect in specified failure cases:
   (a) If the EM detects an internal failure and enters safe state
   (b) If the disconnect actuator is in-operable due to internal or external failure
4. Connect when the system should be active. Fast connection is required due to:
   (a) Boosting to support ICE
   (b) Brake energy recuperation

Requirement 2, *Disconnect at threshold speed to avoid EM damage*, is set by the EM specification. The EM assumed to be used for this system has a maximum speed of approximately 20 000 rpm, which using a standard tire circumference D_{tire} and an assumed gearing U of 25, gives a vehicle speed of

$$V_{max} = \frac{V_{max,EM}}{60} UD_{tire} \cdot 3.6 = \frac{20000}{60} \cdot 25 \cdot 1.95 \cdot 3.6 \approx 90 km/h$$

In addition to the system requirements, the system must hit the specified metric targets. Furthermore, assumptions and values for calculations and system design are specified by BorgWarner. It is also important to specify whether the system should be mono- or bistable.

3.5.1 Power Usage

The system will be limited to the voltage supplied by the vehicle, 12 V (14.4 V at full capacity), and the current cannot exceed the rating of the actuator or any related electronics. If the actuator requires too much power, the risk of interference or damage within or outside of the system increases as well. The potential power needed for keeping the dog clutch engaged for longer periods must be at a reasonable level. The system specification provided by BorgWarner conveys that the maximum mean effect should be no more than 15 W and no more than 200 W during system activation. A maximum current of 15 A should be used during activation.

3.5.2 Torque Requirements

Specifications on what torques and forces should be expected on the system were supplied by BorgWarner. The nominal values are used for the general performance calculations, in other words the disconnect operation and other forces during normal system use.

The torque on the powertrain can be configured in two different ways, one is a three-part gear train, and the other a two-part gear train. The gear ratios are 25 and 15 respectively for configurations one and two. Configuration one is used for this report. Gear ratios going through the differential are assumed to be one. Table 2.1 specifically shows the architectures with case two, but the dog clutch design stays the same for case one. The specifications are shown in table 3.3.

Table 3.3 Torque specifications.

Torques [Nm]	**Value**	**Unit**
Max. EM torque	80	
ASC torque [T_{ASC}]	5.6	
EM drag [$T_{drag,EM}$]	0.6	
Intermediate EM axle 1 drag [$T_{int,1}$]	0.1	
Intermediate EM axle 2 drag [$T_{int,2}$]	0.1	
Differential drag [T_{diff}]	1.0	

Note that the ASC torque is only the torque from the engine and does not take into account gear ratios or losses. The torque experienced by the dog clutch will, for most architectures, be higher.

3.5.3 Transmission Drag

There is always drag in vehicle transmissions due to contact- and splash losses in gears and oil suspension. These values have been specified in table 3.3, but are approximate values which do not necessarily reflect reality. These approximations are made for dimensioning of the system. The torque loss must be calculated for different positions along with the drivetrain, using the values above. For example, a disconnect system placed right at the EM will only experience the drag of the EM (with the addition of ASC, T_{ASC}, for those cases). This is demonstrated in equation 3.1. If the DC system is moved downwards along the drivetrain, the losses are added to the required disconnect torque as these become relevant. Torque at the first and second intermediate axes (for the case of a three-step gearing) are calculated according to equations 3.2 and 3.3. The total torque for the output from the differential is calculated according to equation 3.4. Keep in mind that all losses, including the non-drag ASC torque, are acting in the same direction. Equations 3.1-3.4 regarding drag losses were provided by BorgWarner.

$$T_{DC} = T_{drag,EM} + T_{ASC} \tag{3.1}$$

$$T_{DC} = (T_{drag,EM} + T_{ASC})U_1 + T_{int,1} \tag{3.2}$$

$$T_{DC} = (T_{drag,EM} + T_{ASC})U_1U_2 + T_{int,1}U_2 + T_{int,2} \tag{3.3}$$

$$T_{DC} = (T_{drag,EM} + T_{ASC})U_1U_2U_3 + T_{int,1}U_2U_3 + T_{int,2}U_3 \tag{3.4}$$

3.5.4 Connection Time

Reasonable connection times observed from competitive products are as low as 300 ms according to specifications from manufacturers as well as discussions with our supervisor. [41]

Connection time target specifications provided by BorgWarner are shown in table 3.4. The target times are applicable for temperatures over 0°celsius, as oil viscosity changes at lower temperatures. [46]

The connection time can be further divided into sub-times for example time for mechanical movement of the components and time for the ECU and signals to respond. These sub-times are shown in figure 3.2.

3.5.5 Sensor Requirement

Some customers directly specify that a sensor is required in the system, which detects the dog clutch position independently of any actuator or torque measurements on the

Table 3.4 Specification of operation times.

Operation	Time [ms]
Normal connect	100
Monostable disconnect	100
ASC disconnect	500

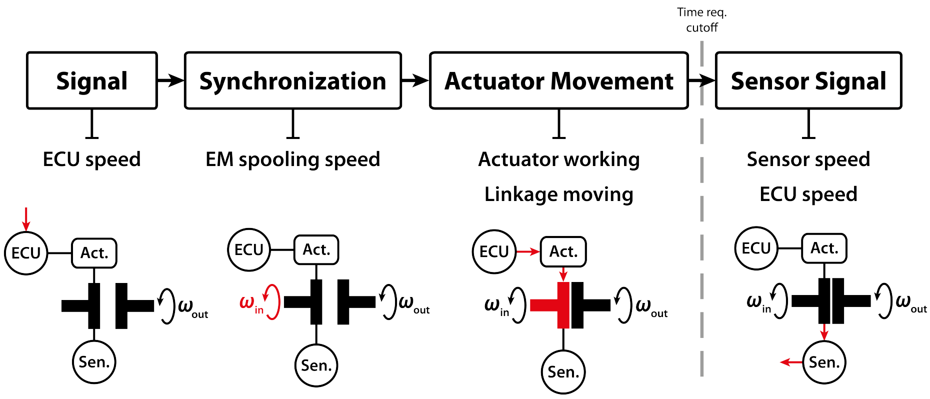


Figure 3.2: Proposed time division for the disconnect system.

axles. An early idea to find out if the dog clutch had successfully connected was to measure the rpm of the engine and the wheels by themselves, and therefore knowing if the connection was completed depending on how the wheel rpm responded to an increase in engine rpm. However, due to safety reasons and redundancy of sensor data, this was not considered a sufficient solution.

An example of needing additional sensors for position data is the use of some hydraulic pumps, which have the advantage of being constantly spooled, though this removes any possibility of knowing the piston position via the electric motor. On the other hand, a solenoid could utilize a built-in sensor that would signal a successful connect or disconnect. There might still be a need for a sensor at a closer position to the dog clutch in some cases, which is discussed during later stages of the concept development.

3.5.6 Maintenance

System is intended to be sealed for life with no service points, which means the system will not have to be opened for the refilling of oil etc.

3.5.7 Failure Modes

The failure modes mentioned in section 3.5 are specified further here. Several failure situations must be handled by the disconnect system. The first is the aforementioned ASC (see section 2.4.2), which is a result of the EM experiencing some kind of internal failure. A disconnect must also be able to happen during an actuator failure, which means that the system should operate without any signal or power input. This could be accomplished by a mono-stable system specification.

There are two main failure modes which have to be observed:

1. If the EM detects an internal failure and enters safe state
    a) Below the corner speed
    b) Above the corner speed
2. If the disconnect actuator is in-operable due to internal or external failure

The cornering speed implicates a clear difference in failure modes 1a) and 1b). A) being below the cornering speed, means that the phase currents will be lowered to zero and the EM should be disconnected while it is under no torque at all. b) implies the use of ASC to disarm any harmful back currents into electronics or batteries, which complicates the failure mode as the ASC results in previously discussed torque (section 2.4.2).

There are of course a potential plethora of additional potential failure modes, but these are outside of the scope of this project and will not be discussed further. These include failure of drive axles or differentials, failure of main ECU, ICE breakdowns. Furthermore, the chance of two failures occurring simultaneously will not be taken into account, as it brings unnecessary complexity into the project at the early concept stages. The ability of the concepts to handle the failure modes will be examined and used to evaluate the concepts accordingly.

Chapter 4

System Functions and Subsystem Division

The disconnect system is divided into proposed sub-systems and system functions to facilitate design choices and actuator technology. The result is a more clear system with distinct divisions between components.

4.1 System Functions

For most physical systems, there are three types of flow: material flow, energy flow and signal flow. Material flow means any material going through the system, e.g. empty bottles being transported on a conveyor belt. Energy flow in this system would then be the electricity supplied to the electric motors driving the conveyor belt. The signal flow might be a laser sensor being tripped when a bottle passes, which activates a nozzle filling the bottle with some liquid product. There are usually a few different ways to divide systems into the respective flows, and thought must be put into making sure the chosen model sufficiently represents the actual system.

The process can also be divided into certain sub-processes, which all transform the so-called operand (in this case, the energy used) into different states. Inputs are the operand, some type of co-input (rpm, signals, etc.), the human user, the technical system, and further operators such as information availability and quality, control, and environmental factors. See figure 4.1. Environmental operators include physical factors such as temperature and contamination (dust, wear particles), as well as psychological, social and financial factors. These final three factors are generally about the state of the user and society, which will influence the usage and quality of the process. A human user stressed about poor roads can make drastic decisions that might endanger the technical system. Poor financial conditions might lower the quality of critical components, or neglect regular inspections and repairs. [47]

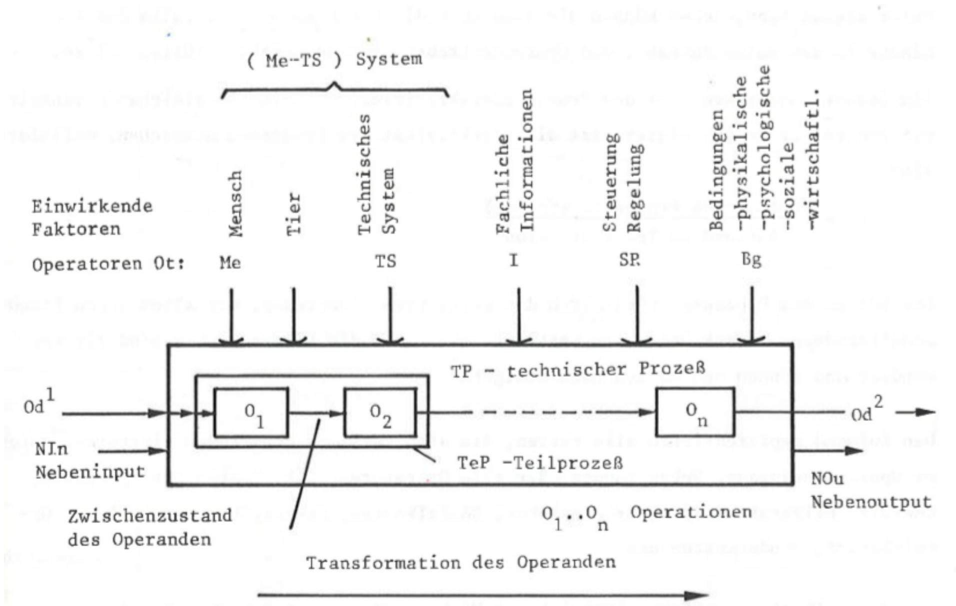


Figure 4.1: A general model of a technical process. [48]

It is important to note that there is a critical distinction between the inputs (operand, Op1) and co-inputs (NIn), and the input from any operators. The operands represent the object that is transformed (transported, treated, etc.) and the input of the operands represent the conditions which influence the quality after the subprocesses (TeP). [47]

Any process, according to the model proposed by Hubka, is characterized by the quality of the outgoing operand (Op2). In this case, one quality of the operand might be the total connecting time. A subprocess might lower this quality, e.g. a linkage flexing and thus providing lower force. A degradation will then be noticeable in Op2. By evaluating the cause and severity of quality degradation, different concepts can be ranked and a qualified decision can be made. The quality can then shown with a comparison of the cost of the system. [47]

For this project mainly the separation and distinction of flows are of interest, as it can make the subsystems more clear. The subsystem theory is presented in section 4.2 below. Figure 4.2 shows the proposed processes and flows of the disconnect system.

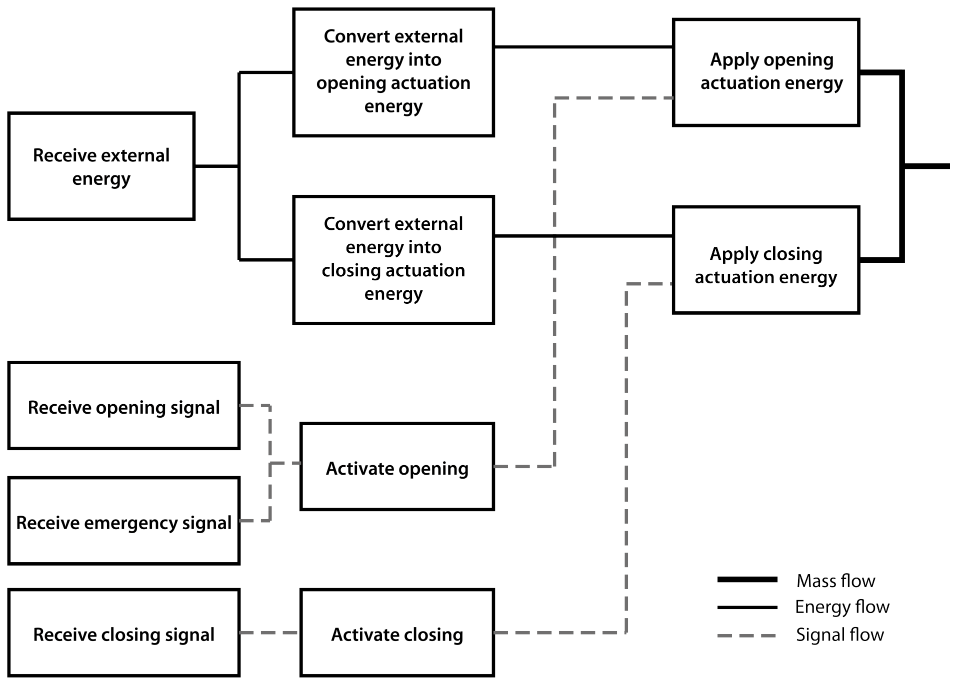


Figure 4.2: System flows and functions, with the corresponding flows marked by lines.

4.2 Subsystems

The system was divided into different subsystems to increase clarity during the concept generation. Heavy focus was put on the definition of the design realm, as some subsystems were assumed to already be complete. A proposed subsystem architecture is shown in figure 4.3, using a solenoid for force generation as an example. The packaging of the product is assumed to encompass all subsystems, as the axle subsystem passes through the main structure. The dog clutch is a complete part, and other than choosing the specific type (see section 2.6), it is removed from any further design and the main design realm. The dog clutch subsystem is divided into two parts, the actuator and the axle sides. These are the parts that connect the electric motor with the wheels during system activation, and effectively represent in- and output for the dog clutch. Subsystems within the design area are divided into the energy source and the actuator, which can be further divided into smaller systems. Important is that these divisions depend slightly on the actuator or energy source used for the concept in question. If a solenoid is used, some of the subsystems could be a coil and a cylinder instead of e.g. a piston and cylinder for a hydraulic actuator. This decision was made as it is very difficult to divide all types of actuators into an equal number of specific subsystems.

By performing this division, it is easier to get an overview of the whole system, and change out or vary subsystems as necessary during the design process. Mainly the distinction of a separate actuator subsystem (or subsystems) was important during the project. The actuator subsystem was scored independently of the concepts, which facilitated actuator changes and correct scoring during later concept evaluations.

