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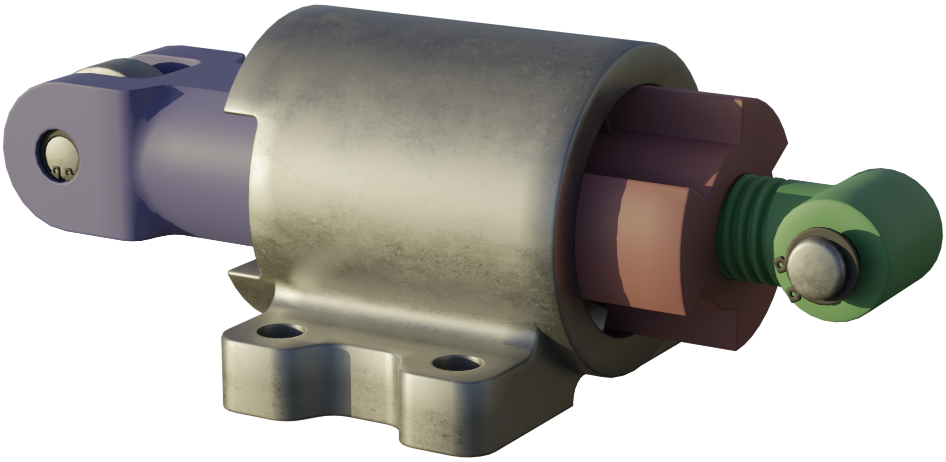

Wear compensation systems for dry clutches

How clutch actuation is affected by wear and how a wear compensation system can be implemented to make clutch operation more consistent

Master's thesis in Automotive Engineering

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Cover: The wear compensation device developed in this thesis.

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Abstract

With the environmental pollution becoming a greater concern one way to reduce emissions is to switch from vehicles with fossil fuel driven internal combustion to battery electric vehicles. This type of vehicle become increasingly popular and the demand for higher total energy efficiency and performance of the vehicle is essential for the customers. One way to do this is to implement a two-speed gearbox with a dry normally open friction clutch controlled by an electrical actuator. During the lifetime of the vehicle an axial wear of the clutch will occur and result in a need for different displacement and/or forces to operate consistently. To lower or eliminate this problem a wear compensation system is implemented.

To understand the type and amount of wear, the theories behind this was investigated. Using existing data from several sources the total axial wear of the dry friction clutch was calculated to be 2.15 mm over the complete life cycle. The next step was to find existing wear compensation solutions. This was done searching through patent databases and through information from the company AVL MTC which are the stakeholder in this thesis. The evaluation of these solutions resulted in a few concepts that was categorised and selected for further analysis. This investigation resulted in schematics and evaluation of their respective functions. When the design of the complete clutch system based on the requirements and specifications was done, only a few of these concepts was determined to work properly without major redesign. Due to the time limitation one of these concepts was chosen to investigate further.

This concept is called the Rod and Nut concept which rotates a nut when wear is present. This motion will increase the length of the rod because of the threads on the rod and nut therefore compensating for the change of displacement needed when the clutch is worn. This compensation procedure will happen when the actuator goes further compared to the normal operation, called the wear check cycle. The actuator will overload the system compared to the normal actuation using a predetermined force. Due to the design of the mechanism it will only compensate if the axial wear is large enough, that is due to the extended travel to reach that predetermined force when the clutch discs are worn. This is something necessary for this design and this procedure is preferably done when parking the vehicle and turning it off after everything is up to temperature from driving. The potential benefits of a compensation system like this, proved to be promising both regarding actuator requirements and actuation consistency. It also become clear that the actuation time performance could be improved significantly throughout the life cycle, using the same actuator power.

Keywords: wear compensation systems, dry clutches

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A special thank you for the people at AVL MTC in Trollhättan. Without them this thesis would not have been completed. Our supervisor Martin Johansson with the help of Martin Schagerlind helped us immensely throughout the whole process with support and answering our questions.

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Sven Bergqvist and Patrik Abrahamsson, Gothenburg, June 2022

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1 Introduction

The environmental pollution is one of the biggest challenges humanity is facing and also becoming increasingly important for the vehicle industry. The industry is now doing their part in reducing the emissions from their products. The current shift from fossil fuel propelled vehicles with internal combustion engine to using more environmentally friendly fuels or using battery electric machines is one big step to reduce emissions from the transportation sector. The implementation of electric powertrains is still quite a new field of technology and the energy density of the batteries is a limiting factor when trying to achieve a competitive vehicle drive range. This means that, even though the electric motor is very energy efficient compared to an internal combustion engine, the strive to increase efficiency is still of high importance. This results in more focus on efficiency regarding the drivetrain as a whole. This efficiency can be increased by implementing a multiple speed transmission with one or more extra gears.

1.1 Background

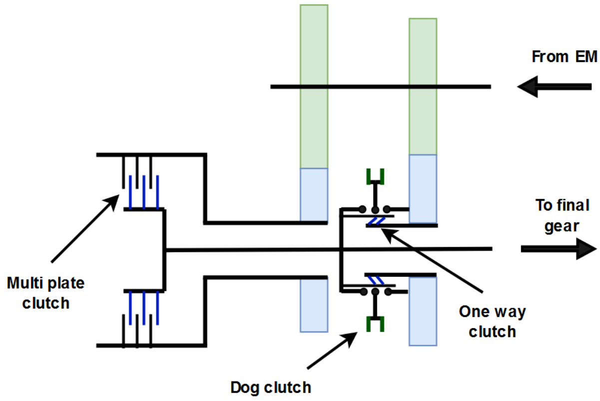
With an automotive industry that to a greater extent involves powertrains with fully electrical prime movers, the needs and requirements of the transmission has changed. The characteristics of electric motors with a broad usable power band compared to internal combustion engines has reduced the need of multiple speed transmissions. The use of two or more forward gears however have the potential of further increase total energy efficiency of the vehicle and make downsizing of the electric motor possible. A downsized motor along with a two speed transmission can still perform better regarding startability, high speed driving and enabling more overall operation within the higher efficiency range of the motor. The current passenger cars that implement a two-speed transmission is strictly high performance battery electric vehicles. The transmission is used to increase overall performance and top speed. Multiple speed transmissions will probably be more important to the fully electric heavy trucks that is currently being developed. Without this type of transmission, the motor would have to be too large and expensive compared to a smaller one coupled to a multiple speed transmission.

With this type of automated multiple speed transmissions comes a need for clutches and/or brakes in order to perform what is refereed to as powershifting, which is when the up shifts during acceleration are conducted with minimal torque interruption and thereby appears smooth to the user. At least one of these brakes or clutches needs to be of friction type (friction element). The actuation system of these friction elements can benefit from using a wear compensation system. The purpose of such a system is to operate as consistent as possible regarding transferable torque versus actuation stroke, throughout the lifetime of the clutch i.e. with decreasing axial thickness of the clutch disk(s) due to wear. In a transmission that allow for power shifts, the level of precision in the actuation of the clutch needs to be high and consistent. Deficiency in this aspect can not only cause uncomfortable shifts but can also lead to complete failure of the transmission. With no wear compensation system included, the clutch requires the actuation system to handle a longer stroke with still full capacity, which will result in a heavier and over-dimensioned actuation system compared to if the axial friction disc wear could be compensated.

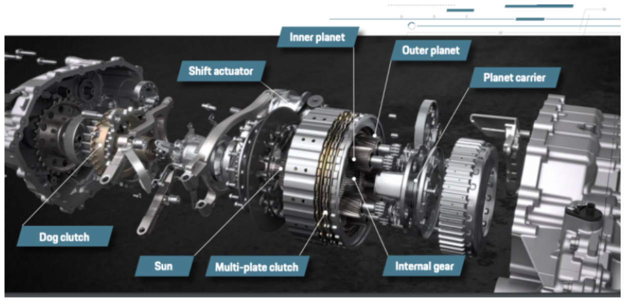
The current two-speed transmissions available that is specifically developed for battery electric vehicles are quite few in number since this field of development is fairly new but there are some concepts whereof one in production and used in the electric sports cars Porsche Taycan and Audi e-tron GT. Figure 1.1 illustrates four of the different concepts that allow for powershifting during acceleration.

The transmission from GKN (Figure 1.1d) [1] differs from the other three by using more than one friction element and for this reason it also allow for powershifting during recuperation. Important concerns regarding actuation of friction elements is whether it is a normally open or a normally closed setup and whether it is dry or wet. A normally open clutch is open in its natural state and is closed (engaged) using an actuation force. A normally closed clutch is instead closed by default and requires an actuation force to open (disengage). A dry friction element have a higher friction coefficient and thereby requires a lower contact pressure than a wet system but it also has a higher wear rate. Another drawback with a wet system is that there is energy losses due to drag when it is disengaged. The wet type friction element however is usually implemented as a multi-plate configuration, which increases the friction area and allows for a smaller package diameter and/or a lower actuation force required.

The Porsche/Audi: 2sp e-axle in Figure 1.1b [2] uses a multi-plate wet clutch and a planetary gear-set while the AVL internal R&D project (Figure 1.1a) and the Oerlikon Gratziano: 2SED (Figure 1.1c) [3] both uses a dry multiplate clutch and no planetary gears. All these three uses a dog clutch combined with a one way clutch in addition to the friction element. Dog clutches are of spline type and does not allow for any slip and is thereby not a friction element. All four transmissions in Figure 1.1 uses a layshaft but the eTwinsterX-System has coaxial input and output shafts.

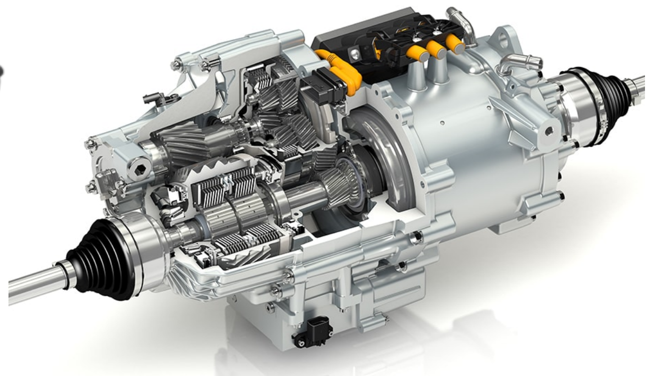


(a) *AVL internal R&D project.*



(b) *Porsche/Audi: 2sp e-axle.*

(c) *Oerlikon Gratziano: 2SED.*



(d) *GKN: eTwinsterX-System.*

Figure 1.1: *Existing two-speed concepts.*

1.2 Aim and Research questions

The aim of this research is to investigate how different friction clutch systems performs during engagement using an actuation force throughout its lifetime. This include determining the effects of wear and model the actuation on this basis in order to find out how much the system can benefit from using wear compensation. The objective is to generate new conceptual ideas for clutch wear compensation systems for typical clutch designs which enables efficient actuation solutions. Wear compensation in this context mainly involves maintaining the functionality of a clutch, regarding transferable torque vs. actuation stroke characteristics and fulfill its requirements throughout its lifetime, independent on wear. Designing a wear compensation system is thereby implicated with the following issues.

- How does different clutch system configurations differ in the way they are affected by clutch wear?
- Investigate how current solutions compensates for wear.
- Generate new conceptual solutions of wear compensation systems.
  - How much does different clutch types wear down during their lifetime?
  - How is the actuation system affected by the change in thickness of the friction discs?
    - The change in actuator displacement where the clutch starts to transfer torque (kiss point).

    * The clamp force of the friction disc(s) at a specific actuator displacement.
    * The transferable torque at a specific actuator displacement
    * The power demand of the actuator
    * The dimensions of actuator system components.
- Evaluate new concepts in order to make sure all requirements are fulfilled.

1.3 Limitations

The focus will be the actuation system of the clutch alone and how a wear compensation system is to be utilized. The idea is to evaluate concepts of wear compensation systems rather than just compare the effects of clutch wear on different kinds of systems. Due to time limitation, the work will therefore be focused on dry normally open clutches when it comes to simulations and validations of new concepts.

1.4 Report outline

Some general background to put the topic of this work into context is needed. This is done in this first chapter. In the second chapter the basic theory needed is presented. The methodology and how this work was conducted is described in the third chapter. The final results and discussion is presented in the fourth and fifth chapter respectively. The sixth and final chapter is the conclusion.

2 Theory

The main part of the theory collected during the literature studies are explained and discussed in this chapter. This include basic tribology and more specific clutch disc wear theories. Also a derivation of the relation between pressure plate force and transferable torque, is included. More theory is also introduced as the implementation of them are described in the methodology chapter.