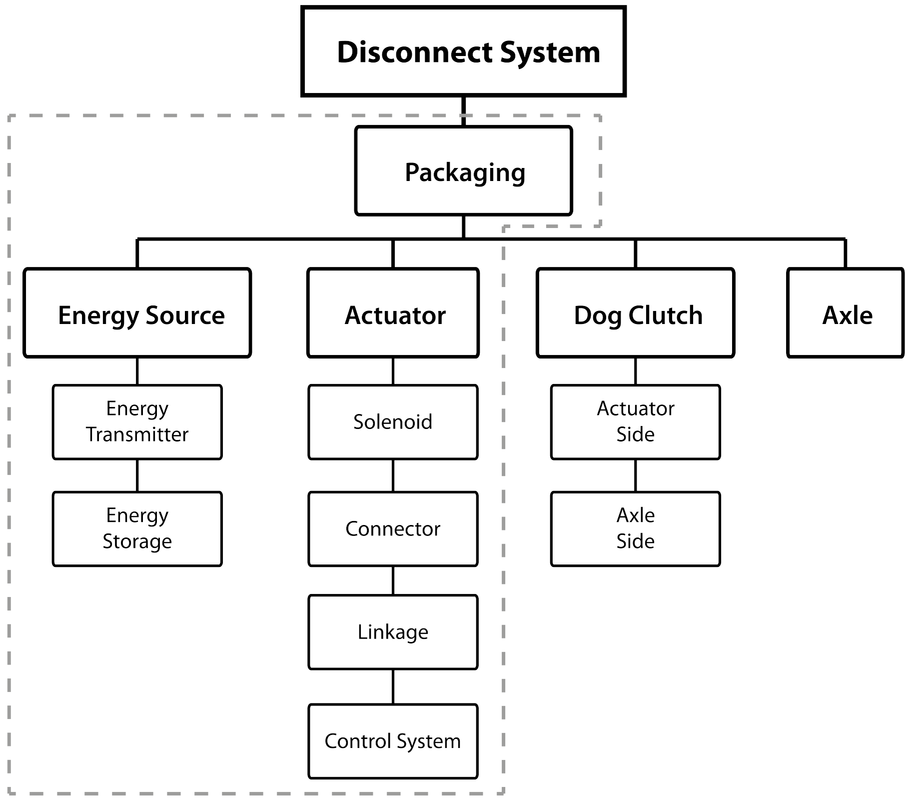


Figure 4.3: A proposed subsystem setup using a solenoid. The design space is marked in gray.

Chapter 5

Architecture Scoring

This chapter explains the architecture scoring that was performed in order to evaluate the potential positions for a disconnect system. It explains the method used, setting criteria, weighting and finally the scoring to determine the best position along the drivetrain.

5.1 Criterias

To limit the number of concepts for the disconnect clutch, it was decided to first score the different possible placements of the clutch, i.e. the architecture. Depending on where the disconnect is located, different problems and challenges might arise, as well as some placements having distinct advantages. There are many possible locations ranging from disconnect systems in the wheel hubs to disconnecting the rotor of the electric motor directly at the beginning of the gearbox. Section 2.7 contains an explanation of the different architectures.

One of the main compromises during driveline design is taking into account balance and torque. The closer a rotating component is placed to the electrical motor, the higher rpm it will be expected to handle. Higher rotational speeds mean a higher tolerance for balance in the part, as a small deviations of the center of gravity can result in damage or excessive noise. The opposite principle applies for torque, as reduction gearing in the drivetrain means equally increased torque the further out along the drivetrain the system is moved. Thus, components further out are expected to survive higher loads.

Another consideration is the amount of rotating components the system will disconnect. For example, if the drivetrain is disconnected at the hubs by the driven wheel (explained in section 2.7.3) at the same time as the electric motor stops rotating, no gears or shafts will move during driving. If just the propeller shaft is disconnected (explained in section 2.7.1), all subsequent gears and shafts will rotate. The problem with large numbers of components rotating is the creation of parasitic and geometric losses. Parasitic losses occur due to friction between gears etc. when the surfaces slide against one another. Geometric losses occur due to all components having a certain moment of inertia and require varying amounts of force to start turning. The outcome is a lower efficiency since the ICE has to work harder to spin the drivetrain. This is an important criterion, especially regarding the environment, but also for the overall performance of the car and durability of the components. A study showed that for an AWD-vehicle (4-wheel drive with combustion engine), disconnecting a part of the driveline that is not driven can provide a significant reduction in fuel consumption. [49] [50] The last criteria to consider is the cost and complexity of the system. A more complex system must not always be more costly to produce, but can be harder to design, assemble, repair, or more likely to break down. The scope of this thesis is limited to the concept and design phases, and therefore cost and manufacturing will not be considered in detail. It is always good to consider cost disparities between different concepts though, since many customers focus heavily on cost and simple designs which still fulfill all their needs.

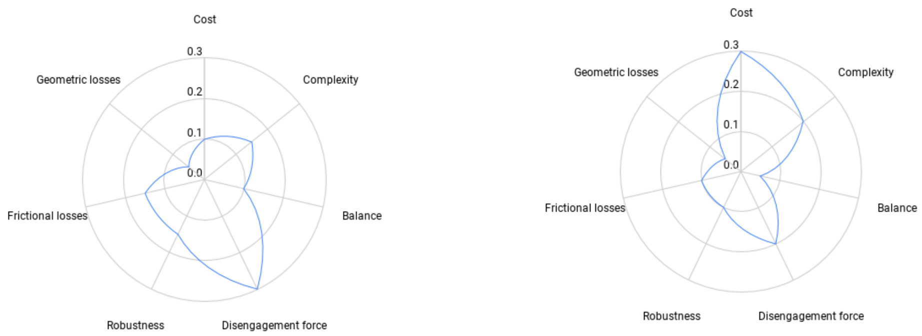
All criteria were discussed with our supervisors and revised multiple times. The criteria are shown in table 5.1.

Table 5.1 Architecture screening criteria and weighting.

Criteria	**Weight 1**	**Weight 2**
1. Cost	0.1	0.3
2. Complexity	0.15	0.2
3. Balance	0.1	0.05
4. Disengagement force	0.3	0.2
5. Robustness	0.15	0.1
6. Frictional losses	0.15	0.1
7. Geometric losses	0.05	0.05

5.2 Weighting

The different categories were weighted depending on the perceived importance (mainly cost) for the end customer, and the potential challenges. Two different philosophies for specifying weights were employed, one focused on performance and the other one on cost. The performance weighting has high values for e.g. *Disengagement Force* and *Robustness* to illustrate the importance of a lasting drivetrain and well-performing system. When focusing on cost, weights were predominantly placed on *Cost* and *Complexity*, see figure 5.1 for a visualization and table 5.1 for values of weight.



(a) Weighting with focus on performance.

(b) Weighting with focus on cost.

Figure 5.1: Two different weighting alternatives for the architecture scoring.

5.3 Scoring

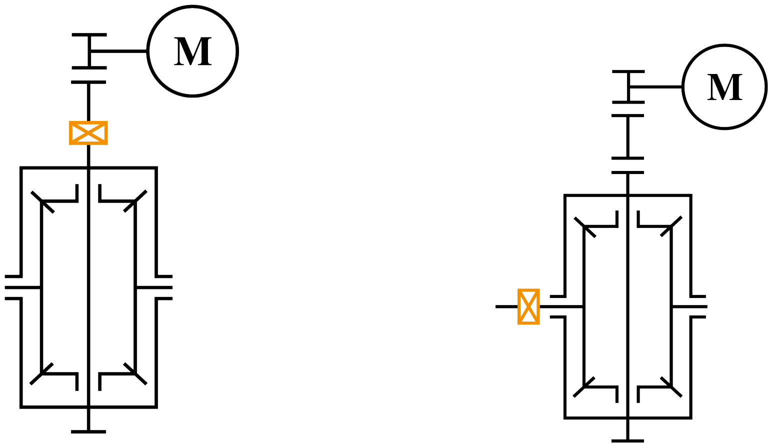
A scale of 1-5 was used during the first concept scoring, where 1 denotes very poor performance and 5 very good performance. Each category received a reference depending on which disconnect placement was the most applicable for that category. For example, the ‘Balance’ category received a concept placed in the ’middle’ of the rear drivetrain. This then facilitated the use of all five grades, as choosing a poor reference for a category can diminish the scale for the other concepts (if a poor concept becomes the reference, all subsequent but better concepts can only receive 4-5 points, effectively decreasing the scale).

5.4 Architecture Scoring Results

The screening showed that two placements of the disconnect were favorable for both categories, i.e. cost and performance. The first was the propeller shaft DC, and the second the single side axle DC. See table 5.2 for the results of both scorings and figure 5.2 for the architectures.

Table 5.2 Architecture scoring results, with ranks corresponding to both weightings.

Architecture	Score 1	Score 2	Rank 1	Rank 2
Prop Shaft DC	2.7	2.9	2	2
Internal Diff DC	2.4	2.6	3	3
Wheel Hub DC	1.9	2.35	5	5
Single Axle DC	2.95	2.95	1	1
Twin Axle DC	2.1	2.5	4	4



(a) The propeller shaft DC architecture.

(b) The side axle DC architecture.

Figure 5.2: The two chosen architectures which will be used for concept generation.

Chapter 6

Alternative Technologies

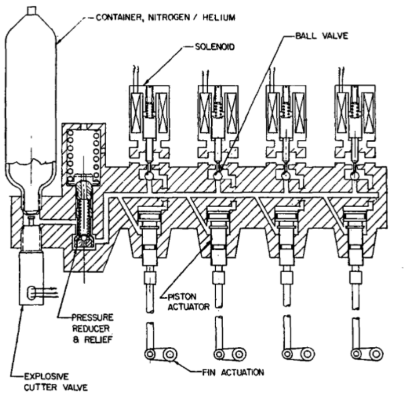
In this chapter products that are similar to the disconnect system developed will be looked into. This was done to get information and inspiration for concept generation performed later in the report and to make sure no patent issues would arise later on.

To gather information and inspiration for the concept generation, alternative technologies and mechanisms were looked into following the concept generation theory provided by Ulrich & Eppinger. These can include almost every other technology, as long as there are similarities with the actual design problem. One approach was to scale up and scale down to similar bigger or smaller products. The focus was put on finding alternative technologies which did not directly fulfill the same task as our disconnect system, but to find new and interesting ways or mechanisms of actuation or energy usage. There were some difficulties finding relevant information and drawings as many of the products are protected. The main alternative technologies looked at are described here.

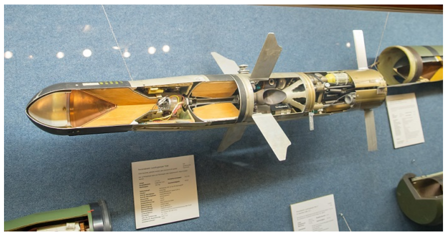
Automotive gearboxes and differentials - As gearboxes and differentials are closely related to the disconnect system it can share many issues. It is known that many gearboxes utilize so-called fork shifters to move dog clutches in the gear train. Self-locking and locking differentials are also interesting as it can have complex but efficient ways to in some way limit the torque in- or output of the differential. One example is a Audi Quattro self-locking differential which uses helical gear axial resultant forces to compress clutch discs. [51]

High-voltage disconnectors - High-voltage disconnectors, such as the ones designed by ABB, use different constructions incorporating springs along with electric motors [52]. This enables them to quickly disconnect if dangerously high voltages are detected, while later being reset manually or by an EM.

Military application mechanisms - Release mechanisms used for military or defense applications are looked into as these systems have high requirements regarding reliability and fail-safes. The actuators are often quick and precise, with high expectations for their lifetime. Gas-powered wing actuators for TOW rockets (*Tube-launched, Optically tracked, Wire-guided*) proved to be interesting, as the requirements for fast actuation times are extreme. When the rocket is launched the fins guide the rocket before impact. This is made possible by a high-pressure gas container that releases gas to actuator pistons through special valves, see figure 6.1a. Reading about the development of the system was very interesting and gave some insight into handling short actuation time frames. [53]



(a) The fin actuator of the TOW system, with one separate valve for each fin.[54]



(b) Cutaway with fins and gas container visible.[55]

Figure 6.1: The TOW missile, steered by gas-actuated fins.

Railway couplings - Railway couplings have been around for many decades, in varying designs. One of these is the so called *Janney Coupler*, which can couple rolling stock (wagons) automatically. The principle is simple, a part of the knuckle rotates and can be locked in place by a pin at its closed position [56]. The idea is a large force (rolling stock tearing at the coupling) being controlled by a small, stationary device (the pin). This gave some ideas for lowering the energy usage of the disconnect system during use.

Aerospace applications - An example of disconnect systems in the aerospace industry is when hydraulic pumps for controlling the wing elements have to be disconnected. If the primary pump is damaged it needs to be decoupled before burning out. One solution to this is to use an eutectic material which is sensitive to heat. The heat are generated when the pump warms up and melts the eutectic material and then disconnects the actuator, protecting the pump itself [57].

Chapter 7

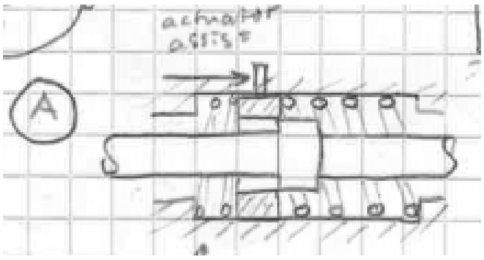
Concept Generation

In this chapter, the method of how the concepts were invented are explained. It then proceeds with presenting some of the most important concepts generated.

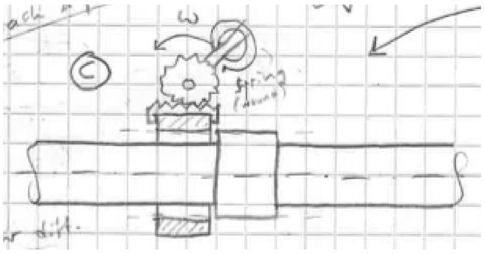
When the general placement and layout of the system had been established, the first concept generation process was started. Concepts were generated by choosing actuators of specific type from section 2.6 and incorporate it with geometry for mechanical transmission of the power. Concepts that were of a more conventional type was generated as well as concepts with new and creative ideas. Concepts that was considered not realizable were also included to pass on ideas and subfunctions to the other team member. At first, each team member worked on their concepts and ideas, searched internally based on the knowledge they already possessed. Then a brainstorming event was held to combine these ideas and share the knowledge. This was performed together with the supervisors.

Doing this created a few other concepts. At last, information was searched externally. In this case, staff from BorgWarner working with similar products, the aforementioned alternative technologies and patents, etc. This was done according to Ulrich & Eppingers methods for concept generation [3]. Some of the concepts were simply developed in CAD-software to further understand the design and functionality when a simple sketch was not enough. This process helped visualize any obvious problems or issues with the concept, as well as help get more of an understanding of packaging the system as a whole might require.

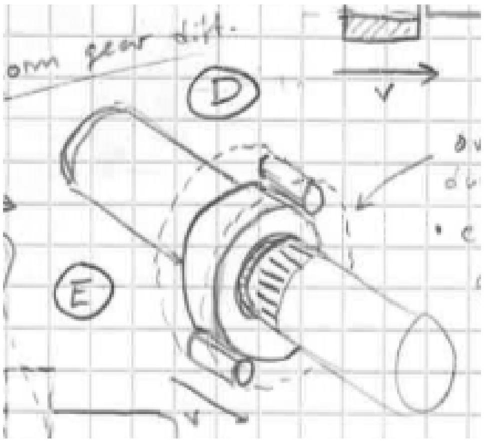
Some of the first generation concepts are shown in figures 7.1 and 7.2. These focus both on varying the mechanical means for axial translation of the dog clutch, as well as incorporating different actuators into the design. Some show similar working principles or components, and may be combined during later concept phases. The concept generation phase resulted in 38 different concepts, which were all included in the subsequent screening process.



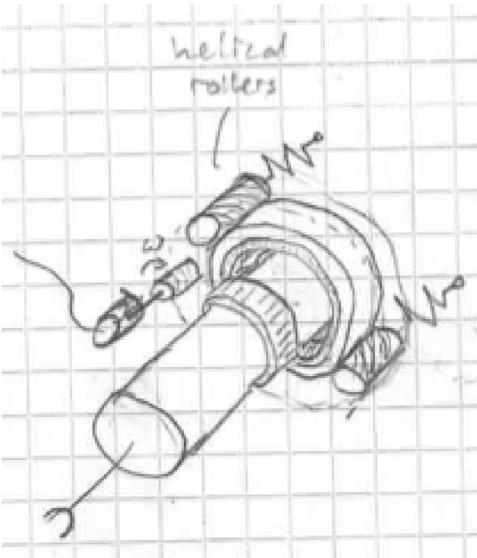
(a) Concept A - *Dual Spring*. Two springs, one assists actuator, the other disconnects dog clutch.



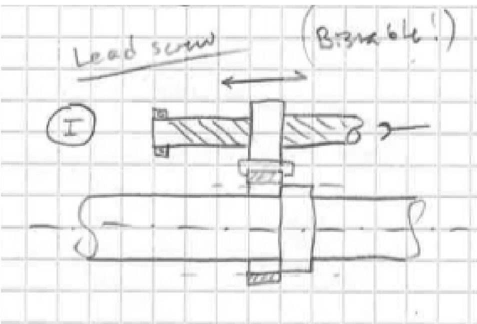
(b) Concept C - *Rack & Pinion*. Rack and pinion with EM move dog clutch sleeve. Spring on EM axle retracts when no power, forcing EM to rotate with it.



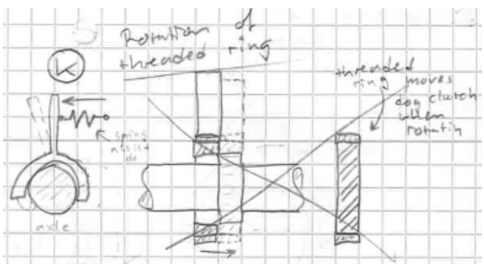
(c) Concept D - *Helical Planetary*. Helical planetary gears are moved by ringgear which results in axial force.



(d) Concept D1 - *EM with Rack & Pinion*. The dog sleeve is moved by an assembly of planetary gears.

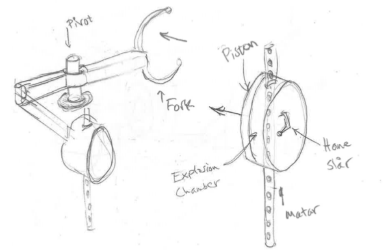


(e) Concept I - *Lead Screw Sleeve*. A lead screw moves the dog sleeve.

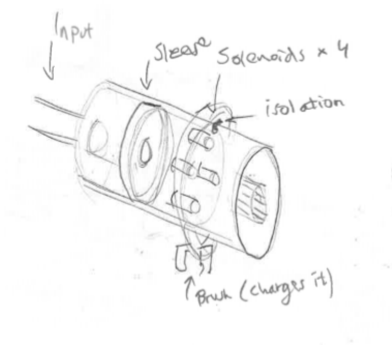


(f) Concept K - *Threaded Ring*. Fork or ring is rotated by solenoid/hydraulics etc., moving sleeve by groove inside. Spring returns when power cut.

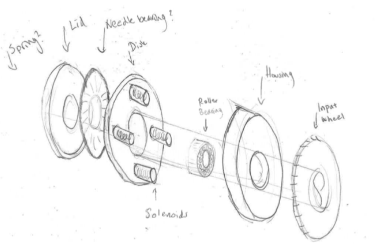
Figure 7.1: A selection of concept sketches during the concept generation phase.



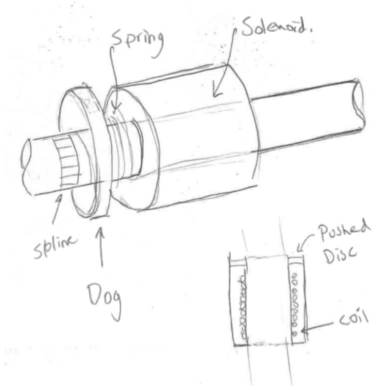
(a) Concept U - *Exploding Drum*. Explosive charges are used to accelerate the clutch into connection.



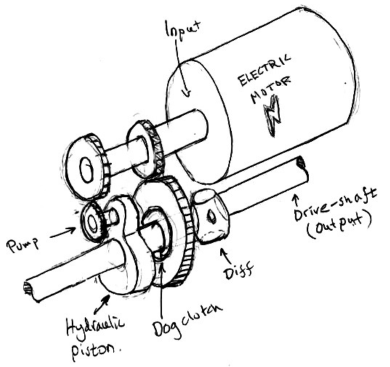
(b) Concept W - *Internal Disconnect with Solenoids*. Rotating with the driveshaft, the sleeve is engaged with the input. An electric conducting ring is wrapped around with a brush to conduct energy.



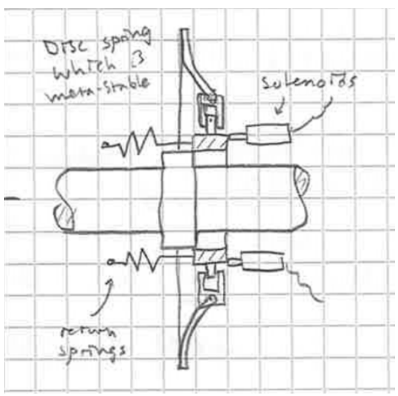
(c) Concept Y - *Multiple Solenoids*. Multiple solenoids placed around the axle are combined to push the sleeve.



(d) Concept AA - *Solenoid on Sleeve*. A large solenoid pushes the sleeve, wrapped around the axle. Spring return.



(e) Concept AD - *EM Generated Hydraulic Piston*. The EM also drives the hydraulic pump used to move the dog sleeve.



(f) Concept AF - *Disc Spring Jumper*. A bistable disc spring is used to move the sleeve.

Figure 7.2: Additional concept sketches from the concept generation phase.

Chapter 8

Concept Screening

This chapter explains the initial screening performed on all the concepts generated from concept generation. It involves the criteria, results and the screening references used to compare against.

The screening was performed to narrow the number of concepts and make sure the ones that pass would fulfill the needs of the final product. It also contributes to improvement for the more promising concepts. The concept screening is based on a method developed by Stuart Pugh in year 1980 [3].

The screening focused heavily on the most important and easily tangible criteria. Most of the concepts are at an early stage of development. Therefore, it has no benefit to going into e.g. detailed connection times and material stresses, but general estimations can usually be made. No weighting was applied, but instead, according to the methodology of Ulrich and Eppinger, the screening was performed rating with +, - or 0 which means better than, worse than or the same as the reference product. The reference product used is briefly explained in section 8.2 [3].

The screening was first made individual and then it was discussed were differences had aroused and a final result was made, including opinions from the supervisor. The criteria that were used were also discussed. How the relative importance and how every concept could be judged against them. The final selection of criteria is shown in table 8.1. The screening criteria from chapter 5 were considered and redefined to fulfill the purpose of the current screening. When the screening was done, reflection was made to remove or add concepts close to the screening cut line. This was done to not miss any critical judgment or weakness in the criteria. A combination of concept and revision of concepts could also be made.

Table 8.1 Concept screening criteria.

Criteria	Comment
1. Complexity	Overall system complexity.
2. Packaging	Ability to be designed compact.
3. Durability	Overall robustness of system.
4. Efficiency	Energy usage peak/while connected.
5. Stability	Ability to be designed monostable.
6. Performance	Potential actuation and connecting speed.
7. Cost	Cost of manuf. and components.

8.1 Selection Criteria

The criteria used during the first screening were developed from the architecture screening. Some changes were made and the criteria are summarized in table 8.1.

Complexity, Packaging, Durability and Cost is self-explanatory and affects the overall quality of the product. Stability is set since the desired function from the specification is to make the disconnect mono-stable. Different concepts can be harder or easier to make mono-stable. Performance includes both estimated actuated time and engagement/disengagement force.

8.2 Concept Screening Reference

An existing concept of a disconnect module was chosen as a reference during this screening, to facilitate scoring the generated concepts. Its mechanical workings are well known and utilize common and relatively simple principles that can be evaluated. There is also data available on the system for potential benchmarking within BorgWarner.

8.3 Concept Screening Results

The concepts were screened using the criteria presented in chapter 8, by giving each concept one of three values for each criterion: +, 0 or -. These denoting whether the concept was better, equivalent, or worse for the specific criteria than the chosen reference (see section 8.2). No weighting of the criteria was applied at this stage.

Table 8.2 shows some results of the screening, with *Y* denoting accepted concepts, *N* denoting unsuitable and removed concepts, *C* showing the concept being combined with others (clarified under *Comments*), and *R* meaning a revision of the concept is preferred. Revision implicates the concept being promising but either requires further work, could be applied differently, or includes some idea that should not be removed from the concept library. The concepts, refined and selected for further development can be seen in table 8.3.

Table 8.2 Results of the first concept screening.

ID	Concept	Net Score	Rank	Action	Comment
A.	Dual Spring	0	5	R	Create mechanism.
B.	**Groove Mover**	2	3	**Y**	
C.	**Rack & Pinion**	0	5	C	Comb. with P & X.
D.	**Helical Planetary**	2	3	C	Comb. with D1.
D1.	**Planetary Lead Screw**	2	3	C	Comb. with D.
I.	Lead Screw Sleeve	2	3	N	
K.	Threaded Ring	1	4	N	
N.	**Electromagnetic Sleeve**	1	4	**Y**	
P.	**EM with Rack & Pinion**	-3	8	C	Comb. with C & X.
U.	Exploding Drum	-4	9	N	
V.	**GenVI Hydro Sleeve**	4	1	**Y**	
W.	**Internal DC with Solenoids**	0	5	C	Comb. with Y.
X.	**Lifted EM**	-3	8	C	Comb. with C & P.
Y.	**Multiple Solenoids**	0	5	C	Comb. with W
AA.	Solenoid on Sleeve	2	3	R	Check viability.
AD.	EM Gen. Hydraulic Piston	0	5	N	
AF.	**Disc Spring Jumper**	3	2	**Y**	

8.3.1 Combined Concepts

Some concepts, noted by a *C* in table 8.2, were either not directly seen as viable enough for further development and/or could easily be combined with other concepts.

- Concepts C, P and X were combined into concept CPX. Concepts C and P were very similar, and were thought to work favourably with the principle of concept X.
- Concepts D and D1 were not directly combined into a new concept but were seen as two versions of the same general idea. Both utilize planetary gearing and could use similar working principles.
- Concepts W and Y were combined into concept WY. The large solenoid of concept W was removed in favour of smaller ones, but keeping the compact sleeve layout.

Table 8.3 shows the combined results of D2, CPX and WY. Concepts C, D, D1, W and Y are shown in figures 7.1 and 7.2.

8.3.2 Revised Concepts

A few concepts needed extra work or some mechanical principle in them could be used for other concepts. Instead of combining or removing the concepts, these were revised (noted by a *R* in table 8.2).

- Concept A had a interesting mechanism which could be used for other concepts, but the concept as a whole was rejected.
- Concept AA was intriguing but could not directly be verified to work as intended, and needed further research.

The two concepts going into revision are shown in figures 7.1 and 7.2.

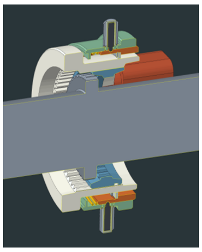
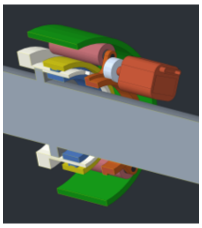
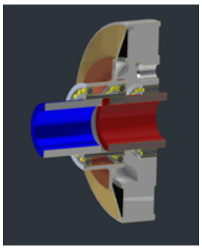
8.3.3 Removed Concepts

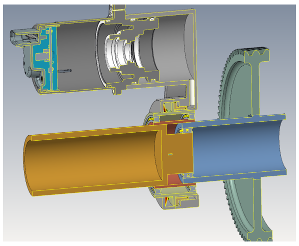
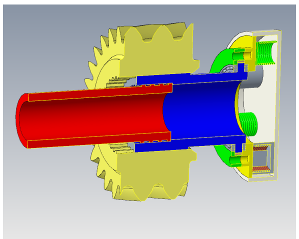
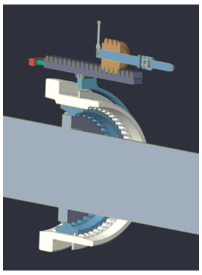
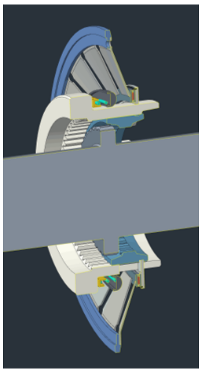
Most concepts were directly removed due to low scores in the screening, and will not be developed further. Some are shown in table 8.2, noted by a *N*.

8.3.4 Accepted Concepts

The accepted concepts, receiving high scores, are listed in table 8.2 (noted by a *Y*) and shown in table 8.3. These are discussed further in chapter 10.

Table 8.3 Concept range after screening.

Concept	CAD Sketch	Description
B. Groove Mover		Rotating sleeve with groove is moved by two solenoid-actuated pins. EM moves pin-ring. If no power pins are pulled and groovesleeve retracts. It is then reset by reset grooves.
D2. Planetary DC		A planetary gearset is used to move a dog sleeve, either with only gears or using lead screws.
N. Electromagnetic Sleeve		Electromagnetic actuator pulls the sleeve with the help of spreading magnetism through the disc spring. The disc spring returns it when released.

V. GenVI Hydro Sleeve		Uses the GenVI hydraulic pump to build up pressure in the house that moves the piston. This engages the sleeve that connects the two drive shafts. Coil spring returns it.
WY. Multiple Solenoids		Mounted on EM-shaft, multiple solenoids actuate on sleeve. The multiple springs returns it.
CPX. Lifted Lead Screw		A lead screw actuated by an EM moves the dog sleeve, and is lifted out of connection with the rack for disconnection.
AF. Disc Spring Jumper		Metastable disc spring is pushed by solenoid which puts it into connected position. If solenoid power is then turned off, disc spring jumps back to disconnect with help from return springs.

Chapter 9

Actuator Scoring

In order to know how well each type of actuator performed, this chapter explains how the actuator scoring was performed. It involves metrics, criteria and results for deciding the most suitable technologies.

As the specific actuator technology used in each concept was not fully specified for most ideas, scoring of the actuator technologies themselves was performed as well. These were evaluated with the criteria shown in table 9.1, which were discussed further with our supervisors.

9.1 Baseline Actuator Specifications

To be able to easily differentiate the actuators, as well as to compare them, general specifications were made. These use both internal specifications about common actuators used by BorgWarner, and specifications from basic actuators used in other industrial applications. The hydraulic actuator used is the GenVI hydraulic actuator developed and used by BorgWarner [58]. The solenoid is used in a similar project and could be used for this application as well. Therefore, data for it was readily available. The electric and stepper motors use base models found in manufacturer catalogs. Models were chosen with a rough estimation of torque and required forces in mind.

9.2 Actuator Scoring Metrics

The metrics and values used are shown in table 9.3. It is important to note that no actuator choices were made at this point, and the metrics were only used to get a broader understanding of the strengths and weaknesses of each actuator type to give them approximate ratings. This was instead used as a tool for the final concept scoring seen in chapter 10.

Table 9.1 Actuator scoring criteria and subcriteria, with weighting.