2.1 Tribology

The mechanical wear is complicated to model but the one which is mostly used traditionally is Archard's law of wear [4] seen in equation 2.1.

$$V_i = k_i \cdot F \cdot s \tag{2.1}$$

Where V_i is the wear volume, F is the normal load, k_i is the specific wear coefficient and s is the sliding distance. Equation 2.2 show the wear displacement h_i. Where A is the area subjected to wear. In equation 2.3 the contact pressure is calculated. When combining equation 2.1, 2.2 and 2.3 the wear displacement h_i is calculated using equation 2.4.

$$h_i = \frac{V_i}{A} \tag{2.2}$$

$$p = \frac{F}{A} \tag{2.3}$$

$$h_i = k_i \cdot p \cdot s \tag{2.4}$$

The wear coefficient k_i is based on experimental data and will differ substantially with temperature, velocity etc. The large variation and complexity of k_i makes calculation of clutch disk wear difficult without experimental data of the specific clutch material and the clutch system configuration it is used in.

2.2 Clutch disc wear rate

A clutch disc with relative angular velocity to a pressure plate in contact, is referred to as clutch slip. A wear rate of a clutch disc is often stated as a volume of friction material worn of per energy developed during clutch slip. This energy is the power P integrated with respect to time during a period of clutch slip, see eq. 2.5. The wear rate is denoted as ε.

$$E_{shift} = \int P(t)dt \tag{2.5}$$

When it comes to calculating the wear of a clutch disc, a wear rate as such is easier to implement since the energy during one shift event E_{shift} can be well estimated using eq. 2.5 and since there are more data available on ε for common friction materials compared to the wear coefficient k_i from Archard's law. According to data provided by AVL, ε for a dry clutch is within the range 8 - 25 mm^3/MJ and will usually stay in the lower end of that interval if extreme operation and/or overheating can be avoided.

Regarding the wear of a friction disc in a wet clutch, the wear rate ε is more unclear. There is a small decrease in disk thickness during its first 200 cycles but no more wear after this is detectable using thickness measurements [5]. The high wear rate during the first cycles in relation to the rest of the lifetime is argued to be caused by collapse of pores in the friction material leading to permanent deformation, rather than actual wear.

2.3 Transferable torque

The relation between the pressure plate force normal to the discs and the transferable torque is important regarding these studies and is derived in this section. Considering a circular rigid disc with inner radius r_i and outer radius r_o. If the disc rotates with an angular velocity relative to another rigid surface in contact with it and thereby also a normal force present between them. A tangential force due to friction for two surfaces sliding, can be expressed as in eq. 2.6.

$$F_T = \mu F_N \tag{2.6}$$

The normal force F_N is according to force equilibrium in the normal direction, the pressure acting on the surfaces p, multiplied by the surface area A. Or more general, as in eq. 2.7.

$$F_N = \int_A p(r) dA \tag{2.7}$$

For an axisymmetric clutch disc, the integrated tangential force, will create a moment M when multiplied with a moment arm, being the radius r as in eq. 2.8. This integral can in turn be rewritten as an integral over the radius as per eq. 2.9.

$$M = F_T r = \mu \int_A p(r) r dA \tag{2.8}$$

$$M = 2\pi\mu \int_{r_i}^{r_o} p(r) r^2 dr \tag{2.9}$$

2.3.1 Unworn disc

When considering a new clutch disc, the surfaces are assumed to be perfectly flat and parallel to each other, meaning that they are in contact over the entire area. For a pair of rigid discs this results in an evenly spread out pressure $p_{unworn}(r)$ i.e. the pressure $p_{unworn}(r) = p_{unworn}$ is constant along the radius. The integrals in eq. 2.7 and eq. 2.9 can then be calculated, see eq. 2.10.

$$\begin{cases} F_{N,unworn} = p_{unworn} \int_A dA = p_{unworn} \pi (r_o^2 - r_i^2) \\ M_{unworn} = 2\pi\mu p_{unworn} \int_{r_i}^{r_o} r^2 dr = p_{unworn} \frac{2}{3} \pi\mu (r_o^3 - r_i^3) \end{cases} \tag{2.10}$$

By using the two relations in eq. 2.10, the pressure can be eliminated and a relation between the the transferable torque and the normal force for an unworn pair of surfaces in sliding contact, is achieved as in eq. 2.11.

$$M_{unworn} = \frac{2\mu\left(r_o^3 - r_i^3\right)}{3\left(r_o^2 - r_i^2\right)} F_{N,unworn} \tag{2.11}$$

2.3.2 Worn disc

The wear distribution along the radius of a friction clutch disc is continuously changing during operation. If, for a new unworn disc the pressure distribution $p(r)$ is constant, the axial wear h_i from Archard's law (eq. 2.4 will increase with the radius. This is due to the sliding distance s, see eq. 2.12, φ is the angle of rotation.

$$h_i = k_i \cdot p \cdot r \cdot \varphi \tag{2.12}$$

As an uneven wear is forming, the pressure p will no longer be constant along r. The integrals 2.7 and 2.8 is thereby no longer possible to evaluate since the pressure distribution $p(r)$ is now unknown. The pressure however can be expressed according to eq. 2.13, using Archard's law. The new coefficient k_{worn} is not dependent on r.

$$p(r) = \frac{1}{k_i}\frac{h_i}{r\varphi} = \frac{k_{worn}}{r} \tag{2.13}$$

Using the pressure from eq. 2.13, the integrals 2.7 and 2.8, now results in eq. 2.14 & 2.15.

$$F_{worn} = \int_{r_i}^{r_o} \frac{k_{worn}}{r} 2\pi r dr = k_{worn} 2\pi (r_o - r_i) \tag{2.14}$$

$$M_{worn} = 2\pi\mu \int_{r_i}^{r_o} \frac{k_{worn}}{r} r^2 dr = k_{worn}\pi\mu(r_o^2 - r_i^2) \tag{2.15}$$

k_{worn} can now be eliminated and the and the transferable torque for a worn disc is expressed according to eq. 2.16. For a new clutch disc, the moment arm is slightly longer than the mean radius of the friction material, since the friction area increases squared with the radius. For a worn clutch however, the moment arm simplifies to being exactly the mean radius. This is due to the balance between higher wear rate towards r_o, caused by longer sliding distance and increased wear rate towards r_i caused by higher pressure.

Figure 2.1 illustrates how much the two models differ at different inner and outer radius combinations. For a clutch disc it is normal that the difference between r_i and r_o is small relative to the total radius of the disc and as can be seen in the figure, this means small differences between the models.

$$M_{worn} = \frac{\mu\left(r_0 + r_i\right)}{2} F_{N,worn} \tag{2.16}$$

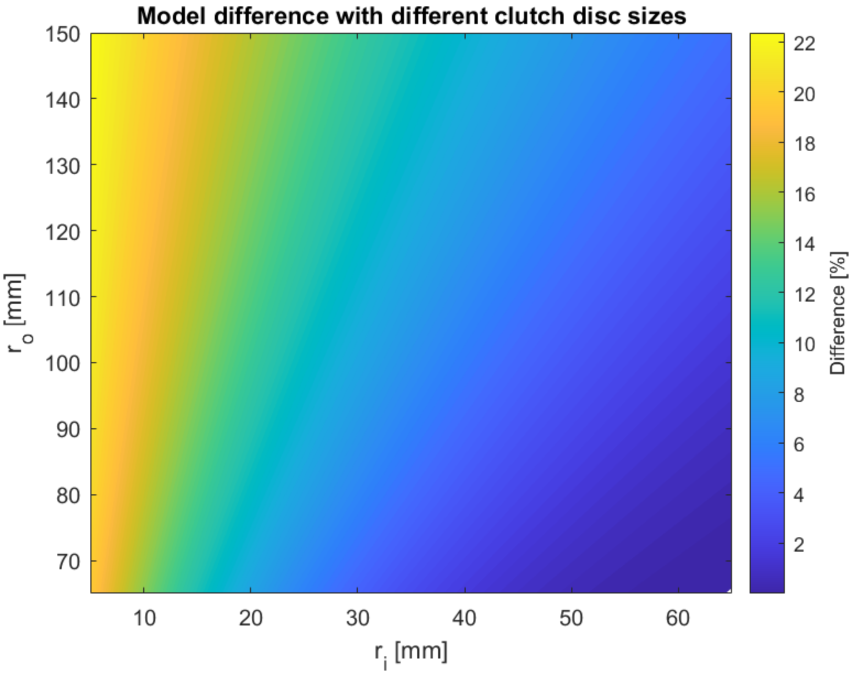


Figure 2.1: *Illustration of how the unworn clutch model differ compared to the model for a used disc.*

3 Methodology

The methodology and work flow used is described and illustrated by the flowchart in figure 3.1. This is an overview of the main processes that was used to reach the final results. This flowchart contain five different stages which will be followed each on their own and in an iterative way to make it possible to answer the questions stated in the problem statement section.

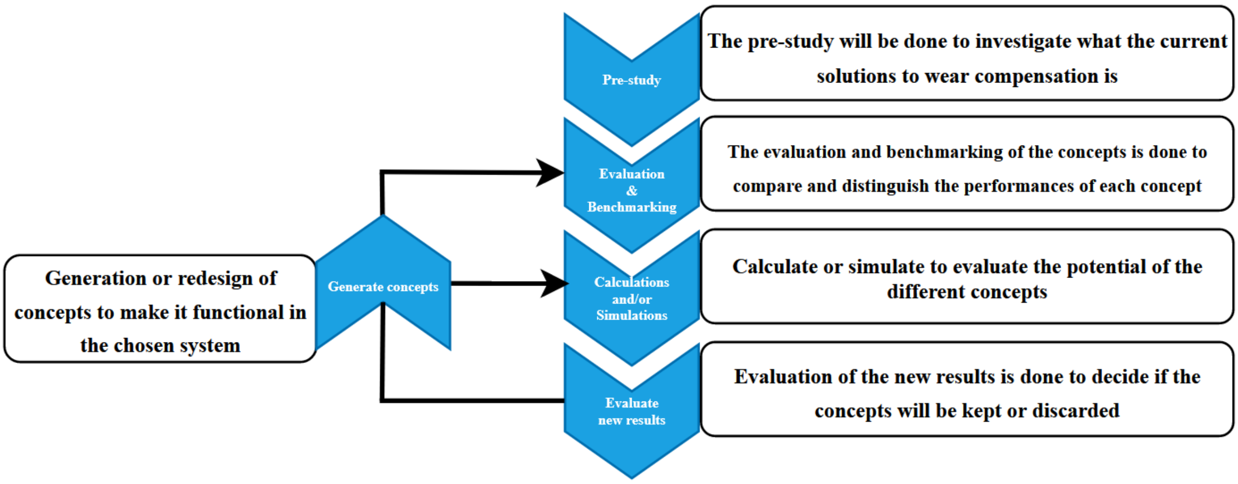


Figure 3.1: *Flowchart of the main process in this thesis.*

The first stage of this flowchart was a literature study of wear and friction modelling theories. One of the most important thing was to closer evaluate the amount of wear as well as the impact of wear, considering the four main different combinations of clutch systems illustrated in Figure 3.2. These four concepts could in turn be divided into single or multiple friction plate configurations, where the number of plates in the multi plate clutch is arbitrary. The main focus here was to investigate how the wear effects differ between the four different combinations, but how the number of discs affected this was also included. During the literature studies it became clear, as explained in section 2.2, that the wear of a wet clutch disc is little to none apart from the very first engagements. A wear compensation system for a wet clutch thereby seemed unnecessary and by focusing entirely on the dry system, the potential of a more thorough analyzes increased. Table 3.1 lists a comparison between the dry and the wet clutch systems regarding some important properties. As can be seen, one of the main issues with the dry clutch system is the wear. By decreasing or eliminating the problems that are induced by wear, the dry system can be a much more competitive alternative.

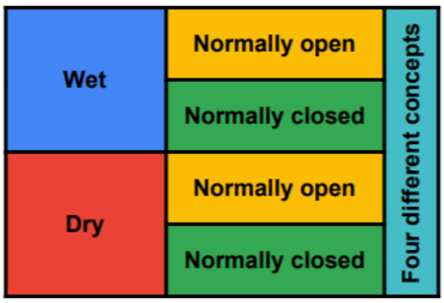


Figure 3.2: *The four main divisions of clutch systems.*

Table 3.1: Comparison between dry and wet clutch.

Property	Wet clutch	Dry clutch
Friction coefficient	-	+
Wear rate	+	-
Controllability	0	0
Cooling	+	-
Efficiency (losses)	-	+
Cost	-	+
Complexity	-	+

The litterature study did also extend to include searching for existing solutions and obtain information about these from various sources. The sources included different patents, literature provided by AVL and other literature found by searching in different databases. The next stage after finding these existing solutions was to evaluate, categorize and then do some benchmarking. This was done to further clarify the benefits and limitation of each different solution. The evaluation and benchmarking was an iterative process since different needs of benchmarking would arise.

In the next step the suitable models evaluated during the literature studies were used for calculations and simulations using the software Matlab. This was done to make sure the concept could function as intended and within the decided requirements.

When the benchmarking and evaluation was done, the next step was to evaluate these results and make a decision on what to further investigate. This included to go back in the loop and iterate the process to make the final results better. The aim was to find or generate the best solution that suits the specific application.

3.1 Existing concepts

This investigation of existing wear compensation systems was conducted to evaluate what is currently on the market. The information was collected from both patent databases and from an internal project at AVL. From these, seven concepts were selected and are described in this section.