Criteria	**Subcriteria**	**Weight**
1. Force		0.2
2. Speed		0.2
3. Packaging	3.1 Actuator size	0.1
	3.2 Power supply size	0.03
4. Flexibility		0.05
5. Power	5.1 Power use peak	0.04
	5.2 Power use connected	0.13
6. Reliability		0
7. Controllability		0.05
8. Weight		0.04
9. Cost		0.06
10. Contamination		0.1
	Total	1.0

The weighting of the criteria was focused on the force and speed of the actuators, with additional focus on power consumption and mass. It was produced by the students but evaluated together with supervisors at BorgWarner. Power was further divided into two subcriteria, *Power use peak* and *Power use connected*. The first specifies the power used during connection of the dog clutch and the other the average power draw while keeping the clutch connected. The *Reliability* criteria was removed after some consideration, and it is discussed further in section 12.1.

9.3 Actuator Scoring Results

Table 9.2 shows the result of the actuator scoring. The hydraulic actuator received a relatively high score, with the electromagnet and the electric motor also being graded high. The stepper motor received a low score and was not seen as viable for any concept implementation.

Table 9.2 Actuator scoring results.

Actuator	**Score**	**Rank**	**Points**	**Comment**
A. Hydraulic	3.62	1	5	
B. Pneumatic	3.01	5	1	
C. Electric Motor	3.17	3	3	Can use built-in brake.
D. Solenoid	3.24	2	4	
E. Stepper Motor	2.99	6	-	Removed following low rank.
F. Electromagnet	3.09	4	2	Loses points due to distance limits.

Table 9.3 Baseline actuator specifications.

Actuator Type	Name	Manuf.	Weight [g]	Dimens. [mm]	Peak Power [W]	Voltage [V]	Peak Curr. [A]	Peak Torq./Force [Nm/F]
Stepper Motor	MIS176S35yyzz66	JVL	900	106	134	7-72		0.8 Nm
DC Motor	PDS4265	Transmotec	456	91	17.1	12	15	1.177 Nm
Solenoid			1400	60x77	20/200	35		250 N
Electromagnetic	Type 10 33111A00	Kendrion	1000	110x21	14.7/-	24		
Hydraulic	GenVI	BorgWarner	700 + cyl.	56x120				16kN at 40bar
Pneumatic	Mini Air Inflator	Oasser	450	45x56x200		12	5	4336 N

Chapter 10

Concept Scoring

In order to proceed with the best concept, it first has to be decided which is the best one. This chapter explains the method used to do so and the proceeds to the results and what concept(s) that were selected for further development.

The scoring was performed with weighted criteria. The criteria used during the first concept screening in chapter 8 were expanded upon and now include subcriteria to further explore the solution ideas. Some of the criteria were omitted or changed. See table 10.1 for the full list of criteria. These criterias were also discussed with our supervisors and were subject to many changes during the development process. The criteria were reevaluated and modified multiple times.

The scoring was, just as the screening, first done by each team member individually and then discussed and altered into a final one. Reflection of the process was then made, comparing the score each concept was given. For example, if two concepts were almost even in score or very uneven, the criteria that had weighted the most were reviewed to not miss out on anything. The supervisor was also consulted in order to cover every aspect of the process.

The different concepts were developed further before this scoring, to better see the viability and possible issues with the ideas. A specific actuator technology was also associated with every concept to facilitate scoring. These actuators are scored according to the actuator screening in chapter 9, which awarded each actuation method with a certain score.

10.1 Concept Scoring Results

Table 10.2 shows the result of the concept scoring. The 'Action' column refers to if the concept was accepted or discarded directly, or if a further discussion was needed. The discussed concepts are mentioned in section 10.1.1. The concept FL-D was included

Table 10.1 Concept scoring criteria and subcriteria.

Criteria	Subcriteria	Weight
1. Energy Source	1.1 Actuator Choice	0.2
2. Complexity	2.1 Number of mechanical components	0.04
	2.2 Number of actuators	0.1
3. Packaging	3.1 Packaging size	0.04
	3.2 Flexibility	0.04
4. Durability	4.1 Force application balance	0.06
	4.2. Contamination	0.045
	4.3 Sensitivity	0.06
5. Efficiency	5.1 Power use peak	0.07
	5.2 Power use connected	0.04
6. Stability	6.1 Ease of monostability	0.01
7. Performance	7.1 Force efficiency	0.013
	7.2 Weight	0.075
	Total	1.0

as a comparison as it is the concept used in the demo vehicle by BorgWarner. It has not been included in the ranking but would have received a rank of 3 if it was, making it a good contender for the best-scored concepts. This also shows that most of the generated concepts are comparable to the existing concept, using these specific criterias.

10.1.1 Concept Rating and Choice

Two concepts, concept AF and concept N (see table 8.3), which received good scores during the concept scoring (table 10.2) were removed due to the insufficient theoretical background being available under the time constraint, as well as no practical

Table 10.2 Scoring of the second selection of concepts. Action refers to 'Yes', 'No' or 'Discuss'.

Concept	Score	Rank	Action	Comment
B. Groove Mover	2.515	7	N	
N. Electromagnetic Sleeve	3.425	2	D	
V. GenX Hydro Sleeve	3.345	3	Y	
AF. Disc Spring Jumper	3.445	1	D	
D2. Planetary DC	3.115	5	N	
CPX. Lifted Lead Screw	2.680	6	N	
WY. Multiple Solenoid DC	3.240	4	Y	
FL-D. Demokoncept	3.390	- (3)	-	For comparison.

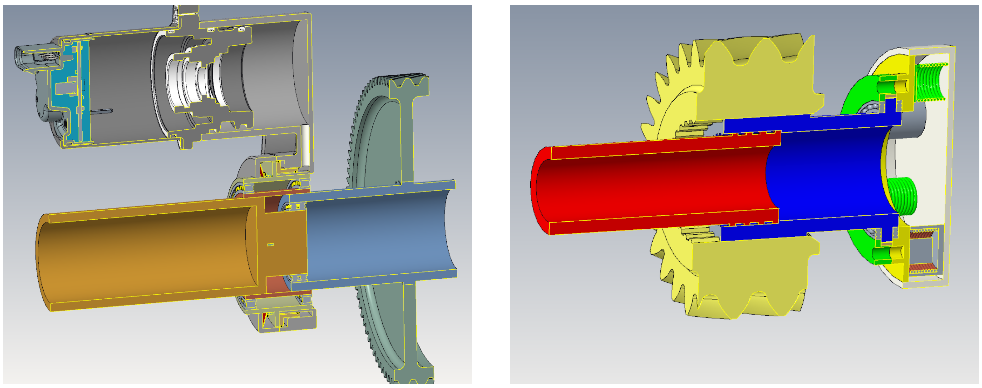
testing being viable. This led to it being difficult to evaluate if the concepts would work in real applications, as well as troubles dimensioning the actual components.

The concepts were discussed with supervisors at both LTH and BorgWarner to investigate any chances of making them more viable. BorgWarner expressed reluctance to accept these two concepts since these were less likely to arrive at a working concept within the time constraints.

10.2 Concept Selection

The concepts that were scored were revised with further development, as well as research of viability. At this stage, the concepts were confirmed to be feasible for further development. A number of concepts to proceed with were also discussed, depending on the results obtained and demands from the company.

After discussion and removal of concepts AF and N, the two concepts with the next best scores were examined, concepts V and WY. These ranked high in the scoring and involved much less risk during the continued development process. Concept V or the *GenVI Hydro Sleeve*, seen in figure 10.1a and 10.2, shows great potential by having an advantageous power density and simple working mechanism. The placement along the drivetrain is one of the output axles of the differential according to the architecture *Side Axle DC*. The other concept WY or *Multiple Solenoid DC*, seen in figure 10.1b is placed much earlier in the drivetrain and should mount on the first output pinion of the EM. This implies a large focus on balance and tolerance chains, as any vibration or radial play in the system can have adverse effects under the quick rotation of the axle. Simultaneously, the placement allows for very low disconnect forces which in turn opens up for smaller, more efficient actuators. There are also many interesting actuator choices that could work for the concept, these are discussed further in chapter 5.



(a) The GenVI Hydro Sleeve concept

(b) The Multiple Solenoids concept

Figure 10.1: Chosen concepts for further development.

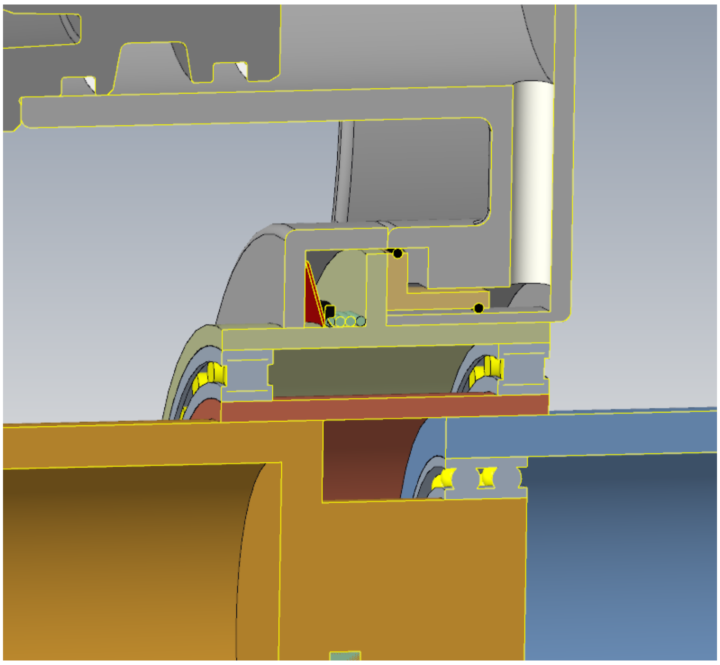


Figure 10.2: Detailed view of the GenVI Hydro Sleeve concept

10.3 Removed Concepts

All concepts ranking lower than the two chosen, V and WY, were removed from further development. This does not mean that these are necessarily discarded as bad concepts, but no more time will be spent on them during this project. Concepts N and AF were also removed from the development process.

Chapter 11

Detailed Design

This chapter explains the continued development of the concepts chosen during the selection phase. Basic calculations are performed to show the viability of design choices. The chapter ends with a full presentation of the concepts. Including the working principles, design changes made, and how calculations have been used to improve the design.

The detailed design of the chosen concepts was the next part of the development. This chapter refers mostly to the refinement and continued development of the reduced number of concepts. Included in this process is resolving major remaining problems such as feasibility, manufacturability, bearings, sealings, return springs, sensors, etc. Calculations were also performed early in this phase, as the more detailed concepts allow for greater accuracy in forces and linkages. The detailed design was done in a way that allows the systems to be adapted to different drivetrains but also allows for easy changes in the design due to different customer needs. The idea was to solve as many common problems within the projects time frame.

11.1 Calculations

The final concepts were analyzed structurally by dimension components such as bearing, piston and springs. This was performed to make sure the size of each component was in the right area and would withstand the loads introduced to the concepts. Calculations to make sure the right size of the actuator had been chosen were also made. However, further calculations should be performed in order to optimize the sizes and make sure the product does not break from fatigue or other unwanted forces in the system.

11.1.1 Disengagement Force

The different types of dog clutch use slightly varying methods for calculating the disconnect forces. The dog clutch designs may also wary between concepts due to design choices and specific loads in different parts of the drivetrain.

In order to set the right dimension for the chosen actuator and optimize it, the disengagement force was calculated. The force could be calculated using FEA-software to obtain a simulated result. However, rough hand calculations were made in order to approximate the size of the actuators and other components. This was considered enough. The disengagement force for type A can be derived from figure 11.1, resulting in equation 11.1, using the torque M_{EM} [Nm] and a gear ratio R that is provided by the EM and geartrain. The torque M_{tot} includes losses.

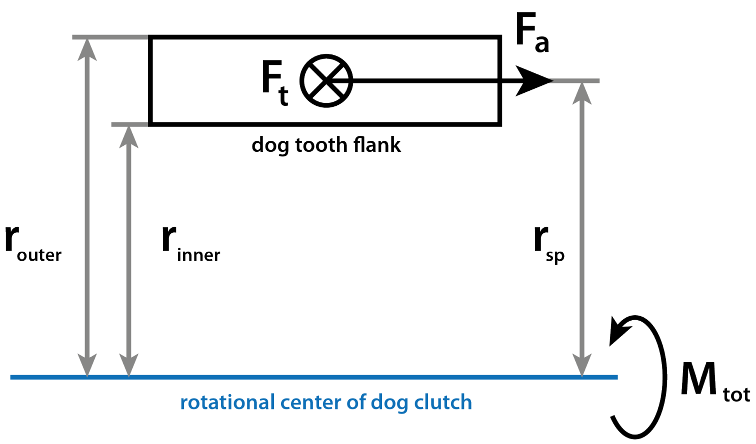


Figure 11.1: Forces acting on a type A dog clutch flank.

$$M_{tot} = M_{EM}\ R \tag{11.1}$$

For the type A dog clutch the mean radius on the tooth surface r_{sp}, where the force acts, can be calculated with equation 11.2.

$$r_{sp} = \frac{r_{outer} + r_{inner}}{2} \tag{11.2}$$

An assumed worst-case scenario friction coefficient μ for the contact surfaces is assumed in this case, but should for more accurate calculations be derived from tests of similar components. The disengagement force $F_{dc,A}$ is calculated in equation 11.3.

$$F_{dc,A} = \mu \frac{M_{tot}}{r_{sp}} \tag{11.3}$$

Type B follows largely the same method but includes the pressure angle of the spline as well, shown as α in figure 11.2. This angle is then increased by the tooth thickness angle, δ. The forces are split into two composants, F_T and F_R, while the resultant force is named F. The effective tangential angle, α_e is calculated in equation 11.7, with the module m and the number of teeth z, which are specified by spline table values, giving p, the pitch.

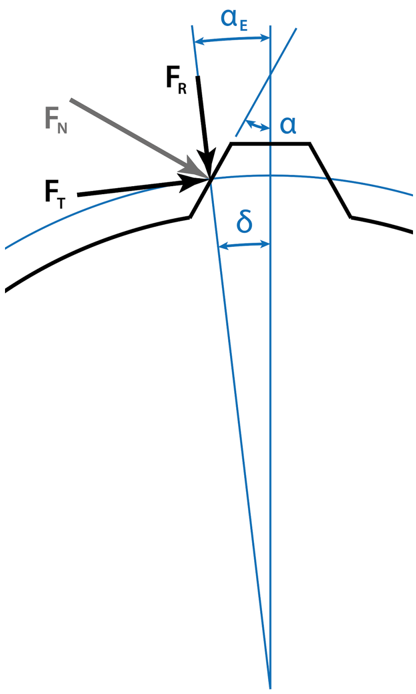


Figure 11.2: Forces acting on a type B spline flank. The pitch circle (blue) is also included.

$$p = mz \tag{11.4}$$

$$t = \frac{\pi}{2p} \tag{11.5}$$

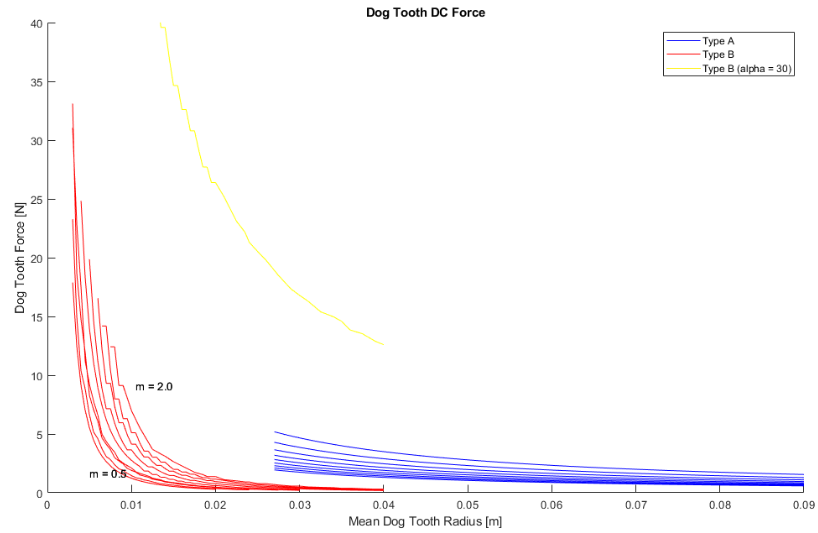
$$\delta = \arcsin \frac{t/2}{p/2} \tag{11.6}$$

$$\alpha_e = \alpha + \delta \tag{11.7}$$

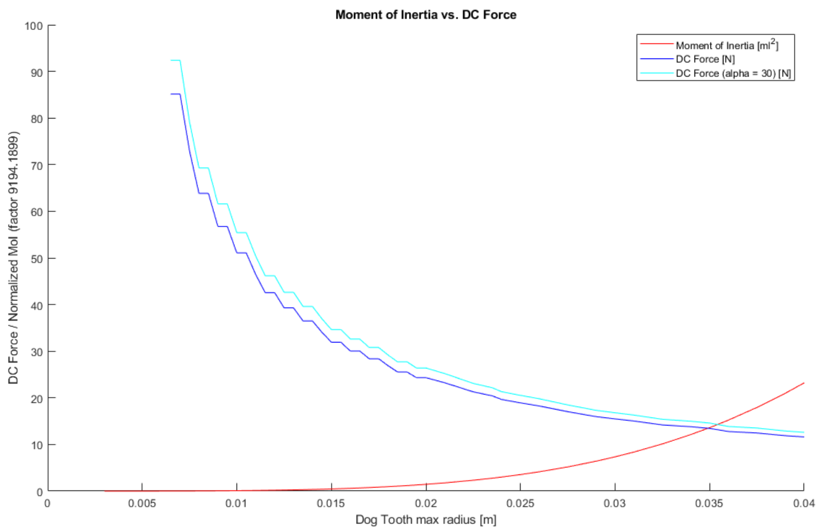
The disconnect force for type B can then be calculated by combining equation 11.3 with the geometry of figure 11.2. The pitch can be used to approximate the force lever arm utilized by the torque over the axle. The forces in figure 11.2 are acting along this pitch radius.

$$F_{dc,B} = \frac{M_{tot}\mu}{\frac{p}{2}\cos\alpha_e} \tag{11.8}$$

To visualize the disconnect force for the different dog clutch types and their diameters, graph 11.3a shows the relationship between these variables. The individual graphs for type A and B are the varying modules (simplified to varying teeth sizes for type A) of the dog teeth. The type B graphs (red) show the forces with a pressure angle, α, of 20 degrees. A graph representing a pressure angle of 30 degrees is also included in yellow. The graph in figure 11.3b attempts to highlight the conflict between a low moment of inertia for an approximated cylindrical axial half of the type B dog clutch (see figure 2.6b) and the total disconnect force. An arbitrary weight factor has been applied to the moment of inertia to receive results in somewhat equal magnitude. The calculation of the moment of inertia for the approximated cylinder is considered trivial. As evident by equations 11.7, 11.8 and figure 11.3a, the disconnect force varies highly with the pressure angle.



(a) The relationship between the dog tooth radius and the dog tooth force. Module 0.5 to 2 with pressure angle α = 20 is shown for type B, with an additional graph (yellow) showing type B with module 1.5 and α = 30.



(b) Total disconnect force for type B versus moment of inertia for an approximated, cylindrical type B dog clutch half.

Figure 11.3: Graphs showing the disconnect force depending on dog clutch radius.

11.1.2 Torque

The output torque varies for the two chosen architectures, and can be described as equations 11.9-11.10 show. The values for specific segments of torque can be seen in table 3.3 in section 3.5. Equation 11.9 shows the torque for concept WY, and equation 11.10 is for concept V. Please note that equation 11.10 must be halved before further calculations, as the torque going into the differential is split between the two output axles. These output axles are assumed to have an equal torque distribution.

$$T_{DC,propshaft} = T_{drag,EM} + T_{ASC} \tag{11.9}$$

$$T_{DC,sidediff} = (T_{drag,EM} + T_{ASC})U_1U_2U_3 + T_{int,1}U_2U_3 + \tag{11.10}$$

$$+ T_{int,2}U_3 + T_{diff} \tag{11.11}$$

11.1.3 Helical Gear Forces

Helical gears give rise to axial forces through the mounted axles. These forces were calculated to see if there was any potential of them affecting any bearings or other components. A helical gear tooth has specified angles, ψ and φ, the *helix angle* and the *pressure angle*. By applying these, we arrive at the following equations 11.12.

$$\begin{cases} W_t = \frac{M_{in}}{r_p} = \frac{2M_{in}}{d_p} = \frac{2pM_{in}}{z} \\ W_r = W_t \tan\varphi \\ W_a = W_t \tan\psi \end{cases} \tag{11.12}$$

Where M_{in} is the input torque in Nm, r_p is the *reference radius* (defined as the radius of the pitch circle of the gear, see figure 11.4) in mm, p is defined in equations 11.4 and 11.13 below (diametrically).

$$p = \frac{z}{d_p} \tag{11.13}$$

The module of a gear is defined as the inverse of the pitch and is defined in equation 11.14.

$$m = \frac{1}{p} \tag{11.14}$$

Equations 11.13 and 11.14 can be combined to calculate the pitch radius shown in equation 11.15.

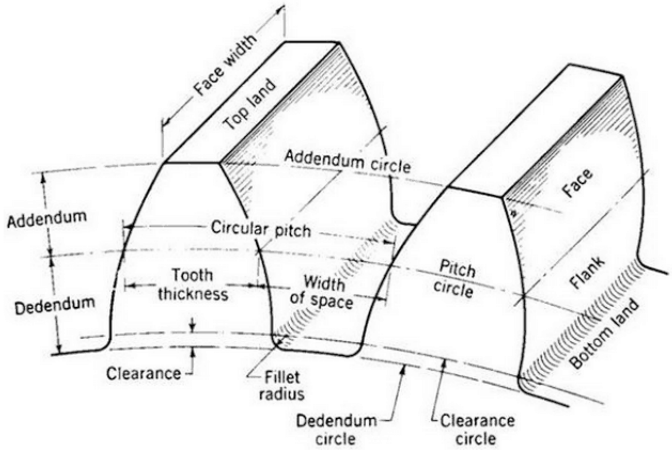


Figure 11.4: Nomenclature for gear teeth, here shown for spur gears but also applicable for helical gears. [59]

$$r_p = \frac{mz}{2} \tag{11.15}$$

The modulus m and z, the number of gear teeth, are known.

11.1.4 Spring Forces

A few different types of springs were examined as possible candidates for the return function during this process. For a commonly used spiral spring, the force exerted is relatively simple to calculate. Every spring has a specified spring constant, c. If only the maximum force exerted by the spring, F_{max} is available as data the spring constant can be calculated according to equation 11.16, using the compressed spring length L_{min} and the full spring length, L_{max}.

$$c = \frac{F_{max}}{L_{max} - L_{min}} \tag{11.16}$$

The pretension of the spring depends on the difference between the disconnected spring space L_{disc}, which can be specified by the design, and L_{nom}. The pretension force F_{disc} is calculated in equation 11.17, and the largest acting force on the actuator, L_{conn}, subsequently. An illustration is shown in figure 11.5.

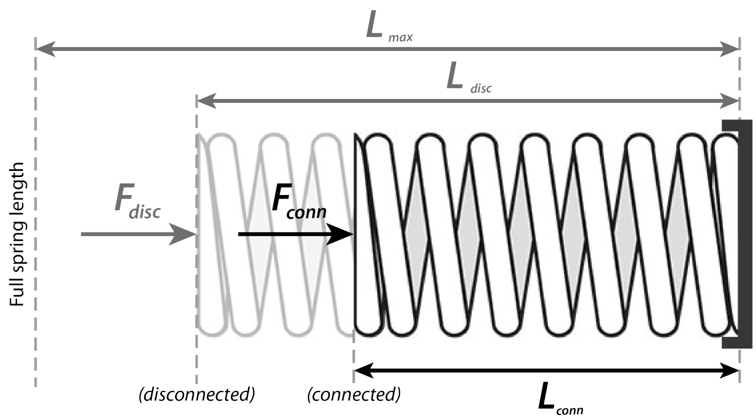


Figure 11.5: Spring forces depending on the location of the dog clutch.

$$F_{disc} = c(L_{max} - L_{disc}) \tag{11.17}$$

$$F_{conn} = F_{disc} + c(L_{disc} - L_{conn}) \tag{11.18}$$

For the return action of the dog clutch, the spring must output a high enough force at the last point of contact during disconnection. This force depends on the ASC torque (maximum DC torque) and the diameter of the dog clutch. In short, F_{disc} must be equal to or higher than the DC force F_{DC} to be able to push the dog clutch out of connection when the actuator is powered down. It is desirable to have a certain safety factor for this return force, called s_{return}.

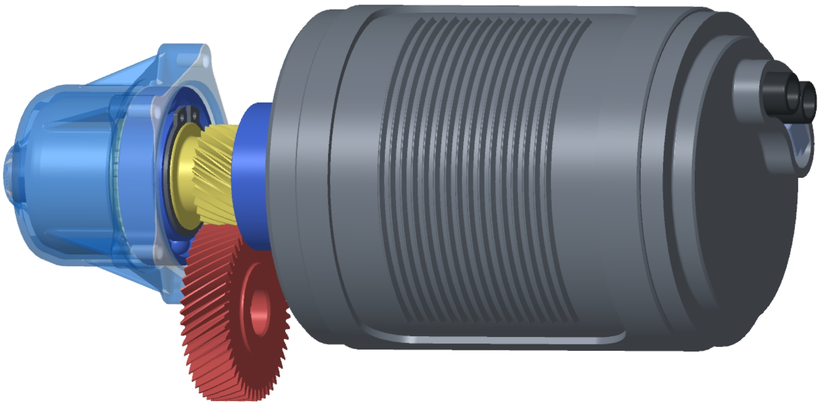
$$s_{return} = \frac{F_{disc}}{F_{DC}} \tag{11.19}$$

11.1.5 Bearings

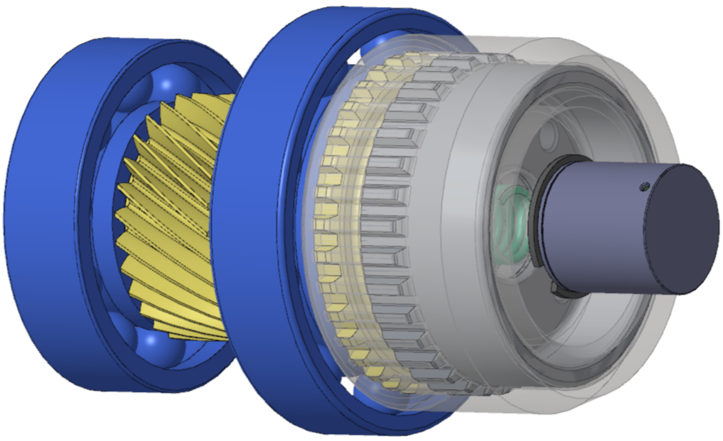
During basic bearing calculations, standard selection methods provided by SKF were used. These are specified and explained in the respective product group catalogs available from SKF without charge [60]. Preliminary bearing choices can be seen in appendix B.

11.2 Final Concepts

The final concepts were V and WY which were developed further after being chosen during the concept scoring. The development included a more detailed layout and construction of the concepts. Feasible bearing layouts, components of the reasonable size (no longer just general sketches), as well as evaluation of the working principles. Many iterations of the concepts were developed, with both large restructuring changes as well as smaller dimensioning issues.



(a) The placement of concept WY next to the EM.



(b) The dog sleeve made transparent to show the dog teeth (the orientation being mirrored from figure 11.6a).

Figure 11.6: The final concept WY, placed directly at the electric motor. The complete drivetrain and housing is not part of the design space and is not shown.

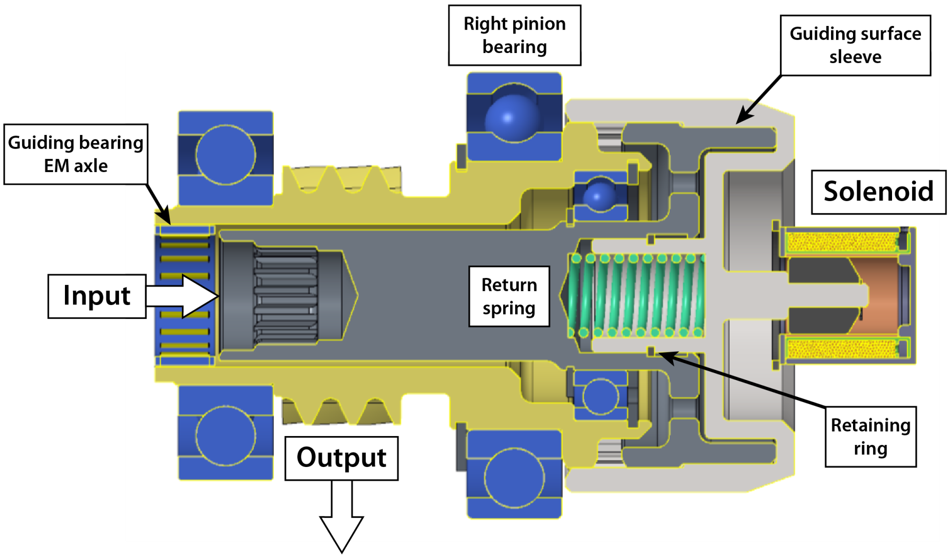


Figure 11.7: Detailed view and points of interest for concept WY. Shown in connected position.

11.2.1 Concept WY - First Pinion

The final system iteration is shown in figures 11.6 and 11.7. A few additional iterations are shown in figure 11.9. The First Pinion concept went from a sleeve-type multiple-solenoid solution to a more robust design with a completely redesigned dog clutch. This was due to difficulties placing bearings and supports suitable for the high rotating speeds present in the system. The packaging was an issue as the EM pinion (yellow in figure 11.7) is relatively small in size and cannot be enlarged without redesigning other components along with the drivetrain. The intention, as shown in section 8.3 is the ability to place the concept directly on the output axle of the EM, while not influencing the rest of the electric drivetrain. Advantages at this position include a low disconnect force, compact design and low total weight of the system. The actuation is also less susceptible to *klonk* due to its position, which can open up for faster actuation times.

Component guidance and assembly stiffness received heavy focus during the development to avoid any balancing issues or misalignment. Minimizing the sliding distance of the sleeve contact points was crucial in the beginning, as these result in substantial losses at high speeds. A design conflict was found at this stage, as the sliding distance is minimized using smaller radii, while the disconnect force should use as large of a radius as possible for low actuation forces. The final design iteration circumvented this problem by utilizing a coaxial solenoid. Proper support and guidance of the axles are imperative for this concept, which is reflected in the extended guiding surface for the dog sleeve as well as a support surface between the EM and the pinion. Both are highlighted in figure 11.7. Guidance on the EM-side of the pinion axle is managed by a needle roller bearing. This bearing motion is eliminated during connected operation, as the input axle from the EM and the pinion have the same rotational speeds.

The largest bearing in the system, the right pinion bearing (marked in figure 11.7), is limited in size by the gear teeth of the pinion. It must fit over the teeth but the size limits its rotational speeds. This might not be an issue during continued development but the potential issue should be addressed early on. The inner bearing is mounted with steel on both sides, which should reduce any risk of jamming due to temperature differences. Holes in the input axle facilitate the mounting of the inner bearing during assembly.

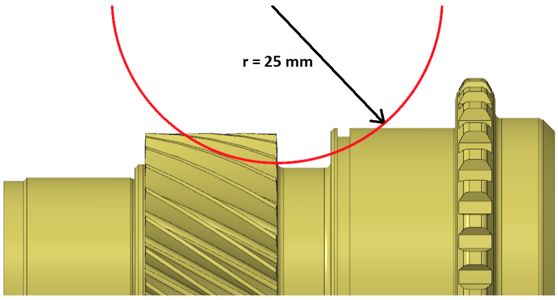


Figure 11.8: Manufacturing by hobbing of the helical gear teeth.