3.1.1 Threaded rod and nut concept

The mechanism of this concept reminds of the one found in a common ballpoint pen where a part is rotated when pushed far enough. The rotating part is in this case a threaded nut that will shorten the length of a rod that connects to the actuation fork of the clutch. This rod, illustrated as yellow and labeled (310) in fig. 3.3 [6], pulls a conventional clutch fork that pushes on a clutch release bearing in a normally closed dry clutch system. The left end of rod (310) in fig. 3.3 pulls on this fork and the right end has a threaded wear compensating nut (500), illustrated as green. The yellow rod (310) is free to slide axially inside a sleeve (300) illustrated as blue in fig. 3.3, it is however, by two flats on top and bottom, constrained from rotating inside this sleeve. The blue sleeve (300) has two external pins (320) that connects to an actuation linkage. These pins (320) also holds one ball bearing each, which can travel back and forth in a slot in the housing (400), illustrated as grey in fig. 3.3. The slot constrains rotation of the blue sleeve (300) along its longitudinal axis guide (420) in the housing (400) constrains four more degrees of freedom for the blue sleeve (300), leaving only axial translation. When the actuation linkage pulls on the blue sleeve (300), it catches on the green nut (500) which pulls on the yellow rod (310). The green nut (500) has external splines (510) that meshes with internal splines (410), situated in the housing (400). The splines (510) restricts the green nut (500) from rotating, meaning that rotation of the green nut (500) only can occur when the blue sleeve (300) is fully pulled towards the right side in fig. 3.4.

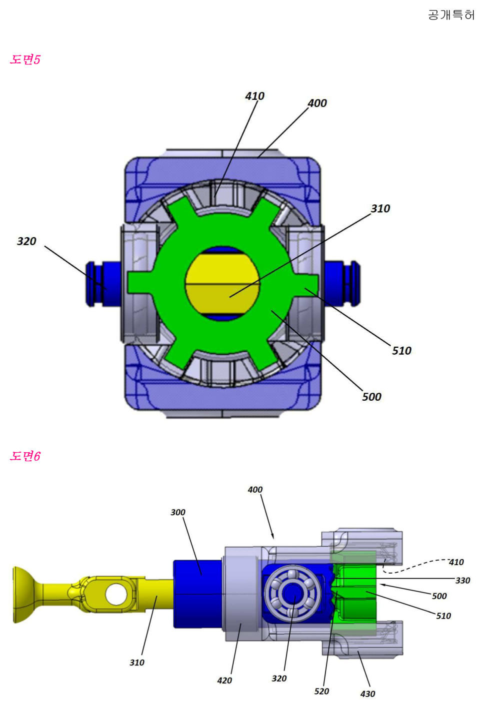


Figure 3.3: *3-D CAD of the threaded rod and nut concept. Side view with released actuation.*

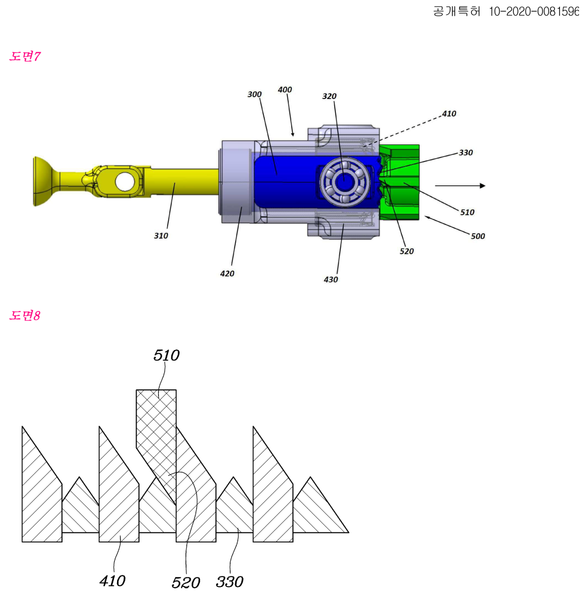


Figure 3.4: *3-D CAD of the threaded rod and nut concept. Side view with engaged actuation.*

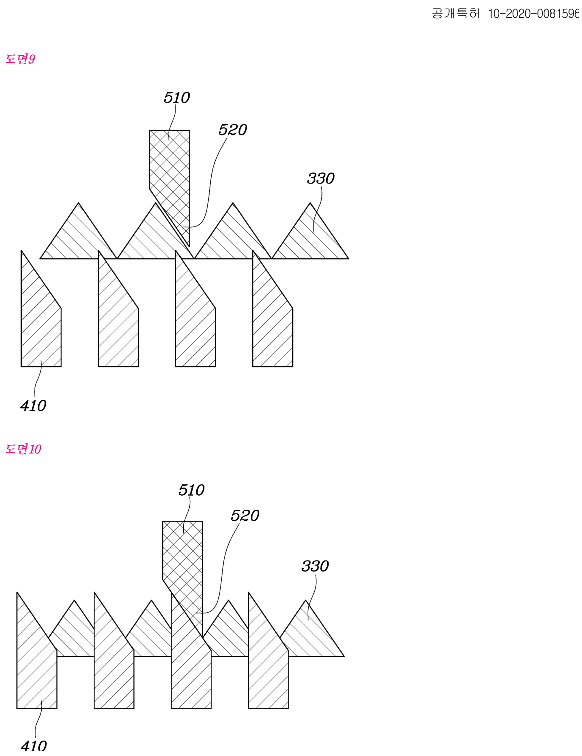


Figure 3.5: *Schematic of the threaded rod and nut rotating mechanism.*

The right end of the blue sleeve (300) has teeth (330) around the circumference which does not interfere with the internal splines of the housing (410). The teeth (330) however, does interact with the green nut (500) via an inclined surface (520) on the end of the external splines of the nut (510). When the lower part of fig. 3.4 and fig. 3.5 is studied, one can realize that when the splines of the nut (510) are pulled past the splines of the housing (410), the teeth of the sleeve (330) will, due to the inclined surface (520), make the nut turn a small amount. When the sleeve (300) with its teeth (330), then returns, the inclined surface (520) now catches on the splines of the housing (410). The nut (500) thereby rotates further as its splines (510) directs their way back to mesh with the splines of the housing (410). Due to the threaded connection between the nut (500) and the yellow rod (310), the rotation of the nut (500) results in an axial displacement of the rod (310) which could be used to compensate for wear.

3.1.2 Step concept

This concept of a clutch compensation device is using two locking rods held apart by a rod elastic member [7]. When the friction clutch plate is worn the two rods will move to the next step thus mechanically compensating for the axial wear of the clutch. Figure 3.6 show the schematic diagram of the invention. It also includes the state when the clutch is worn represented by the dotted lines. Figure 3.7 is showing the invention when the clutch actuator is active and the clutch is disengaged. Figure 3.8 is showing when the rods have reached the next step because of the friction clutch plate getting thinner due to wear. The two rods (170) are opened apart from each other by the wedge (152) compressing the rod elastic member (156) and therefore the rods will move to the next step in the housing (190). This is the concept of wear compensating regarding this invention.

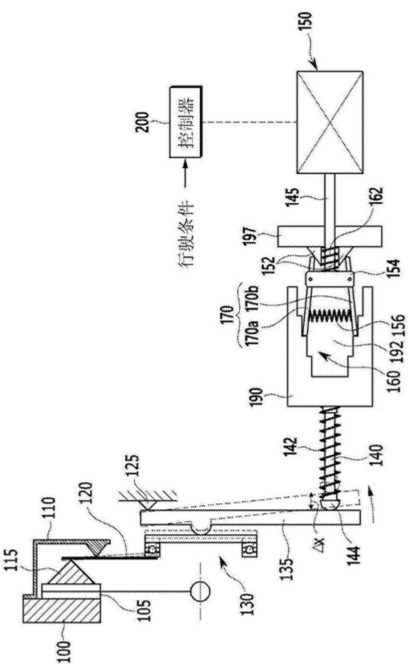


Figure 3.6: *Schematic of the step concept in an engaged clutch position.*

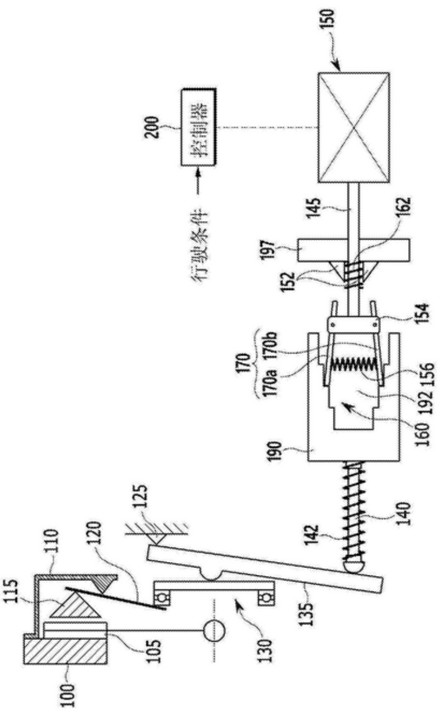


Figure 3.7: *Schematic of the step concept in a disengaged clutch position.*

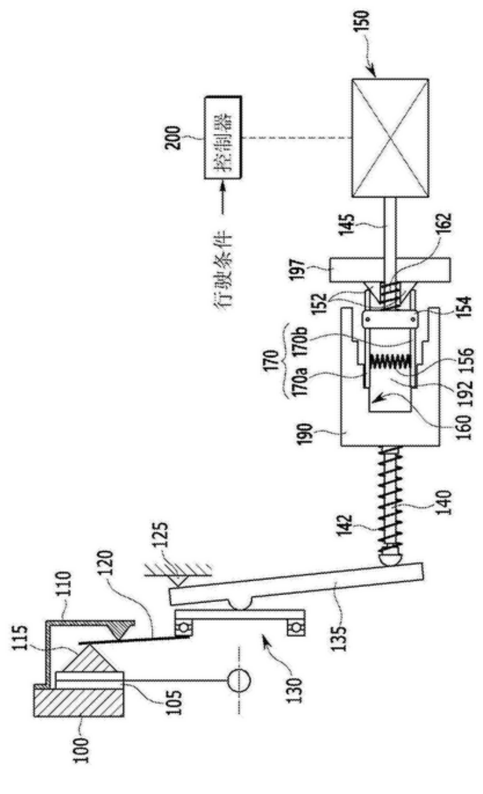


Figure 3.8: *Schematic of the step concept in an engaged position, when the clutch have compensated the wear by one step.*

3.1.3 Wedge concept

This concept of a wear compensation device uses a similar actuation linkage system as the previously described step concept. The wear compensation of this invention however is based on two pairs of inclined surfaces at two different angles [8]. Fig. 3.9 shows a schematic of the actuator assembly and the clutch in an engaged position. Fig. 3.10 shows an activated actuator (20) that via a wear compensation assembly (100) pushes on the clutch fork (17) and the clutch is thereby disengaged. In Fig. 3.11, the actuator (20) is back at a deactivated position but now in a case with a worn friction disc (11). The wear compensation assembly (100) in Fig. 3.11 have a total length reduced by Δx in order to compensate for the axial wear Δy of the friction disc (11).