Manufacturing of the helical gear teeth will most likely be performed by hobbing, which requires space on both sides of the teeth during operation. According to our supervisor, the smallest hob diameter is 50 mm, and dimensions were set accordingly (see figure 11.8). A small part of the bearing seat for the right pinion bearing will be removed, but this should not be an issue.

The weight of the system is approximated to 1.65 kg at this stage, with the rough housing version weighting an additional 0.3 kg. The heaviest components are the three main components: the input axle, output pinion and dog clutch sleeve. The total system length is 128 mm, measured from the left end of the output pinion to the end of the solenoid housing (see figure 11.7), excluding housing.

Working Principle

By moving the sleeve of the dog clutch while synchronizing the rotational speeds of the in- and output axles, the output pinion of the EM can be connected to the rest of the drivetrain. The input axle (coming from the EM), moves through the output pinion, which facilitates the bearing layout as well as negates some disturbances along the axle. In early iterations, the solenoid moved an actuation fork, which in turn pushed the dog sleeve. Return movement was made possible by a spring package placed coaxially with the solenoid. Later iterations instead place the solenoid coaxially with the input axle itself, while the plunger mounted directly on the sleeve is free to rotate inside the solenoid coil. The plunger could also be suspended with a bearing, no longer rotating inside the solenoid housing. The return spring package is placed between the input axle and the solenoid. In order to limit the axial movement of the sleeve, the inside of the sleeve abuts against the edge of the input axle (at the end of the sleeve guiding surface, see figure 11.7) while the system is connected, while the disconnected end stop is the retaining ring placed around the return spring holder. Disassembling this ring requires some force and a well-chamfered design for the outer edge of the inner recess, the end stop position of the retaining ring.

Iterations

All iterations are shown in figure 11.9.

WY - Concept is sketched without bearings and multiple solenoids in a sleeve configuration. Axial needle thrust bearings keeps the solenoids from rotating. A brush conducts electricity to the solenoids. The EM pinion placement is specified from the first iteration. See figure 10.1b.

WY1 - The EM axle is still placed through the pinion, but the dog clutch is changed to type A, splined on the inside of the pinion. A first bearing setup is chosen.

WY2 - New bearing setup and multiple solenoids connected to clutch with translation heels. Fork is added as an alternative to multiple solenoids, to reduce the number of actuators. Actuation via EM with an off-center bearing is also considered at this state.

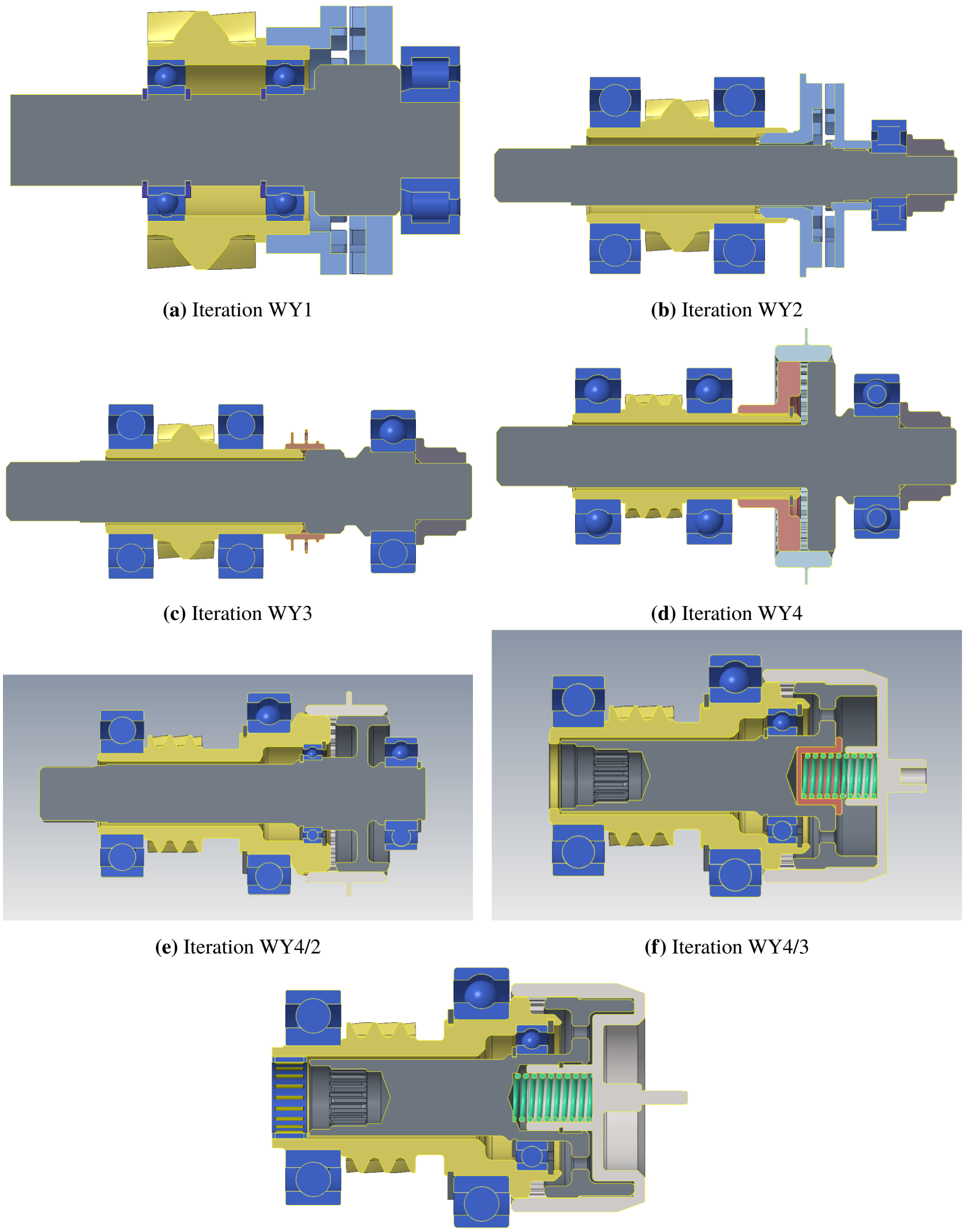
WY3 - Dog clutch is changed to type B and rough housing is sketched. A new bearing setup is made, right bearing change. Rough calculations of a solenoid and a return spring are completed.

WY4 - Dog clutch is enlarged to minimize disconnect force, the fork is redesigned slightly. Output half of the dog clutch is designed as a separate part (red) to facilitate mounting of bearings.

WY4/2 - Pinion (yellow) is redesigned, the dog clutch is now incorporated with it (fewer components) and slightly redesigned. Bearing setup is changed once again, the inner bearing is added for extra support. The right pinion bearing is enlarged to barely fit over the pinion teeth for mounting. The sleeve is made longer and some manufacturing issues are addressed (hobbing of helical gear).

WY4/3 - Pinion is shortened, the dog clutch is redesigned for shorter splines but with a support surface for the sleeve to stabilize it. Holes in the input axle for assembly. The rightmost bearing is removed and the inner bearing is made larger. The concept is prepared for a solenoid pushing the sleeve axially, with the plunger integrated with the sleeve. The return spring is moved to the inside of the input axle (gray).

WY4/4 - The layout is made more compact, with the return spring moved further into the axle. The solenoid is incorporated into the main housing. A support surface with a needle roller bearing is added between the pinion axle and the EM axle. Rough solenoid design is added. The sleeve disconnected end position is placed inside the return spring channel, acting upon a retaining ring.



(a) Iteration WY1

(b) Iteration WY2

(c) Iteration WY3

(d) Iteration WY4

(e) Iteration WY4/2

(f) Iteration WY4/3

(g) Iteration WY4/4

Figure 11.9: Iterations of concept WY.

Components
The number and layout of components vary slightly between iterations, but the final design, see figure 11.7, consists of the following parts:

- Input axle (gray)
- Output pinion (yellow)
- Dog clutch sleeve (light gray)
- Solenoid plunger (black)
- Solenoid coil (orange)
- Solenoid coil bobbin (green)
- Solenoid housing (dark gray)
- Return spring (aquamarine)
- Bearings (blue)
- Lock/retaining rings (dark gray)

Calculations
In order to get a better overview of the required dimensions and materials, general calculations were performed. The most important force for the disconnect system is the disconnect force, $F_{DC,B}$, which is calculated according to equation 11.8. By examination of graph 11.3a a suitable dog clutch radius was chosen to be 30 mm. The spline is dimensioned with a module of 1.75, which according to spline tables set the number of teeth, z, to 33. The pressure angle, α, is set to 20 degrees following the results in section 11.1.1. The effective angle is calculated to $a_e = 20.03$, with $\delta = 0.03$ being virtually negligible. To disconnect the dog clutch, the actuator must thus exert the force calculated as

$$r_{p,spline} = \frac{p}{2} = \frac{m_{spline} z_{spline}}{2} = \frac{1.75 \cdot 33}{2} = 28.875mm \tag{11.20}$$

$$F_{dc,B} = \frac{M_{tot}\mu}{r_p \cos\alpha_e} = \frac{5.7 \cdot 0.14}{0.028875 \cdot \cos 20} = 29.41N \tag{11.21}$$

The maximum axial force that can be transferred through the splined dog clutch is then calculated. The maximum torque output by the EM is assumed to be 80 Nm.

Table 11.1 Dimensions for Lesjöfors spring 2971.

D_t [mm]	D_m [mm]	D_i min [mm]	L_0 [mm]	L_n [mm]	F_n [N]	c [N/mm]
1.6	12.5	10.6	39	18	90	4.3

$$F_{max,B} = \frac{M_{max}\mu}{r_{p,spline}} = \frac{80 \cdot 0.14}{0.028875} = 387.9N \tag{11.22}$$

This force helps to evaluate at which point the bearings will start taking up any high axial loads through the system. As the disconnect force is known, the required return spring can be dimensioned. The disconnected space for the return spring is first set to 26 mm, and a suitable spring is the Lesjöfors compression spring (catalog number 2971, see table 11.1). The pretension and maximum return force, F_{conn}, and return safety factor s_{return} can now be calculated.

$$c = \frac{F_{max}}{L_{max} - L_{min}} = \frac{90}{39 - 18} = 4.29N/mm \tag{11.23}$$

$$F_{disc} = c(L_{max} - L_{disc}) = 4.3 \cdot (39 - 30) = 38.7N \tag{11.24}$$

$$F_{conn} = c(L_{max} - L_{conn}) = 4.3 \cdot (39 - 24) = 64.5N \tag{11.25}$$

$$s_{safety} = \frac{F_{disc}}{F_{dc,B}} = \frac{64.5}{29.4} = 2.19 \tag{11.26}$$

Helical forces can also be calculated, and the pinion gear is specified in table 11.2a. The axial and radial components of the helical forces are calculated according to equation 11.12.

$$r_{p,helical} = \frac{m_{pinion}z_{pinion}}{2} = \frac{1.3 \cdot 27}{2} = 17.55mm \tag{11.27}$$

$$\begin{cases} W_t = \frac{80}{0.01755} = 4558.4N \\ W_r = 4558.4 * \tan 20 = 1659.1N \\ W_a = 4558.4 * \tan 30 = 2631.8N \end{cases} \tag{11.28}$$

Calculating the solenoid speed is difficult without any testing being done, but similar solenoids can be examined for an estimate. A Ledex solenoid (part number 129440-0XX) is used for comparison, which has an actuation time of 10 ms for approximately 5 mm of travel (actuation length, L_{act}) at a 10% duty cycle at 125 W [61]. This could be sufficient to assume the total time of the system will be below the ideal target value of 100 ms, although further testing is necessary. Ledex solenoids have been examined previously for use at BorgWarner. This specific solenoid might be oversized for this application but the general actuation speed should be helpful. Please note that the speed might decrease the following load application. The specific solenoid force used for the system can be specified after final design elements are set.

Table 11.2 Input data and output values obtained from calculations for concept WY.

(**a**) Input data.

Input	Value
M_{EM} [Nm]	5.7
$M_{EM,max}$ [Nm]	80
n_{max} [rpm]	20 000
μ [-]	0.14
m_{spline} [-]	1.75
z_{spline} [-]	33
α_{spline} [°]	20
m_{pinion} [-]	1.3
z_{pinion} [-]	27
ψ_{pinion} [°]	30
φ_{pinion} [°]	20

(**b**) Output data.

Output	Value
$r_{p,spline}$ [mm]	28.875
$F_{dc,B}$ [N]	29.4
$F_{max,B}$ [N]	387.9
W_r [N]	2632
W_a [N]	1659
F_{disc} [N]	38.7
F_{conn} [N]	64.5
$r_{p,pinion}$ [mm]	17.55
L_{act} [mm]	5.5
t [ms] (5.5 mm)	<100

11.2.2 Concept V - GenVI Hydro Sleeve

The final design of concept V can be seen in figure 11.10 and 11.11. The project name of the concept was stated *GenVI Hydro Sleeve* since it uses a BorgWarner hydraulic pump GenVI and a sleeve collar dog clutch to actuate. Compared to 10.1a it can be seen that some changes have been made in order to improve the design and make it more realizable and are explained below.

Iterations

V1 - Splines for sleeve and shafts were designed with dimensions set according to similar projects. This was done to achieve a reasonable interface and geometry.

V2 - General dimensions changed, both in diameter and depth to comply with the bigger spline diameter of the sleeve, piston, depth of spring, etc.

V3 - The piston was changed to include a smaller area for hydraulic force and larger area for spring to act on. This was done to raise the spectrum of force the actuator would apply. The spring in turn can then be of larger amplitude to generate force enough to disengage the dog clutch.

V4 - Standard deep groove ball bearings fitted in the design. The bearings chosen can be seen in appendix B. This was performed early in the detailed design phase since a limited amount of standard bearings were to be used and some interfering geometry had to be changed accordingly. It was confirmed through bearings data from the provider that the bearing could withstand relevant axial loads from engagement.

V5 - Surrounding surfaces to bearings were altered with shoulders to lock them axially and not to build up tension in the structure. Lock rings according to standard DIN 471 and DIN 472 were fitted to make the assembling of components possible.

V6 - The spring was changed to a coil spring instead of a disc spring to comply with dimensions and required properties. It was realized that if a standard disc spring was to be used it would have to be stacked multiple times to achieve enough spring force and displacement. In the end, custom made disc springs or coil springs which satisfy the requirements can be easily produced.

V7 - Sealing was fitted on the piston to seal from hydraulic oil getting out and allow for sliding when actuating.

V8 - Return channel for hydraulic flow and a high-pressure valve were included. These were designed with regards to manufacturability with channels in an angle so drilling will be possible. The high-pressure channel was also changed to be integrated with the high-pressure outlet from the GenVI pump, see figure 10.1a.

V9 - Design of the housing was improved and assumed as casting as a way of manufacture and aesthetic looks. Support structures and radii around edges were introduced as well.

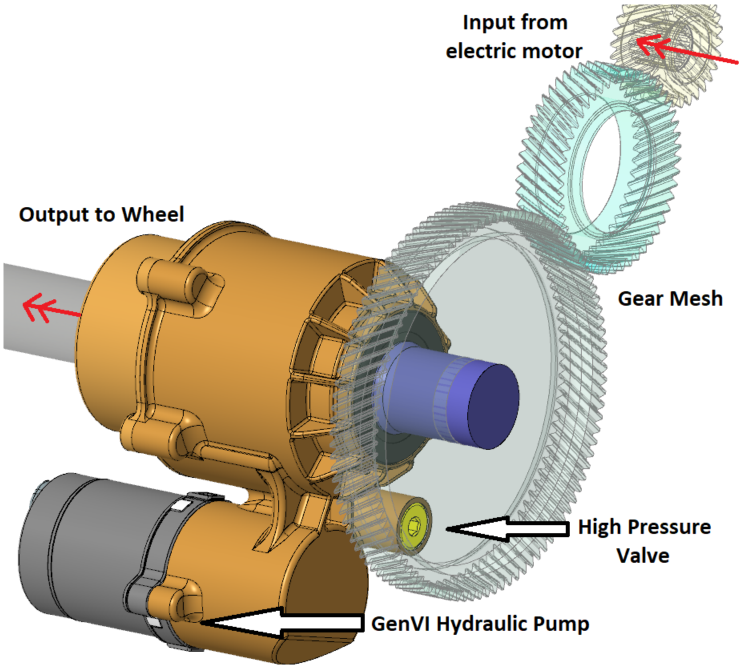


Figure 11.10: General view of concept V.