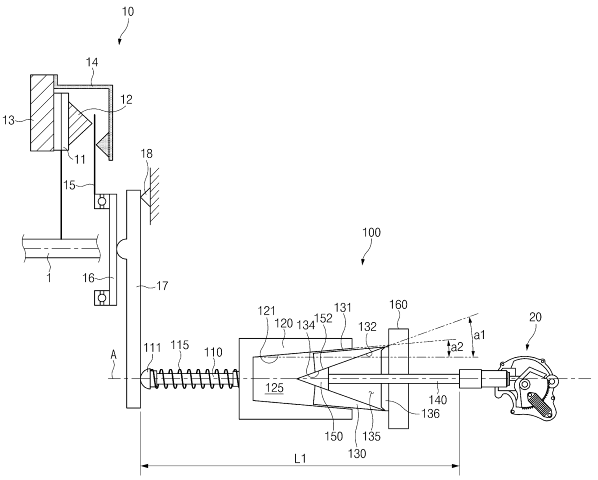


Figure 3.9: *Schematics of the wedge concept in an engaged clutch position with an unworn clutch.*

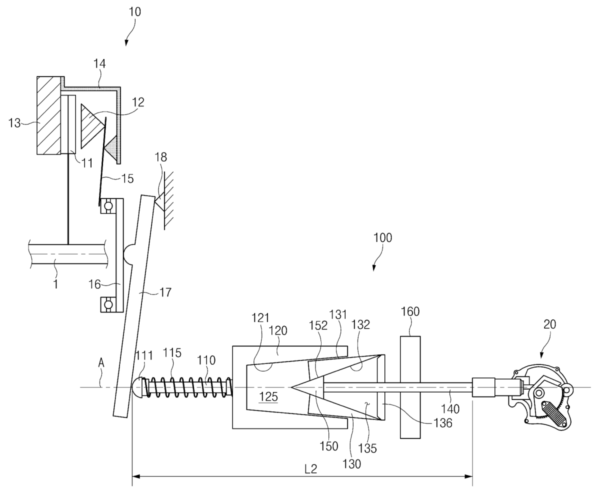


Figure 3.10: *Schematics of the wedge concept in a disengaged position with an unworn clutch.*

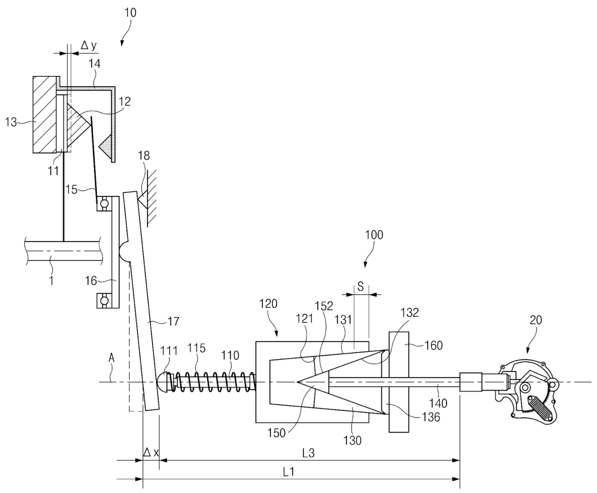


Figure 3.11: *Schematics of the wedge concept in an engaged position with a worn clutch.*

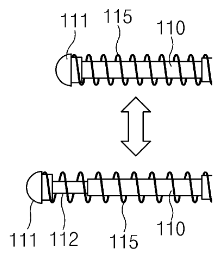


Figure 3.12: *Schematics of the push rod assembly.*

A push rod (110), is rigidly connected to an outer member (120) with an inclined inside surface (121). This surface is in sliding contact with another inclined surface (131) on an inner member (130). A guide member (160) is fixed in space (connected to a housing) and guides an actuation rod (140). The actuation rod (140) is connected to a wedge member (150) that wedges into a cavity (134) in the inner member (130), that in turn is wedged in a cavity (125) in the outer member. The outer surface (152) of the wedge member (150) matches the inner inclined surface (132) of the inner member (130). The inclined surfaces (132) and (152) are at an angle (a1) from the horizontal plane. The other pair of inclined surfaces (121) and (131) are at a different angle (a2) which is smaller than (a1). The different angles leads to different frictional forces acting in the sliding contacts that is between the two pairs of inclined surfaces (121) - (131) and (132) - (152). As the actuator rod (140) returns to its initial position and the back cover (136) of the inner member (130) comes in contact with the guide member (160) and wear has occurred as in Fig. 3.11. The outer member (120) is able to retract an extra distance (S), in order to compensate for Δx. As the actuator then advances for the next clutch release, the actuation rod (140) pushes upon the wedge member (150), wedging the inner member (130) and the wear compensation assembly (100) again moves as one unit but with a new total length (L3). The push rod (110), shown in more detail in Fig. 3.12, has an inner rod (112) and a coil spring (115). This allows the push rod to extend and retract while always keeping an elastic force between the push member (111) and the outer member (120).

3.1.4 Plate concept

This invention is based on a sandwiched plate, movable relative to two outer plates, adjusting the axial position of a push rod using two different inclined surfaces [9]. A schematic of the assembled unit with one half of the housing removed can be seen in Fig. 3.13. Fig. 3.14 shows the assembly from the other side with the entire housing (50) removed. Fig. 3.15 is a schematic of the assembly in an exploded view.

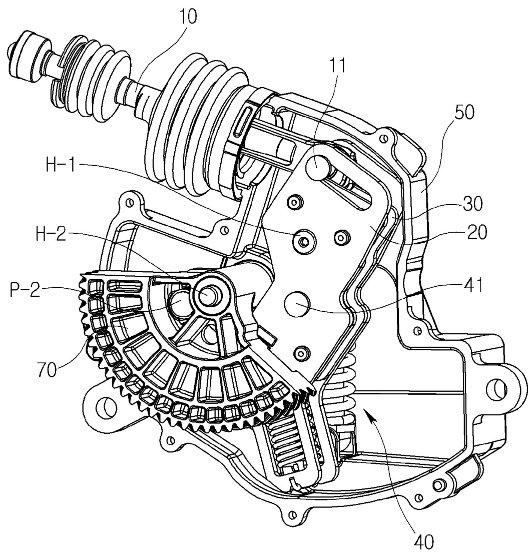


Figure 3.13: *Schematics of the adjuster plate concept.*

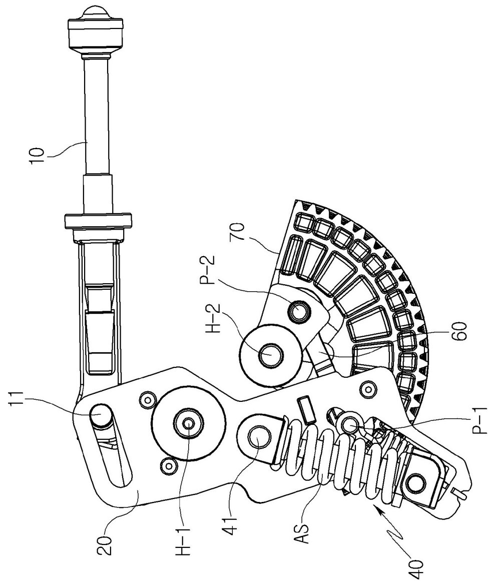


Figure 3.14: *Schematics of the adjuster plate concept with the housing removed.*

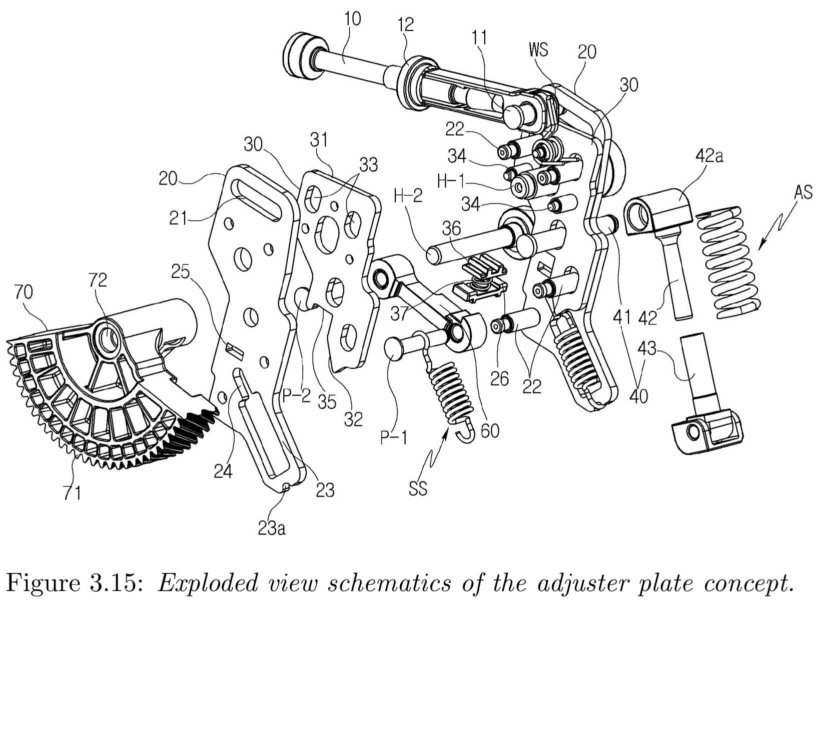


Figure 3.15: *Exploded view schematics of the adjuster plate concept.*

The housing (50) have a hinge pin (H-1) which a plate assembly pivots around, forming a lever between an connecting rod (60) and a connecting pin (11) which transfer the force to a push rod (10). An actuator can act on the connecting rod (60) via a reduction gear (70), connected to the housing (50) by another hinge point (H-2). The plate lever assembly consists of a total of four plates, where the two inner plates are referred to as an adjuster plate (30) and the two outer plates are referred to as a lever plate (20). The rightmost plate (20) in Fig. 3.15 have four locating pins (22) connected to it which penetrates the adjuster plates (30) through elongated slots (33) and sets the spacing between the lever plates (20). The adjuster plate (30), consisting out of two plates spaced apart by three spacing pins (34), is allowed to move in one direction relative to the lever plate (20) due to the elongated holes (33). The lever plate (20) has a slot (21) in which the connecting pin (11) for the push rod (10) can travel. The actuating force applied to the reduction gear (70) is transferred onto the lever plate assembly via two connecting pins (P-1) and (P-2) with the connecting rod (60) in between. The connecting pin (P-1) is, during advancement of the actuator, pushed against an inclined surface (32) on the adjusting plate (30). The connecting pin (11) is, during advancement of the actuator, wedged in the slot (21) of the lever plate (20) due to another inclined surface (31) on the adjuster plate (30). In order to disengage the clutch, an actuator turns the reduction gear (70) anti clockwise in Fig. 3.13, pushing upon the inclined surface (32) via the connecting pin (P-1), making the lever assembly pivot anti clockwise about the hinge pin (H-1). The connecting pin (11) is now pushed by the inclined surface (31), wedging it in the slot (21) and the push rod (10) is advanced. As the clutch instead is to be engaged, the actuator turns the reduction gear (70) in a clockwise direction and pulls on the connecting pin (P-1). This reduces the force on the inclined surface (32), allowing the adjuster plate (30) to move relative to the lever plate (20). This enables the connecting pin (11) to push down the adjuster plate and travel in the slot (21), allowing the push rod (10) to retard further in a case when wear of the clutch disc has occurred.

An assist spring (AS) is connected in between the housing (50) and a connecting pin (41) on the lever plate (20), reducing the torque requirement on the actuator. There is also a wedge support spring (WS) embodied in the concept, situated on the uppermost of the spacing pins (34), in order to support the connecting pin (11) in staying wedged in the slot (21). Finally, a support spring (SS) is situated in a slot (23) in the lever plate (20), supporting the connecting pin (P-1) in staying in contact with the inclined surface (32) of the adjusting plate (30).

3.1.5 Pawl concept

This concept have a wear compensation function that includes a pawl mounted on the clutch plate[10]. Figure 3.16 is a schematic overview of the concept. The diaphragm (6) and the plate (1) is fully engaged. When in this position, the control tongue (29) of the pawl (20) have a free end that bears on the tooth (30) of the ratchet wheel (13). When the clutch is disengaged, diaphragm (6) is tilted which gradually will release the plate (1) and the pawl (20). When the clutch friction lining (4) is worn the ratchet (13) in figure 3.17 will rotate around its axis (12) which also causes the worm gear (14) to rotate. The ring member (9) have an inclined surface and is held in place by the diaphragm (6). When the worm is free to rotate it will make the plate (1) exit cover (5) and compensate a bit for the wear when the opposing ramps are mating. This engagement and separation process may need many repetitions to fully compensate for the wear. Figure 3.18 and 3.19 show the pawl in more detail.