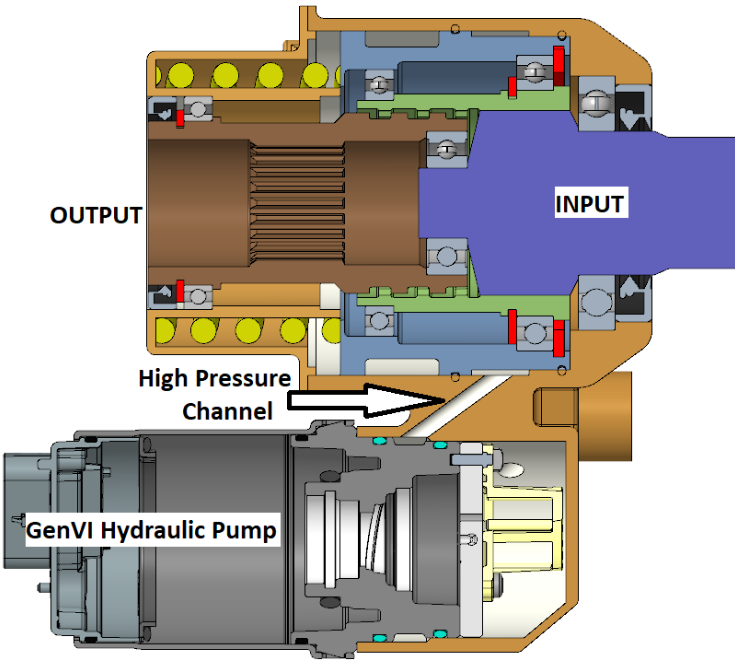


Figure 11.11: Section view of concept V. High pressure channel visible.

Components
A section cut of the disconnect can be seen in figure 11.11 where all internal parts are visible. Most parts of the GenVI hydraulic pump have been hidden.

The concept consists of following parts:

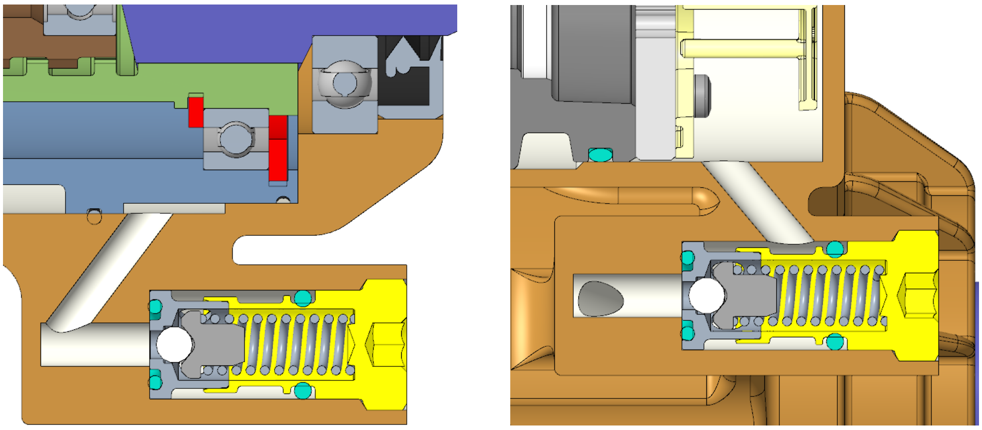
- Housing (Light Brown) split in two
- GenVI hydraulic pump (partially hidden)
- Input shaft (Dark Blue)
- Output shaft (Dark Brown)
- Sleeve dog clutch (Green)
- Piston for engaging dog clutch (Light blue)
- Return spring (Yellow)
- Bearings (Grey)
- Locking rings (Red)
- O-rings (for sealing piston) (Black)
- Radial sealings (Black)
- High pressure valve (visible in figure 11.12)

Standard components chosen from manufacturers like bearings and lock rings can be seen in appendix B.

Working principle
The GenVI hydro sleeve actuates the dog clutch with the help of hydraulic force and returns it with a spring, making it a monostable disconnect device. When the dog clutch is at a rest, pushed back by the spring, the splines of the sleeve spins freely in the cutouts of the output shaft. When engagement occurs the GenVI pump creates a hydraulic pressure by spinning the pump. The EM rotation speed is synced up to the rotation speed of the driveshaft. Through the high-pressure channel, the fluid pushes the small edge on the piston which engages the sleeve and the partial splines interfere with the splines of the output shaft. At this stage, the device is connected and transmits power from the electric motor to the wheels. The green sleeve and the blue input shaft have splines all way that interferes.

When the device is to disengage, the hydraulic pressure is released and the return spring pushes the piston back. The two bearings around the green sleeve are designed so that only the sleeve and shafts rotate, creating fewer problems for the piston and spring.

Return channel and high-pressure valve can be seen in figure 11.12. If too much pressure builds up with the risk of damaging components the ball pushed down by the small spring in the valve will lift and let the flow back to the reservoir seen in figure 11.12b.



(a) Return Channel.

(b) Low pressure channel.

Figure 11.12: Section view of hydraulic channels with high pressure valve visible.

Calculations

All input values and generated output data for the GenVI hydro sleeve concept can be seen in table 11.3. The displacement, rotational speed with corresponding pressure and efficiency for the GenVI pump is provided by BorgWarner and not shown in the table.

The total moment M_{tot} in the final drive, along the driveshaft is calculated with the help of equation 11.1. This is divided by two since the moment is split amongst two driveshafts along with the differential and the disconnect only acts on one. M_{EM} is derived from equation 11.10. The mean radius of the spline r_{sp} is calculated with equation 11.2. These values input in equation 11.8 gives the disengagement force $F_{dc,B} = 592\ N$ for a worst-case scenario with high coefficient of friction. Output data are summarized in table 11.3b.

Other input data used for spline design on the sleeve such as the number of teeth z, pressure angle α and module m can also be found in table 11.3a.

Table 11.3 Input and output data obtained from calculations for concept V.

(**a**) Input data.

Input	Value
M_{EM} [Nm]	8.2
R [-]	25
r_{inner} [mm]	24
r_{outer} [mm]	25
μ [-]	0.14
A $[m^2]$	0.000530
ϵ_P [-]	1
η $[Ns/m^2]$	0.039 (SAE grade 20)
m [-]	1.75
z [-]	32
$\alpha[°]$	30

(**b**) Output data.

Output	Value
M_{tot} [Nm]	102.5
r_{sp} [mm]	24.5
$F_{dc,B}$ [N]	592
F_{spring} [N]	708
$F_{holding}$ [N]	795
F_{engage} [N]	2120
$q_{holding}$ $[m^3/s]$	$7.94 \cdot 10^{-6}$
q_{engage} $[m^3/s]$	$1.23 \cdot 10^{-5}$
v_{engage} [m/s]	0.0233
t_{engage} [ms] (5 mm)	215
$\eta_{totP,holding}$ [-]	0.72
$\eta_{totP,engage}$ [-]	0.81
$E_{holding}$ [W]	14.80
E_{engage} [W]	61.2
I_{engage} [A]	10

The engagement force, flow and power use can be divided into two steps.

The first one is the holding force when the device is connected and is denoted with subscript 'holding' for variables in results. When holding, it is important that the power usage does not exceed the continuously power usage of 15 W. It is also important that the force generated is larger than the force generated by the spring F_{spring}.

Hydraulic force $F_{holding} = 795\ N$ was calculated with equation 2.4. Area A for current design can be seen in table 11.3a. The power needed for holding the dog clutch was provided by the help of equation 2.5. This gave that for actuation in 20 degrees Celsius with given dynamic viscosity η, relative modulation ϵ_P, pressure p and corresponding rotational speed n (BorgWarner data) the effect raised to 14.80 W, fulfilling the requirements specified of max 15 W while actuating. The total efficiency $\eta_{totP,holding}$ was estimated to 72% with help of equation 2.6 and 2.7.

A standard spring was chosen from *Lesjöfors AB* with catalog number 1328 and had to overcome both disengagement force $F_{dc,B}$ and some hydraulic resistance from pushing the fluid back in the channel. The hydraulic resistance can be assumed small when large return channels are used. Specifications of spring chosen can be seen in figure 11.13 and table 11.4.

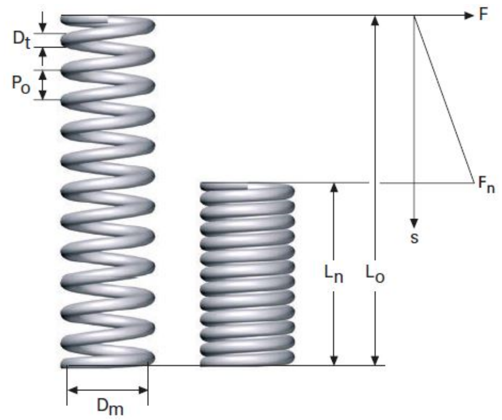


Figure 11.13: Dimensions of coil springs. [62]

Table 11.4 Dimensions for Lesjöfors spring 1328.

D_t [mm]	D_m [mm]	D_i min [mm]	L_0 [mm]	L_n [mm]	F_n [N]	c [N/mm]
6.3	63	55.9	78	36	902	21

The spring was assembled so a preload would exist and calculated with the help of equation 11.17. When the device is disengaged, the spring will be compressed from 78 mm (L0) to 50 mm (chosen in the current design). Since springs can be considered linear (c in table 11.4), linear interpolation can be used and gives a force of 601 N when spring is compressed to 50 mm. When actuating 5 mm to engage the dog clutch, the spring will in total be compressed to 45 mm and a corresponding force $F_{spring} = 706\ N$. The piston force $F_{holding} = 795$ calculated must overcome this value and F_{spring} must overcome the value of $F_{dc,B} = 592\ N$.

The second step is when the device is engaging and are denoted with subscript 'engage'. When engaging it is important with high speed and not to overcome power usage during actuation of 200 W as specified.

When engagement occurs, the hydraulic pump is spun up to maximum speed and corresponding pressure available (BorgWarner data). This is done to acquire maximum speed for engaging. The flow q_{engage} was calculated according to equation 2.2 with given displacement and rotational speed for the specific case. Piston speed v_{engage} was calculated with equation 2.3 and actuation time t_{engage} for actuating length $L = 5\ mm$ was then calculated as $t = L/v$. The time was calculated to $t = 215\ ms$.

Power usage during actuation was calculated with the help of equation 2.5 with corresponding rotational speed, pressure and viscosity in oil. It was calculated to

$E_{engage} = 61.2\ W$ and is lower than the 200 W specified. Efficiency for specific case raised to $\eta_{totP,engage} = 81\%$

With experimental data provided by BorgWarner, the corresponding current was given to 10 A during actuation with full pressure and is lower than the 15 A specified in the requirements.

The current dry weight of concept V is 2.67 kg. This is an estimated weight without the final material set or hydraulic oil included. The housing and piston are estimated as aluminum and sleeve and the output shaft is estimated with steel. Input driveshaft is not included. Weight from the actuator is provided from BorgWarner. Standard components such as bearings and lock rings are provided by the manufacturer. The weight was then provided using tools in CAD-software. It is important to note that this is an approximate value without any optimized structural analysis performed to minimize weight.

Chapter 12

Discussion

This chapter presents a discussion of the method used for product development of the disconnect device. The discussion raises questions such as errors in the method, things that could have been done differently and other difficulties. It also raises some questions regarding design problems that were encountered.

12.1 The Development Process

Phase 1 and 2 of the development process described in section 1.5.1 according to [3] were during this project separated by a very loose boundary, as the phases were worked on in tandem during some periods. As an example, during this project, the initial stage of gathering information about if and how a product should be developed was skipped. This was due to the company already providing the fact that there was a need for this kind of device. Some of the data needed was also provided. This was customer needs that were interpreted into metrics by the team. The projects process were iterative, were different steps have been run multiple times. Some parts of the process were dependent on information that sometimes was not available until a later stage of the project. These parts had to be adjusted as time went on. In hindsight, more time should have been spent on the specifications to really be sure when to proceed to the next step. This would have created a more rigid base develop from.

The process outlined by Ulrich and Eppinger is an extensive and powerful tool for the development of well-defined and structured products and systems but is a process that can require a lot of time. Especially if the system is not very well understood from the beginning and there must be time invested in research and understanding of it. Their process puts heavy emphasis on the initial structure and assumptions or interpretations of customer needs, metrics, etc. As for example some of the customer needs and parts of the target specification were received later during the process, instead of the beginning, the timeline had to be modified correspondingly. Customer needs are the primary guidance method of the concepts and need to be expanded upon properly to be effective. For example, a customer need stating *The DC does not connect accidentally* was first included for the project, but later removed due to it being more relevant for control system design. This is probably one of the reasons why there were some uncertainties regarding feasibility when screening and scoring of the initial concepts had been performed. The actuator technologies and relevant physical theory would also be a part of this initial research. A good understanding at the beginning facilitates good decision-making later on.

Patents are often researched while developing new concepts to gain an understanding of different technologies and get fresh ideas. One problem is the risk of patent issues later during the project if concepts are too similar or if no patent research has been performed at all. This could mean that ideas performing extremely well during the concept development turn out to already exist in a patent. One of the concepts (Concept K) was found to be very similar to patents held by competitors and is a commonly used mechanism in other driveline systems.

Individual Screening

The method of having the two team members first do the screening on their own and then discuss them to create a final one was proving to be a promising method. The idea was to not be affected by each other too much and that everyone's opinion mattered. When screening about 38 concepts it can easily happen that one of the team members zone out and let the other one do most of the screening but with this method, the two screenings are compared so as not to miss out on anything. It was proven useful since some of the screening differed and discussions aroused, comparing criterion were one team member e.g. had not judged it correctly or miss-understood the concept.

Architecture Scoring

The reason for initially doing an architecture scoring of where to put the disconnect device was simple. Since different positions had both pros and cons, making an architecture scoring to eliminate bad positions and raise the good ones saved time in the project. The architecture scoring also proved knowledge of the whole drivetrain for the team.

It was realized during the process of architecture scoring that more time could have been spent on reading up on the driveline system. It is a very important step in the process since if an unfavorable placement was chosen, it would affect all the following phases and in the end, result in a bad product. More studies could have been done on how cost and manufacturing would affect scoring. It is an important criterion that will affect if the product can be sold with a profit or not. Since the supervisor was consulted during the scoring it was concluded that the results were acceptable in the end.

Choosing Criteria

At some stages of the process, it was difficult to choose suitable criteria. Some criteria were chosen but scrapped during the ranking of the concepts, as these had a promising ulterior motive but impaired the selection process in the end. One of these criteria was the *Energy Source Flexibility* during the first concept screening. It was added to illustrate the advantage of a solution being flexible for application with many different actuators, but we soon realized it mostly degraded concepts which were though-out with a specific actuator technology in mind. If the concept was drafted for use with a solenoid, it would receive a negative score simply for having an actuator technology specified. During the first concept generation the preferred actuator technology was chosen. This became detrimental for the later concept screening. Some limitations could have been made to only choose mechanisms for the dog clutch movement in the early stages, and choose the specific actuator at a later stage.

The *Performance criteria* was difficult to evaluate at early concept screening stages, as the specific connection times and force outputs for the individual actuators and concepts were not fully known at that time. The concepts that made it to the scoring were evaluated further regarding performance. For the screening, however, this would mean an indispensable amount of time and might not have been worth it in the end since a large amount of the concepts might have failed in other ways.

Valuable feedback was received through discussions with the supervisors with new ideas on how to set up the criteria. This was important since in some cases, knowledge and experience are critical, and it is hard to calculate or estimate many of the criteria in a correct way.

It was challenging to find relevant data to compare for different actuators. Some of the data for the hydraulic pump and solenoid could be gathered internally from BorgWarner but other actuators had to be searched for externally. One way could have been to ask for expert knowledge from various companies producing different types of actuators, and receive information relevant to the system. Unfortunately, this was considered too time-consuming in comparison to other activities during this part of the project. In the end, it was decided that actuators with similar size and power would be used as data from different providers and would be sufficient enough for a rough scoring. The criteria *Reliability* was first included in the actuator scoring but later removed due to it being very difficult to accurately gauge the subjective reliability of an arbitrary actuator model. It made more sense to place weight on criteria that could be described more accurately.

Concept Discussion
It is worth to mention a brief discussion of the concepts that seemed promising but were removed at a later stage. A common problem with these was proving a stable mode of operation, to continue further development.

Concept AF - Disc Spring Jumper - There were early difficulties with concept AF, see table 8.3. The inherently quick nature of the bistable spring proved challenging to handle with a single solenoid. There was a risk of the spring damaging the solenoid by pulling too hard during a very short timeframe, or the solenoid simply not keeping up with the spring, introducing more uncertainty to the system. By using the return spring to catch the disc spring, some of the speed and force could be negated. This return spring would also assist the solenoid when pulling back the disc. The inherent risks in the design ultimately made it lose the selection,

Concept N - Electromagnetic Sleeve - This concept was promising during early concept development, but the use of electromagnets proved difficult to verify as a feasible concept. There are many issues to solve before the concept can be reliably tested, with total system weight and gap size between the magnet and the magnetic material being the main problems. The idea of magnetizing the disc spring to transmit magnetic force to the dog clutch was discussed with supervisors but in the end, provided to many uncertainties to go further with.

Concept D2 - Planetary DC - The planetary solution did not pass the concept scoring, mainly due to concerns about the speed of the system and the possible difficulties when making it monostable. Otherwise, the concept showed great potential with very beneficial force paths, stable movement of the dog sleeve and relatively compact structure. The actuation with an electric motor was flexible with possibilities for varying gearing, to facilitate force transfer.

Difficulties during screening
As can be seen in the results from table 10.2, some of the concepts that got high scores were removed. It was realized in a later stage of development that these concepts were difficult to realize, and a lack of time for extended research made it hard to verify their modes of operation. At this stage, there was a discussion amongst the team why these concepts had made it through the screening. After all, a screenings first priority is to eliminate concepts that do not work. Under the time constraints set for the research face it was difficult to receive a full understanding of all working principles considered. It could have been beneficial to extend the research phase.

12.2 Final Concepts

One of the main advantages of choosing two final concepts was the ability to work on one concept for each team member, while also being able to support each other during the development. Furthermore, it was decided that presenting two different concepts with two distinct placements and levels of already available technology would be a good idea from a customer perspective. One (concept V as explained further below) builds on the technology used by BW and introduces fresh ideas into a concept explored in the past. The other (concept WY) moves the disconnect device fully to a new location within the drivetrain and explores other actuation methods, such as electromagnetic. The placement along with the drivetrain for both concepts chosen has its pros and cons, as explained in 2.7. In the end, the customer can choose between two different concepts with different pros and cons and what specifications to prioritize. It is worth mentioning that the final concepts saw many iterations and the process of refining them was accomplished over a longer time period than expected.

12.2.1 Concept WY - First Pinion

One of the main difficulties while developing the first pinion concept was the placement and setup of bearings. The setup was changed several times following design changes of the components and as new and better placements became apparent. The first bearing setup, WY1, had a long tolerance chain going towards the dog clutch radii, which was avoided in the subsequent iterations by moving the bearings to the outside of the pinion. It is then mounted directly in the housing.

Only the outer bearings should be stressed during unexpected high loads, which could be accomplished by choosing a smaller mounting gap and thus letting these pick up any large axial forces. The same applies to axial forces from the helical gears during high EM torque situations. As the inner bearing must fit between the input axle and pinion geometry, as well as survive the high rotational speeds for longer periods, it is difficult to make it able to withstand very large axial forces.

To verify the connection speeds of the system, real tests using suitable solenoids would have been helpful. Measuring the force characteristics for different actuation lengths and duty cycles, as well as holding forces, would also have been advantageous. The solenoid solution used for the final iteration must be verified before it can be implemented for any real product. High rotational speeds lead to the danger of an uneven rotation of the plunger in the solenoid, which could lead to complications. This should be avoided with an air gap. The material used must also be very homogeneous, as slight variations in the permeability of the cross-section inside the coil could lead to variations in magnetic force or current. This drives up material costs for the plunger.

12.2.2 Concept V - GenVI Hydro Sleeve

Concept V has the advantage of building on an actuator known and used by BorgWarner, lowering the amount of testing having to be done before suitable parameters and designs can be found. It is today mass produced by BorgWarner and is a relatively cheap component compared to other actuators. The method of actuation (hydraulic) has been looked into by BorgWarner before, and this concept can hopefully show some further viability and new ideas for future development.

One of the main difficulties when designing this concept was the design of the piston, since this required a unique design and not according to standard cylinders. The reason for changing the geometry of the piston was due to high pressure from the hydraulic component. The GenVI works best around 6-40 bar of pressure and with the initial design this would mean to much actuating force and components could risk being destroyed. To increase the resolution the area that hydraulic forces acted upon were minimized. Compare figure 10.2 and 11.11 for reference. On the left side of the piston in figure 11.11. A shoulder were created for the spring to rest upon that had to cover more area.

The spring could be incorporated more into the design. This could have been done by choosing a spring that has a bigger diameter and placing it more compact in the primary house. A standard spring that would fit was hard to find so this will have to be done with a custom made coil spring. This is, according to supervisor, easy to do and is included on further work to do. An alternative to coil spring is the use of a disc spring. The problem with a disc spring was the lack of actuation length and force and this would mean stacking multiple of them. This generates a more complex and expensive product in the end.

Bistable Concepts

All concepts past the screening were evaluated for bistability, and a short description on how to achieve this was set for each one. This was performed since some customers might want a bistable option for their hybrid drivetrain and was asked for by the company if needed later.

Concept B - Remove return spring, possibly redesign groove pattern and reverse rotation of EM for one direction.

Concept N - Remove return spring and incorporate one extra electromagnetic actuator. Permanent magnets might be included to reduce holding force when not actuating.

Concept V - Remove return spring. Include a valve to determine the direction of flow in the hydraulic system.

Concept AF - Remove return spring, additional solenoid on opposite side for return action. Alternatively let single solenoid retract without disc, then perform start-up routine with a solenoid to retrieve the spring.

Concept D2 - Remove return springs, reverse EM rotation for one direction.

Concept CPX - Remove lifting functionality, reverse EM rotation for one direction.

Concept WY - Remove return spring. Include one extra solenoid that actuates on the opposite direction.

12.3 Future Work

To be able to arrive at a feasible and working concept, there remains additional work that must be completed. The design and layout of the concepts must be verified and finalized, with all components and dimensions receiving more concrete specifications.

To evaluate stresses and potential risks of failure in the system, an extensive finite element analysis should be performed on critical components such as axles, bearing seats, housings, etc. These analyses would use a specified *abuse torque*, which is much higher than the nominal values. This was originally planned during this project, but the changed schedule made it difficult to accommodate these simulations.

Balancing must also be a focus during continued development, as especially concept WY must be very well-balanced and controlled to avoid unnecessary vibrations or damage. A testing phase of the GenVI actuator would be preferred to validate it's function since it has not been used for this specific type of system.

The working principle of concept WY, with a rotating solenoid plunger, must be verified in practice. Any misalignment between the plunger axle and the coil can potentially lead to considerable issues, which puts a large emphasis on high-grade material properties and manufacturing methods. It must also be evaluated if any misalignment in the plunger (both axially and within the material itself) lead to currents or magnetic forces during operation.

General optimization of the speed difference when engaging the clutch must be performed, especially for concept V which is subjected to the full gear ratio. Optimizing the speed difference is a trade-off, as a low difference can result in a slow connection time and high differences can lead to a high energy quantity which has to be absorbed by the components. A dampener between the two connecting axles is an alternative to remedy some of the dampening issues, although an expensive one.

When a final concept has been chosen and fully designed, target specifications must be verified and updated according to the product development theory used.

Chapter 13

Conclusion

Hybrid vehicle technology is making enormous strides towards a greener environment and more advanced drivetrains in the future. Many challenges remain until hybrids can fully undergo the enormous shift present in the automotive industry today, and one of them is the seamless transition between internal combustion engines and electric motors.

Following the methodology of K. T. Ulrich and S. D. Eppinger was a powerful and diverse method for product development, which helped us make educated and motivated decisions. The method helped during all process steps by providing tools to take on different types of problems.

Even though many questions about the concepts remain for future development, many different technologies and ideas have been explored during the project. Exploration of many ways to actuate the dog clutch has been performed, weighing them against each other and choosing the best one. Since the concept not has been set for a specific type of vehicle this provides a good base for the development of a similar type of disconnect device.

Continued progress for the system includes testing and verification of the components, as well as testing of the promising but removed concepts if time permits. The presented concepts and the way to disconnect a driveline can hopefully help fuel this technological shift towards making hybrids a reliable cornerstone for the future of transport.

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Appendix A

Work Division & Time Plan

A.1 Work Division

The thesis work was divided evenly for most parts of the project, and was split when two concepts were chosen for detailed design. At that point, each student received one concept to continue work on, in order to gain better and more detailed understanding for the respective concepts. Oscar continued with concept V, while Matthis worked with concept WY.

During the detailed design phase, different parts of the two concept developments proved challenging. This led to the concepts reaching different design stages at slightly different times, but the general (extended) time plan was the same for both.

A.2 Time Plan

Due to both unforeseen outside factors as well as unexpectedly long development times, the *CAD* and *Structural Comparison of Concepts*-phases were extended. The final time plan reflects this, see figures A.1 and A.2, with the two mentioned phases being prolonged by six and five weeks respectively.

The extension of the CAD phase was prolonged both by a change of PDM system at BorgWarner, which hindered any substantial CAD work during one week, as well as the challenges of working more independently from home during the health crisis. Communication with colleagues at BorgWarner online did mostly work well, but was naturally slower than face-to-face contact at an office.

The structural comparison phase was begun even though the CAD phase was not fully finished, see figure A.2, and was a good complement for the more detailed CAD work. It was also extended mostly due to the CAD models being finished at a later point.

Writing parts of the report during the project proved very helpful at the later stages, as many parts of e.g. the theory were mostly complete.

Goal Description	**Duration**	**Comment**
Concept Generation Theory	1 w	*All theory ongoing during project.*
System and Functionality Theory	½ w	
Basic Model Understanding	½ w	*In order to understand critical components.*
Define System Requirements	2 w	*Including modularity & flexibility.*
Generate Concepts	4 w	
Basic Structural Comparison	1 w	*Hand and Excel calculations.*
Evaluate Concepts	2 w	
CAD of Concepts	2 w	*Alongside concept generation & evaluation.*
Structural Comparison of Best Concepts	2 w	*FEA-simulations.*
Sensitivity Analysis	1 w	*"Wrap up" phase.*
Report Writing	4 w	*Final four weeks, also ongoing during project.*
Total:	20 w	

Figure A.1: The first time plan as it was written in the goal document.

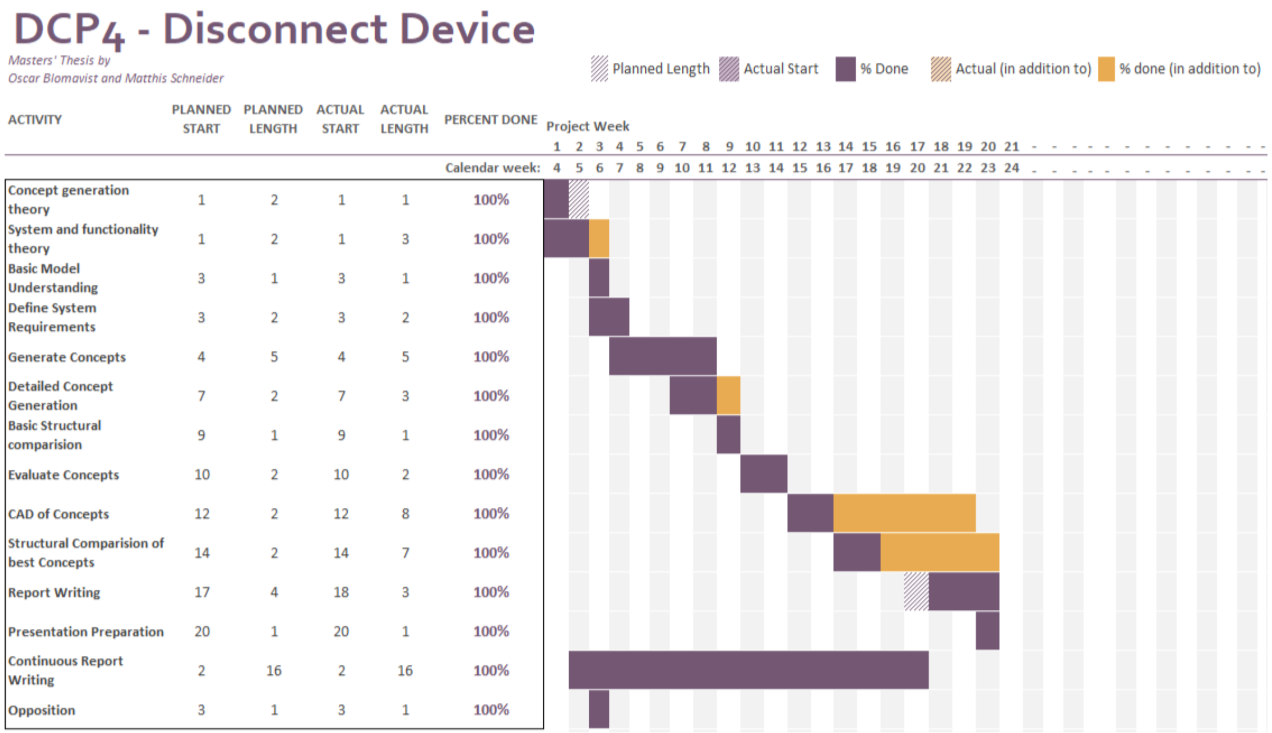

Figure A.2: The final time plan, modified for additional project weeks.

Appendix B

Standard Components

Standard components used for Concept Wy can be seen in table B.1 and for concept V in table B.2

Table B.1 Standard components used for concept WY

Type	Catalogue Nr.	Provider
Deep groove ball bearing	6206N	SKF
Deep groove ball bearing	6009	SKF
Deep groove ball bearing	61905	SKF
Needle roller and cage assembly	K 20X24X10	SKF
Lock ring DIN 472	04712 45	Mattssons
Lock ring DIN 472	04712 25	Mattssons
Lock ring DIN 472	04712 4	Mattssons
Lock ring DIN 472	04722 42	Mattssons
Retaining ring DIN 472	7777	Lesjöfors
Retaining ring DIN 472	7741	Lesjöfors

Table B.2 Standard components used for concept V

Type	Catalogue Nr.	Provider
Deep groove ball bearing	61811	SKF
Deep groove ball bearing	61810	SKF
Deep groove ball bearing	61808	SKF
Deep groove ball bearing	16007	SKF
Deep groove ball bearing	6003	SKF
Lock ring DIN 472	04722 72	Mattssons
Lock ring DIN 471	04712 55	Mattssons
Lock ring DIN 471	040A	Internordic
O-ring sealing	080000200	Internordic
O-ring sealing	083000200	Internordic
Radial Sealing	0350580945	Internordic
Radial Sealing	0400520715	Internordic