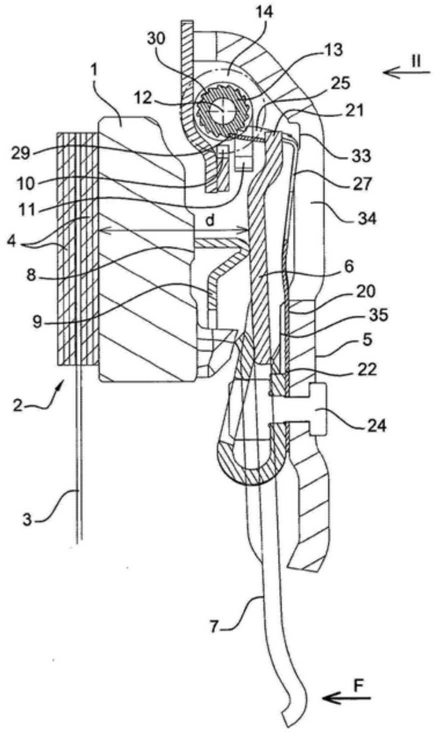


Figure 3.16: *Schematic overview of the pawl concept.*

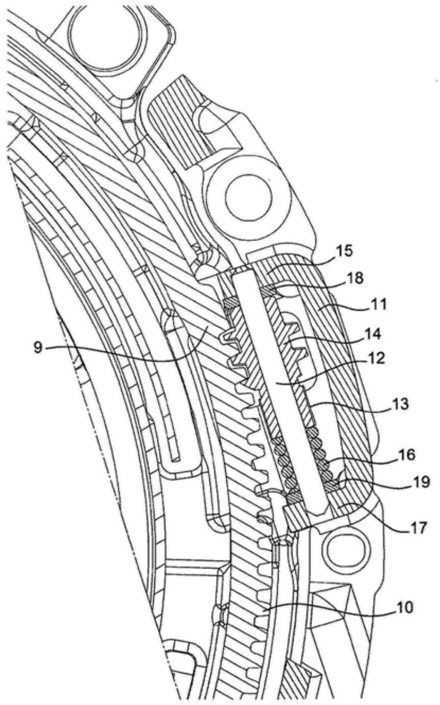


Figure 3.17: *Schematic of the pawl concept and the worm gear.*

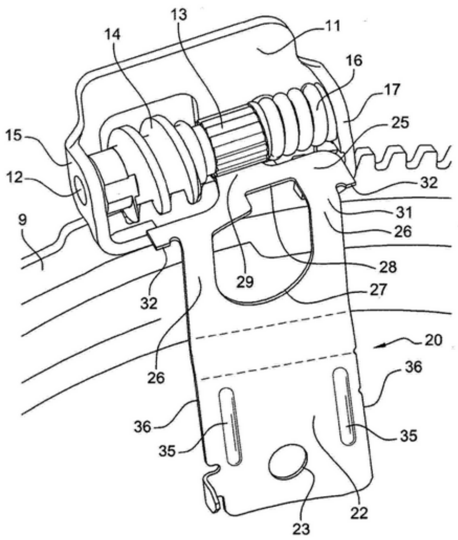


Figure 3.18: *Schematicw of the pawl concept and the worm gear.*

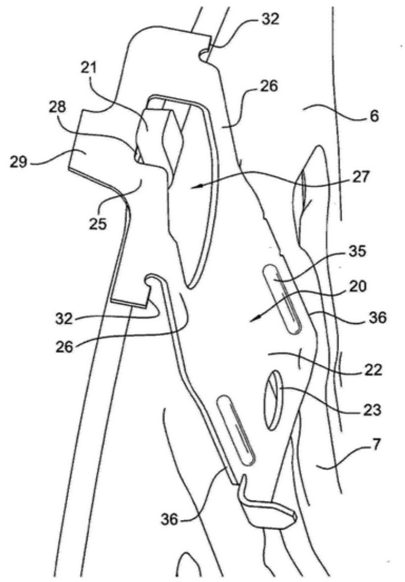


Figure 3.19: *Schematic of the pawl and the tooth.*

3.1.6 Lever concept

This concept uses a wear detection lever to move the diaphragm spring to adjust the gap between the pressure plate and clutch plate when the friction material is worn [11]. Figure 3.20 is showing the longitudinal view of the concept. The clutch housing hosts the pressure plate (14) which will be pressed against the flywheel (12) by the membrane spring (16). The clutch pins (20) is the pivot points of the diaphragm spring (16). Attached to the pressure plate (14) is a carrier (22) with an adjustment device (24). The adjustment lever (26) and the cutting ring (18) is acting as a wear compensation device. The carrier (22) also carry a worm (30) which will rotate around the axial advance element axis (28). The worm (30) is used as the axial advance element

(30). when rotating because of the worm (30), the inclined surface (118) which is sitting on top of the inclined surface (38) make the cutting ring (18) move axially to compensate for the wear seen in figure 3.24.

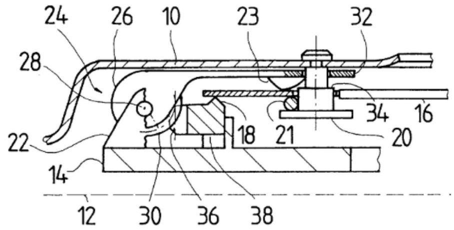


Figure 3.20: *Schematic overview of the lever concept.*

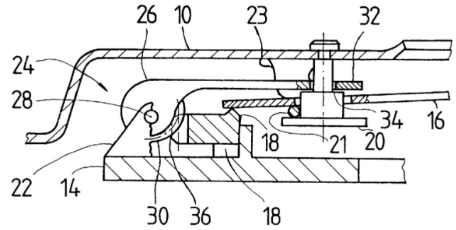


Figure 3.21: *Schematic of the lever concept when the clutch is engaged.*

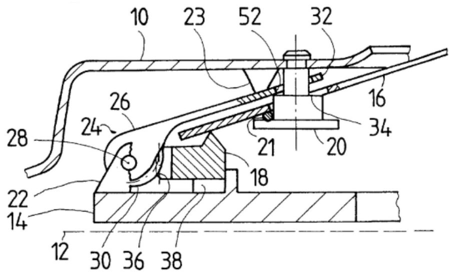


Figure 3.22: *Schematic of the lever concept when the clutch is engaged and wear occurs.*

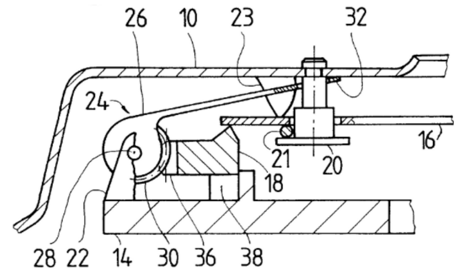


Figure 3.23: *Schematic of the lever concept after the wear adjustment have occured and the clutch is disengaged.*

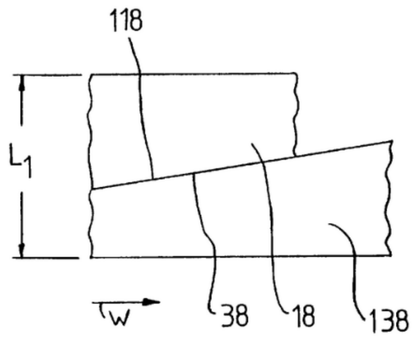


Figure 3.24: *Schematic of the lever concepts inclined surfaces between (18) and (38).*

Figure 3.21 is showing the concept when the clutch is engaged. When wear occurs and the wear adjustment is working as shown in figure 3.22 when the clutch is engaged. Figure 3.23 is showing the concept after the previous wear adjustment when the clutch is disengaged.

3.1.7 AVL force limiter concept

An AVL R&D project, uses a force limiter concept in a normally open clutch, in order to compensate for clutch disc wear. The clutch is in this case operated by a pivoting fork, actuated using an inclined shift drum turned by an electric actuator. The basic mechanics of the force limiter are shown schematically in Fig. 3.25.

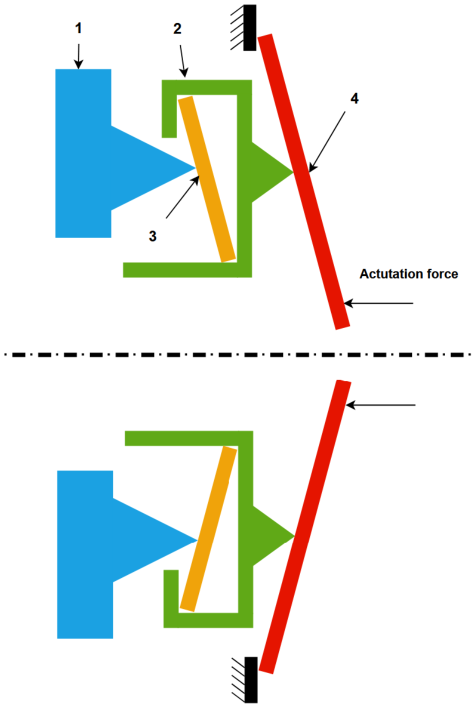


Figure 3.25: *Schematics of the AVL concept.*

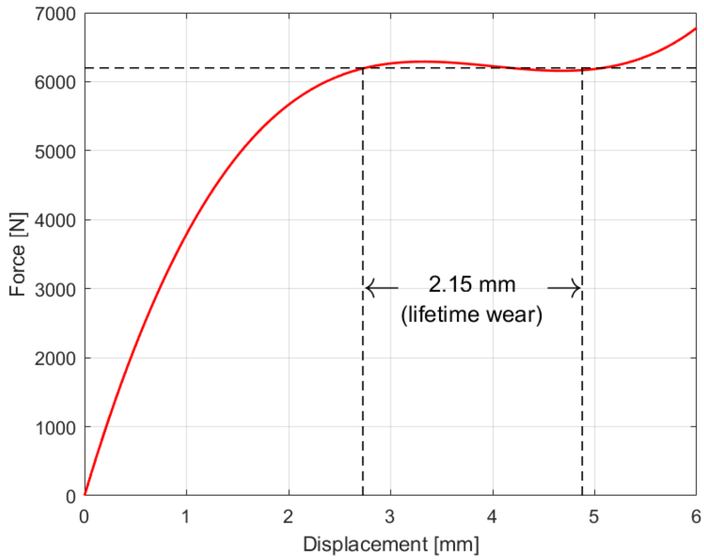


Figure 3.26: *Force vs. displacement characteristics of an example diaphragm spring to be used as a force limiter.*

The clutch, as in a typical normally open clutch, is engaged by an actuation force applied near the inner circumference of a pressure plate lever (4). But instead of having the diaphragm spring pressing directly upon the pressure plate (1), this concept has a carrier member (2) situated in between the pressure plate (1) and the pressure plate lever (4). The force limiter (3) is a diaphragm spring, that sits preloaded to a desired force inside the carrier member (2). As a result, the carrier member (2) acts as a solid spacer between the pressure plate lever (4) and the pressure plate (1) as long as the force transferred not exceeds the preload force.

Due to the characteristics of a diaphragm spring (see Fig. 3.26), the force limiter (3) can be designed to have close to a constant force vs. displacement curve, in the proximity after the point of which it is preloaded to. Regarding the characteristics of the example spring shown in Fig. 3.26, a preload force of 6.2 kN is marked with a dashed line. The diaphragm spring (4) is thereby allowed to be compressed beyond the point where the desired force (the preload force) is reached, without the force continuing to increase. The actuation system can, because of the force limiter concept, be designed to have enough travel in order to engage a worn clutch and still function properly without excessive forces on the parts when the clutch is new.

The force diagram in Fig. 3.26 is based on the relation expressed in equation 3.1 [12], with F being the axial force acting on the inner radius of the diaphragm spring and δ being the axial displacement. The parameters used are listed in table 3.2. The dimensions are chosen to match the clutch size considered in the dry clutch wear calculations in previous section. The Young's modulus E and the shear modulus G, are taken from Swedish Standards Institute [13]. The Poisson's ratio ν is calculated from the relation shown in equation

3.2.

$$F = \frac{E\delta}{(1-\nu^2)r_o^2}\left[C_1(h-\delta\left(h-\frac{\delta}{2}\right)t + C_2t^3\right], \quad \begin{cases} C_1 = \left(\frac{\alpha+1}{\alpha-1} - \frac{2}{\log\alpha}\right)\frac{\pi\alpha^2}{(\alpha-1)^2} \\ C_2 = \frac{\pi\alpha^2\log\alpha}{6(\alpha-19^2} \\ \alpha = \frac{r_o}{r_i} \\ \beta = \frac{h}{r_o-r_i} \\ \varphi = \frac{\delta}{r_o-r_i} \end{cases} \tag{3.1}$$

$$\nu = \frac{E}{2G} - 1 \tag{3.2}$$

Table 3.2: Parameters used for the example diaphragm spring.

Parameter	**Notation**	**Value**	**Unit**
Inner radius	r_i	50	[mm]
Outer radius	r_o	75	[mm]
Spring height	h	4	[mm]
Spring disc thickness	t	2.7	[mm]
Young's modulus	E	210	[GPa]
Shear modulus	G	80	[GPa]

3.1.8 Compilation and Categorizing

The concepts were divided into three different categories of wear compensation, where the chosen seven concepts of wear compensation is sorted to their respective category. The different categories are shown in the list below.

- **Compensation by displacement**
  - Compensation near discs
    - Pawl concept
    - Lever concept
  - Compensation near actuator
    - R&N concept
    - Step concept
    - Wedge concept
    - Plate concept
- **Force compensation**
  - Compensation near discs
    - AVL concept

3.2 Expected axial wear during the lifetime of the clutch

The axial wear over the lifetime of a traditional single disc dry clutch is around 2 mm [14]. How the life time expectancy may differ for a multi plate clutch system like the one to be investigated, is hard to predict, it depends of the size of the clutch, materials, amount of slipping, temperatures and of the climate in which it is operating. Using the wear rate ε, an estimation of the axial clutch wear of this system was carried out and will be explained in this section.

To estimate the total wear of a clutch, the number of actuations during expected life cycle need to be considered. Using data provided from AVL and reasonably assumptions shown in table 3.3, the total wear can be calculated based on the energy input to the clutch. The calculations are based on a system where the friction clutch is located at the intermediate shaft as in Figure 1.1a. The energy during one shift event E_{shift} will be the time integral of the input power $P(t)$ according to eq. 2.5.

Table 3.3: Parameters used to compute total clutch wear.

Parameter	Notation	Value	Unit
1st gear ratio	i_1	3.5	[-]
2nd gear ratio	i_2	2	[-]
Final gear ratio	i_{final}	4	[-]
Wheel radius	R_{wheel}	0.35	[m]
Active friction material	afm	0.86	[-]
Friction disc outer diameter	d_o	150	[mm]
Friction disc inner diameter	d_i	120	[mm]
Wear rate	ε	10	[mm^3/MJ]

The power is the input torque $M(t)$ times the relative velocity of the discs in the clutch $\omega_{rel}(t)$ (clutch slip speed) and is approximated using a mean value of the torque M and a mean value of the slip speed, being half the maximum value $\omega_{rel,max}$, see eq. 3.3. This maximum of the slip speed occurs at the very beginning of the shift and when the shift is completed the slip speed is zero. The slip speed can be calculated as in eq. 3.4, where $Shift\ point$ is the vehicle speed of when the shift occurs. The total wear volume of friction material V_{wear} is calculated using the accumulated shift energy and the wear rate ε, see eq. 3.5, n_{shifts} is the number of shifts. The axial clutch wear is finally calculated by dividing the wear volume with the active friction area as in eq. 3.6.

$$E_{shift} = \frac{M \cdot \omega_{rel} \cdot \Delta t}{2} \tag{3.3}$$

$$\omega_{rel} = \frac{Shift\ point}{R_{wheel}} \cdot \frac{30}{3.6\pi} \cdot \frac{(i_1 - i_2) i_{final}}{i_1 - i_2} \tag{3.4}$$

$$V_{wear} = \varepsilon \cdot E_{shift} \cdot n_{shifts} \cdot 10^{-3} \tag{3.5}$$

$$Axial\ clutch\ wear = \frac{V_{wear}}{afm \cdot A_{friction}} \tag{3.6}$$

The shift events are categorized into seven different cases, regarding different levels of load and whether it is an uppshift or a downshift. The cases will differ considering shift point, clutch slip speed, total slip time Δt, average clutch torque and the number of shifts. The accumulated energy $\sum E_{shift}$ is calculated for each case separately followed by the wear volume and the axial wear. The results are shown in 3.4.

Table 3.4: Gear shift scenarios with resulting axial clutch wear.

Scenario	Direction	Load	Shift point [km/h]	Clutch slip speed [rpm]	Total slip time [ms]	Average clutch torque during shift [Nm]	Shift energy [kJ]	Number of shifts	Wear volume [mm^3]	Axial clutch wear [mm]
1	Up	Light load	80	2425,22	350	100	4,44	60000	2666,67	0,47
2	Up	Medium load	80	2425,22	350	190	8,44	50000	4222,22	0,74
3	Up	High load	80	2425,22	350	280	12,44	20000	2488,89	0,43
4	Up	Full load	110	3334,67	450	250	19,64	5500	1080,36	0,19
5	Down	Light load	70	2122,07	300	100	3,33	27500	916,67	0,16
6	Down	Medium load	70	2122,07	300	190	6,33	10000	633,33	0,11
7	Down	Full load	45	1364,19	300	700	15,00	2000	300,00	0,05
								175000	**12308,13**	**2,15**

The wear rate and the friction disc diameters was changed to investigate how these affected the total axial wear of the clutch as seen in figure 3.27. The wear rate ε can differ between 8 to 25 depending on different parameters, including material properties and operating temperature. A value between 8 to 12 is the one used for clutches during normal driving operations and will be used to compare different size of clutches as seen in figure 3.27.

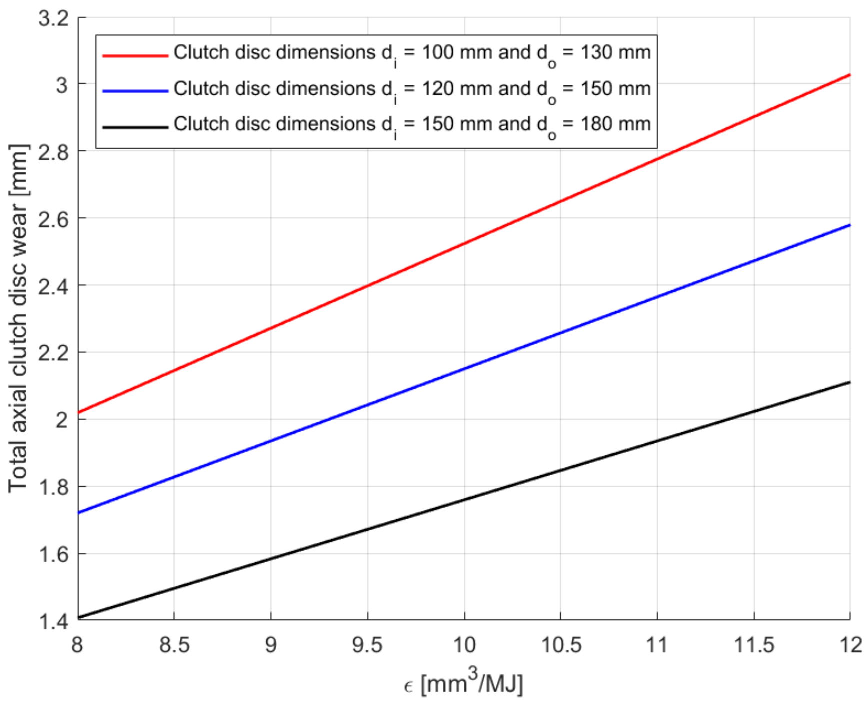


Figure 3.27: *Illustration of axial clutch wear during a complete life cycle.*

3.3 Conceptual system evaluation

This section will go through the iterative process that was conducted in order to arrive at one final concept of wear compensation with the potential to operate properly in a specified actuation system. This included the steps of designing an actuation system linkage with specifications based on clutch system requirements and the needs of the different wear compensation concepts. It also included design evaluations of the concepts, both regarding robustness and regarding possibilities to adopt it into different system configurations. Finally the actuation process was simulated, so that the system performance could be understood and that the concepts and actuation system design could be reevaluated and redesigned if deemed necessary.

3.3.1 Actuation system design

In order to deduce reasonable requirements for a wear compensation mechanism, a complete clutch actuation system needs have to be specified. The specifications will be set both concerning the clutch system requirements and the actuator requirements they result in. Since the main focus is on the wear compensation and not the rest of the actuation system, the specifications of the clutch disk package and the actuator will be set and not varied between different concepts.

3.3.1.1 System schematics

The system used from this on forward consists of several different parts. Figure 3.28 is showing the schematics of the complete system without the wear compensation device. The clutch package is in the lower left corner in the figure and consists of several clutch discs and plates, where the rightmost of them being the pressure plate. The black box is the throwout bearing. Between the throwout bearing and the pressure plate is, for the normally open system a number of pressure plate levers and for the normally closed system, a diaphragm spring. The part illustrated as brown is the actuator which by a gear drives an intermediate shaft with an input gear illustrated green and an output gear, illustrated yellow. The output gear drives a ramp that in turn operates the clutch actuation fork which apply force on the throwout bearing.

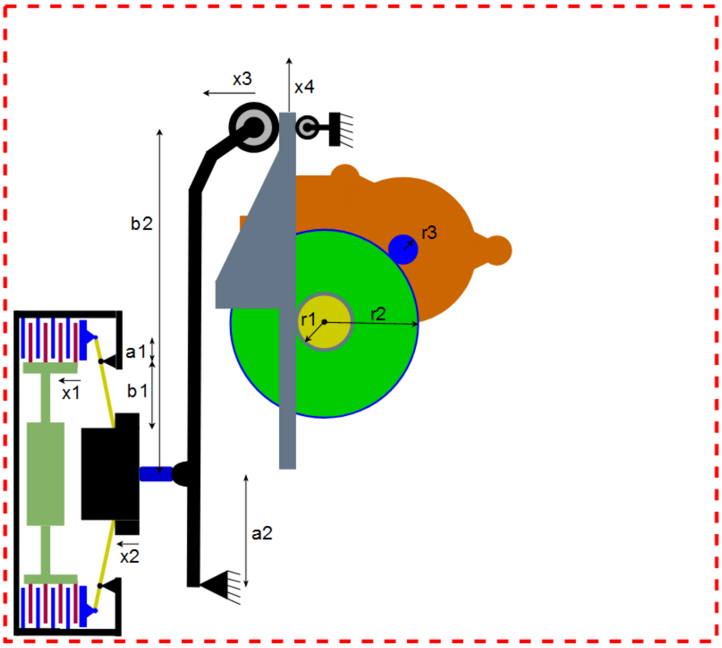


Figure 3.28: *Schematics of the complete shift system.*

The different systems that have been used are four in total. This is with normally open or normally closed clutch in combination with compensation near discs or near actuator. Figure 3.29 is showing the position of compensation near discs with a normally open clutch. Figure 3.30 is showing the position of compensation near discs with a normally closed clutch. The compensation device in these two cases will be positioned close to the discs, preferably in connection with the pressure plate.

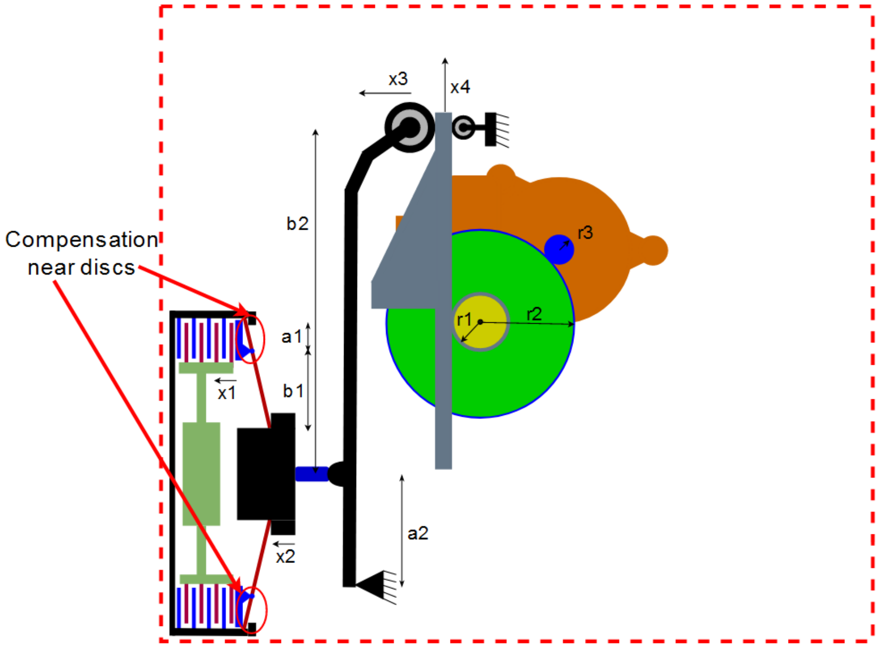


Figure 3.29: *Schematics of compensation near discs with a normally open clutch.*

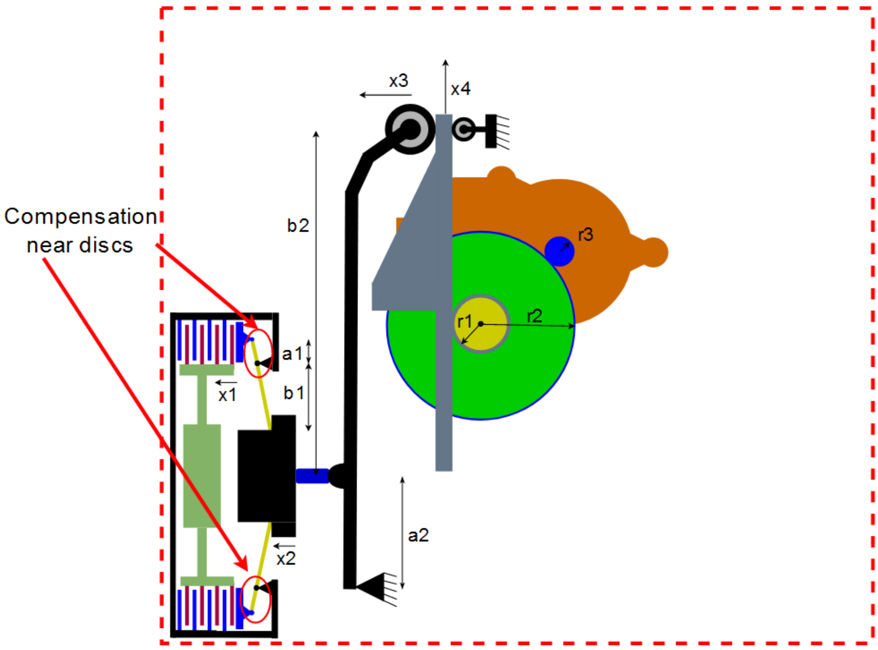


Figure 3.30: *Schematics of compensation near discs with a normally closed clutch.*

Figure 3.31 is showing the position of compensation near actuator with a normally open clutch. Figure 3.32 is showing the position of compensation near actuator with a normally closed clutch. The compensation device is positioned between the actuation ramp and the clutch actuation fork.

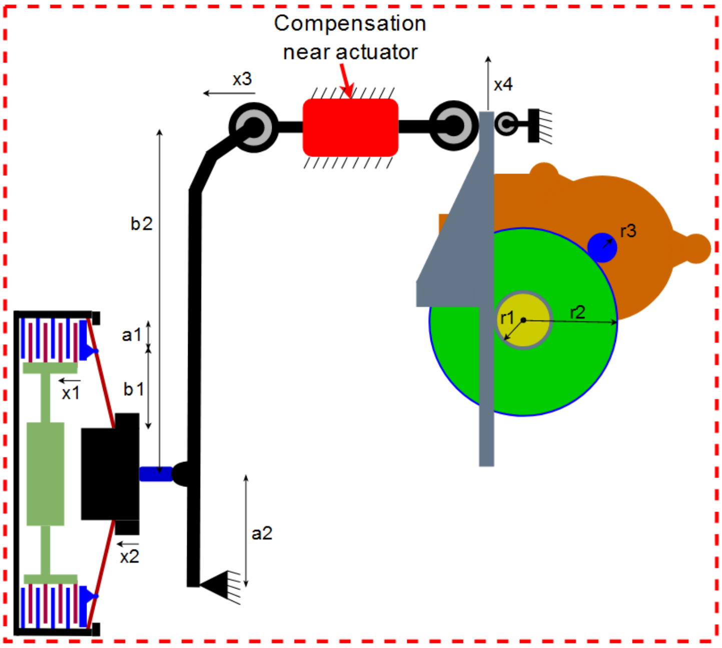


Figure 3.31: *Schematics of compensation near actuator with a normally open clutch.*

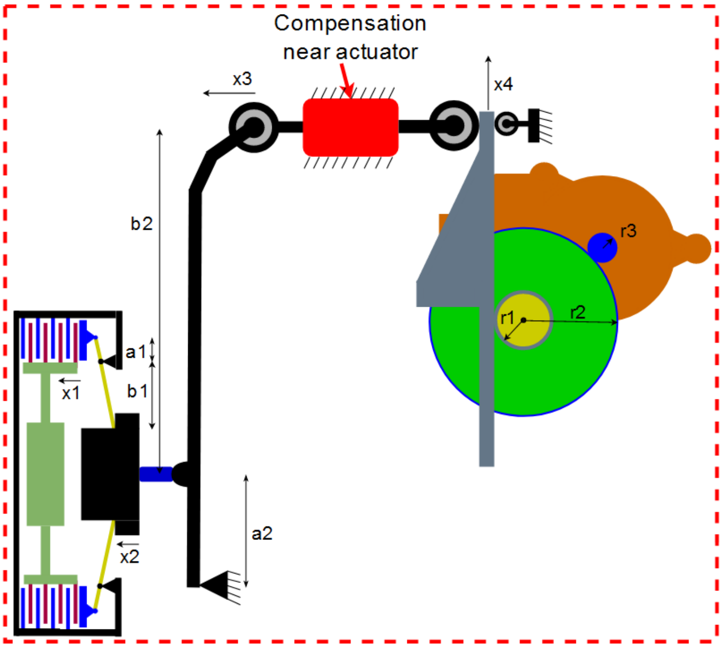


Figure 3.32: *Schematics of compensation near actuator with a normally closed clutch.*

3.3.1.2 Requirement specification

The requirement specification is divided into three layers, L0 to L2, where each level has a set of requirements and a set of specifications. Figure 3.33 illustrates the three levels and how different requirements and specifications affect each other.

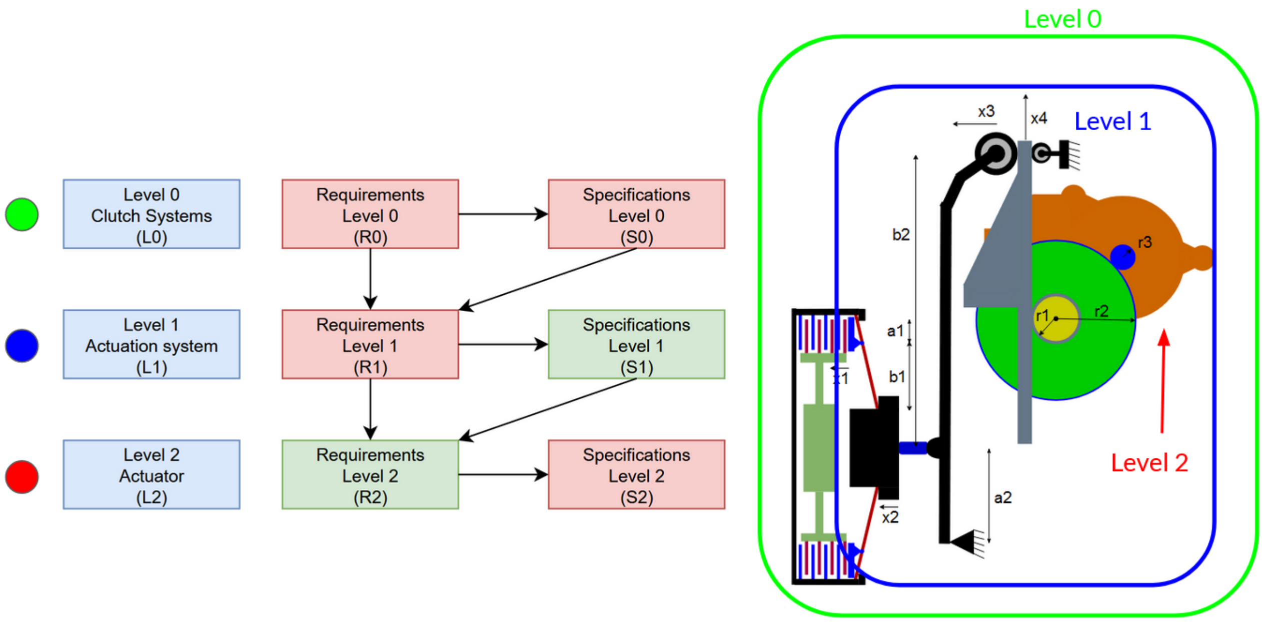


Figure 3.33: *Requirement specification overview.*

Requirements R0 for the clutch system are chosen based on application and benchmarking. Based on R0, also the specifications S0 are established, see table 3.5. R0 and S0, now sets the requirements for the actuation system R1. R1.1 for instance is calculated based on R0.1 and S0 with an safety factor of 1.3. Similarly, R1.2 comes from R0.4 and S0.3, where an initial axial gap of 0.3 mm per friction disc is considered. Note that R1.2 only states the displacement needed for the pressure plate. The performed displacement of the rest of the actuation system need to correspond to more than this in order to meet R1.1. R1.3 and R1.4 comes directly from R0 and the rest of R1 are chosen in a similar manner as the ones in R0. As mentioned, the actuator specifications are also determined. They can be seen in S2 and are based on data received from AVL. The actuation system is designed so that S1 can meet R1 and so that the resulting requirements on the actuator R2, can be met by the specifications S2.

Table 3.5: Requirement specification.

	Req. ID	Requirement Statement	Value	Unit	Spec. ID	Specification statement	Value	Unit
L0 (clutch system)	R0				S0			
	R0.1	Transferable torque	> 700	[Nm]	S0.1	Outer diameter	170	[mm]
	R0.2	Shift time	< 600	[ms]	S0.2	Total thickness (disc pack)	45	[mm]
	R0.3	Total number of shifts	>175000	[-]	S0.3	Number of friction discs	4	[pcs]
	R0.4	Wear management	>2.15	[mm]	S0.4	Coefficient of friction	0.30	[-]
L1 (actuation system)	R1				S1			
	R1.1	Peak actuation force required upon pressure plate	5620	[N]	S1.1	Actuation system radial measurement		[mm]
	R1.2	Pressure plate displacement*	>3.35	[mm]	S1.2	Actuation system axial measurement		[mm]
	R1.3	Number of shift cycles	>175000	[-]	S1.3	Actuation system weight**		[kg]
	R1.4	Shift time	< 600	[ms]	S1.4	Total ratio between actuator and pressure plate		[rad/mm] [kN/Nm]
	R1.5	Additional radial space	< 165	[mm]				
	R1.6	Additional axial space	< 330	[mm]				
	R1.7	Added weight	< 3.50	[kg]				
L2 (actuator)	R2				S2			
	R2.1	Actuator torque output		[Nm]	S2.1	Actuator stall torque	1.10	[Nm]
	R2.2	Actuator power output		[W]	S2.2	Actuator no load speed	5000	[rpm]
	R2.3	Actuator weight		[kg]	S2.3	Actuator peak power output	144	[W]
	R2.4	Number of shift cycles	>175000	[-]	S2.4	Actuator weight	1.00	[kg]

*R1.2 Itself only makes sure that all gaps between clutch discs can be closed at all states of wear.

**Actuator weight excluded.

3.3.2 Comparison between the different concepts

To better give an overview of the different concepts a schematic map of the different alternatives was made and can be seen in Figure 3.34. The concepts are first divided between a system with normally open or normally closed clutch. The next divider is where the compensation concept is located. The compensation concept could either be located near the clutch discs or near the actuator further up the system. The concepts is then sorted under the previous stated dividers. The concepts are designed to be positioned near the clutch discs or near the actuator.

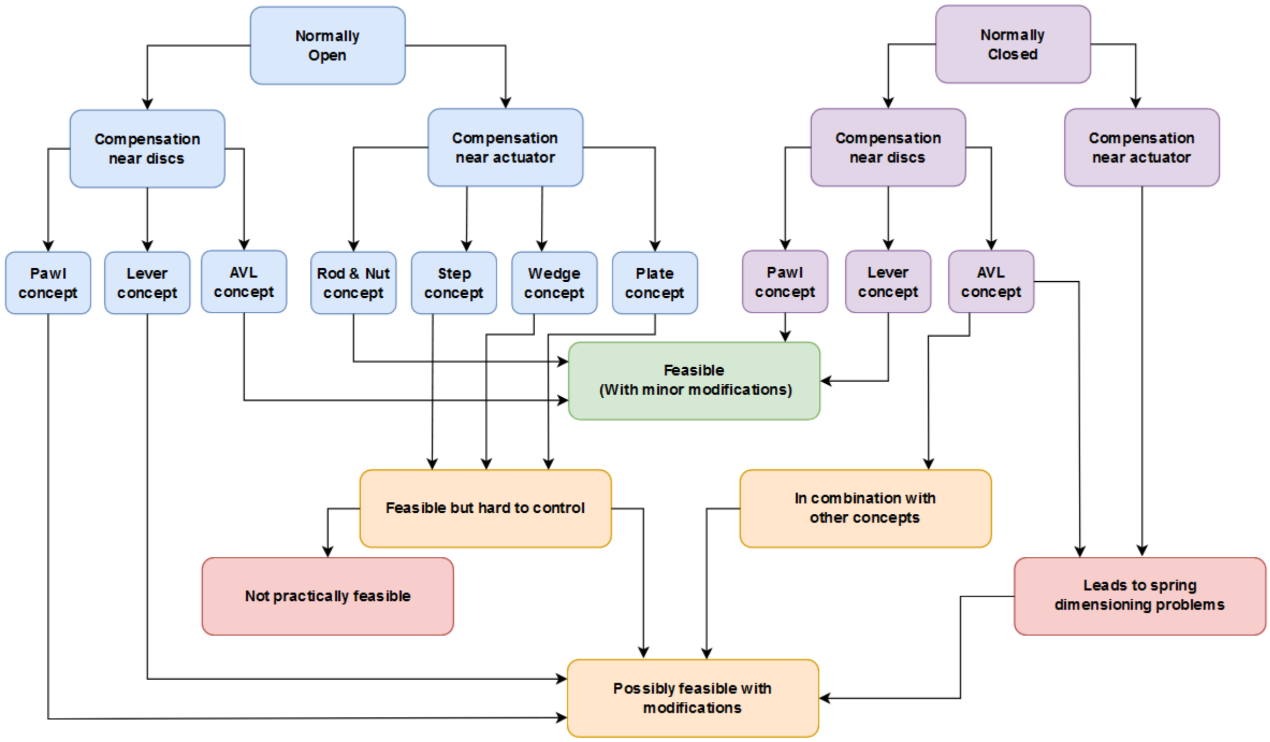


Figure 3.34: *Tree showing the different concepts and if they will function in the system.*

3.3.2.1 Normally open with compensation near discs

The Pawl concept is considered to be feasible with modifications. Since this concept in its original configuration is designed for a normally closed clutch, modifications needs to be done. This is something that can seemingly be challenging due to the location of the mechanism but it should not be completely discarded as a possible solution. The same applies for the Lever concept since it is meant to function in a similar manner in the same location.

The AVL concept is designed for a normally open clutch and should work as intended with or without some minor modifications.

3.3.2.2 Normally open with compensation near actuator

The Step concept, Wedge concept and Plate concept are similar in their intended functions. When the actuator is moving far enough backwards when returning from the shift sequence the wear compensation should adjust itself. The problem with these concepts is that it is one way adjusting. In a normally open system, if there is some friction further down the actuation linkage, the actuator could move separately from the rest of the system, resulting in an unwanted compensation. With these current solutions it is not possible to reverse this compensation without taking apart the compensation device. These concepts are thereby potentially hard to predict and were considered hard to control.

Unlike these other three near actuator concepts, the Rod & Nut concept did not have the risk of unwittingly compensate and thereby is controllable and robust. Modifications were considered to be needed but not to cause any direct problems.

3.3.2.3 Normally closed

In the normally closed system the pressure plate force is provided by a diaphragm spring. This kind of clutch system have been around for a long time in manual transmissions and thereby have a highly specialized design to work with the large displacements of a manual clutch linkage. In order to be able to do calculations on these systems, a mathematical model for the diaphragm spring is needed. It turned out to be hard finding enough data to be able to model a specialized spring like this. If instead a standard diaphragm spring were implemented, as the one in section 3.1.7, it lead to dimensioning problems in combination with the near actuator concepts. This since the displacement at the throwout bearing still will be large. The same problem also applied for the AVL force limiter since it does not include any displacement compensation. Even though the Pawl and Lever concepts should not be a problem to implement in a normally closed clutch, a decision to only focus on the normally open system was taken.

3.3.3 Calculations and simulations

Simulations was utilized in order to understand the relations between the forces and displacements in the system and how these are effected by wear. Using Matlab calculations, three clutch actuation and deactuation sequences were simulated where forces and displacements were plotted versus time. Figure 3.35 show how this looks like for, in this case, a system without any wear compensation. Figure 3.36, show the same simulation, but with a time independence for extra comparability. In this graph the sequences are marked on the x-axis. The displacements x_1, x_2, x_3 and x_4 are as described in Figure 3.29 and 3.31. θ_1, θ_2 and θ_3 corresponds to angular displacements of the three gears with radii r_1, r_2 and r_3. See appendix A for all graphs.

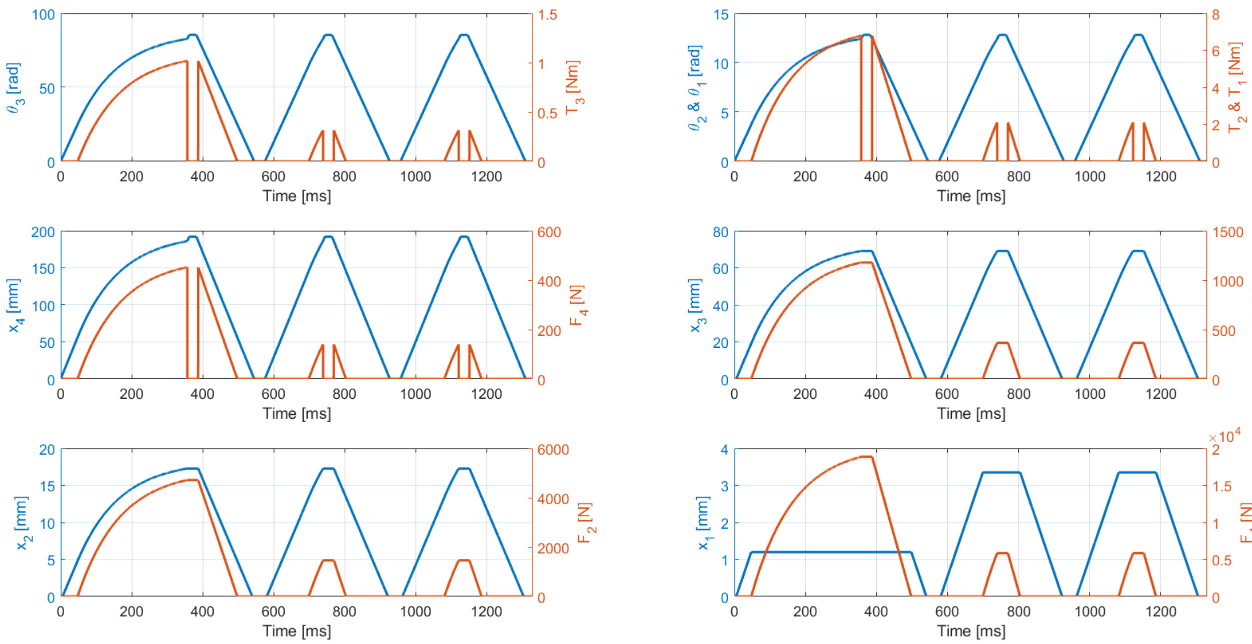


Figure 3.35: *Force and displacement simulation of a normally open clutch system without wear compensation.*

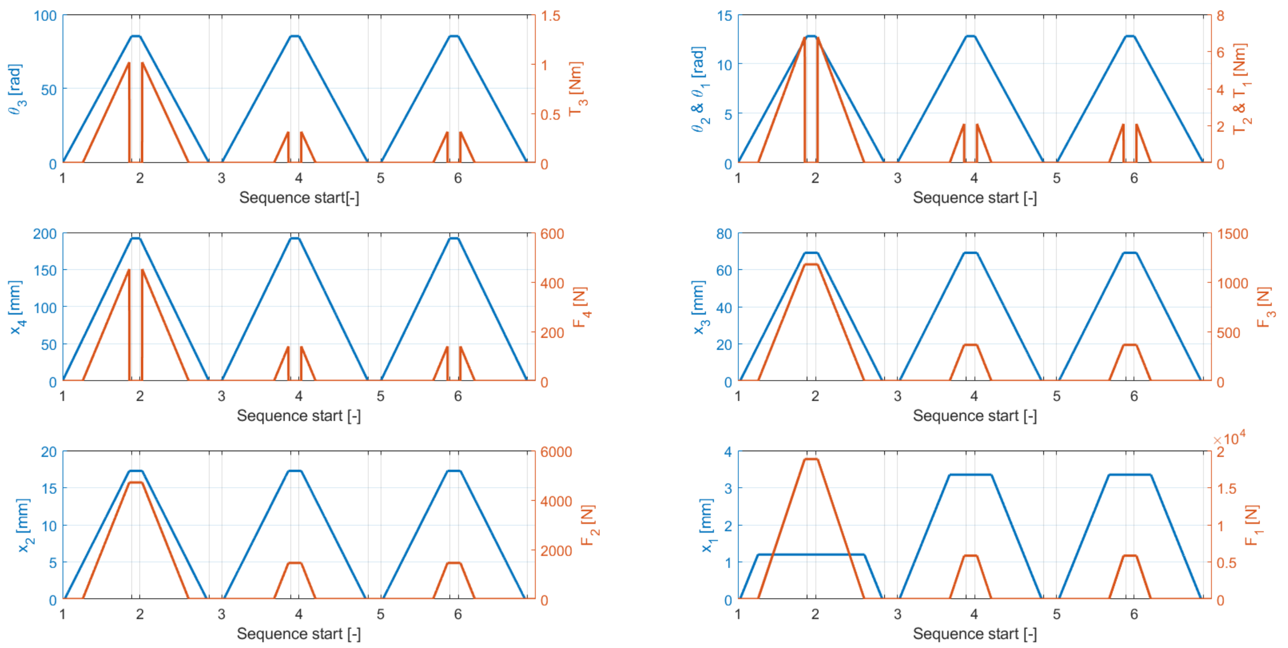


Figure 3.36: *Force and displacement, time independent simulation of a normally open clutch system without wear compensation.*

The simulations are based on an actuator with a linear torque versus speed curve and specifications as per table 3.5. The speed of the actuator was set based on the actuator torque T_3 in the previous time step. Using the actuator speed, the displacements θ_3 to θ_1 and then x_4 to x_1 were calculated with the different gear ratios and ramp specifications. The displacement of the pressure plate x_1 was specified not to exceed the total initial gap plus the current amount of wear. A variable δ was used to express the bending of the pressure plate levers to in turn be able to calculate the pressure plate force F_1. The $\delta(t)$ was calculated as in equation 3.7 i.e. the difference between the displacement that would apply for x_1 with no clutch discs stopping it (first term) and the actual displacement of x_1 (second term). Equation 3.8, which was used to calculate the force F_1 is based on equation 3.9[15]. This is an elementary case of a simply supported beam in bending with a point load P acting on a distance αl from the first end of the beam and a distance βl from the second end of the beam with $\alpha + \beta = 1$ and l is the length of the beam. $\delta(\alpha)$ will then be the deflection of the beam at a distance α from the first end of the beam. With F_1 calculated this way, the rest of the forces and torques all the way back to the actuator was calculated using the same ratios as for the displacements. All other parts of the actuation system were considered rigid in these analyzes and no friction or inertia was taken into account, hence a quasi-static model.

$$\delta(t) = x_2(t)\frac{a_1}{a_1 + b_1} - x_1(t) \tag{3.7}$$

$$F_1(t) = \delta(t)\frac{n_{lever} 3EI(a_1 + b_1)}{(a_1 b_1)^2} \tag{3.8}$$

$$\delta(\alpha) = \frac{Pl^3}{3EI}\alpha^2\beta^2 \tag{3.9}$$

Also structural evaluation of the parts was needed to make sure nothing was at risk of yielding. The limiting factor in this regard was the pressure plate levers. In order to evaluate the stress levels in these levers, the software Ansys was used for finite element analyzes.

3.3.4 Reevaluation and redesign

Based on the calculation results, both the wear compensation concepts and the actuation system specifications were altered in order to fulfill all requirements. This, as mentioned earlier was an iterative process where problematic concepts were discarded as other performed better. Towards the end of the research, Catia V5 was used to further evaluate the actuation system properties and to investigate the function of the actual wear compensation mechanism.

4 Results

In this chapter the final design of the actuation system used for these wear compensation studies, will be explained and the specifications will be presented together with the resulting requirement specification. Also the wear compensation mechanism will be presented regarding its design and how it is meant to function.

4.1 Actuation system linkage

The actuation system linkage presented in the following subsections is based on the schematics in section 3.3.1.1. The exact specifications are the result of the needs and requirements of the chosen wear compensation concept, the actuator and other limiting factors of the actuation system. The parameters are confirmed using Matlab calculations and simulations.

4.1.1 Performance

The results of the axial wear of the discs proved to have a big impact on the forces acting within the actuation system. As described in section 3.3.3, Matlab calculations were used to evaluate the performance of the actuation system in order to tune the parameters. In this section, the effect of wear is presented in Figure 4.1. This one shows two actuations by a system with no wear compensation. To the right is when the clutch is at the end of its life cycle with 2.15 mm of axial clutch wear. As can be seen, an angular displacement of almost 90 radians is required at the actuator in order to reach the required pressure plate force of 5620 N. The reason for the large displacement is partly because of the extra displacement needed at the pressure plate due to wear but also because of the torque limitations of the actuator. As can be seen in the left of Figure 4.1, which shows the actuation of a new clutch with no wear, the actuator here reaches a torque of almost 1.1 Nm.

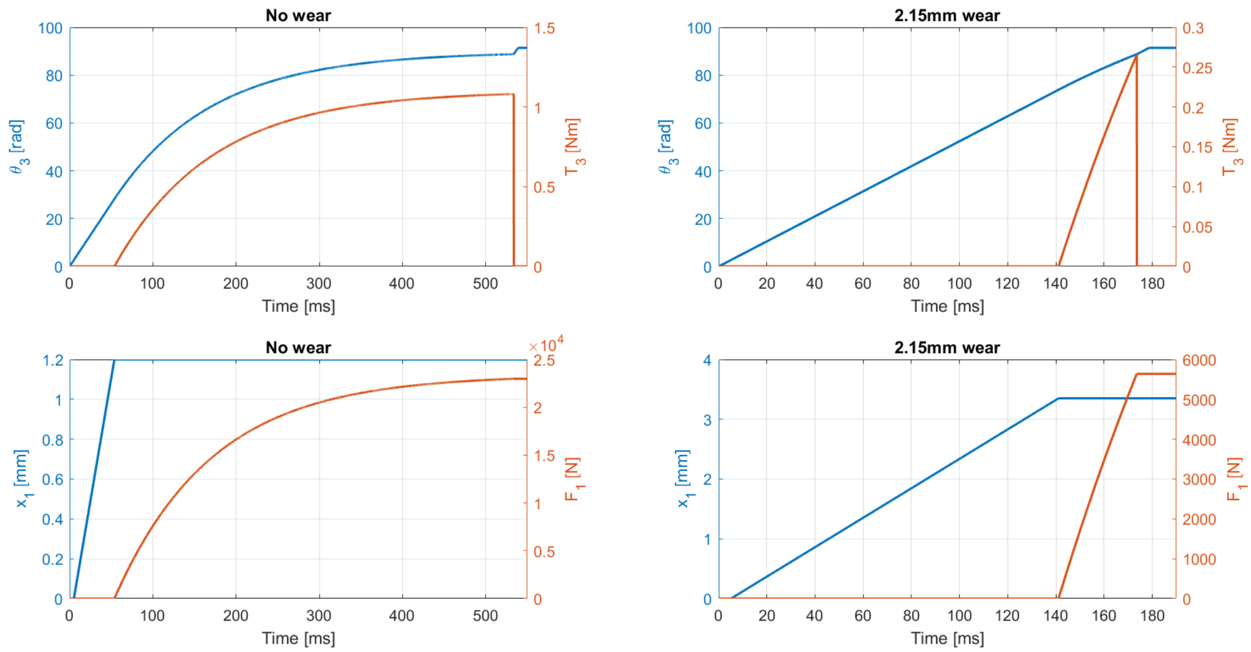


Figure 4.1: *Clutch actuation performance without wear compensation.*

Together with the parameters that is presented in the upcoming sections, the needed ramp height for the non compensating system is 64.8 mm. In order to reach this ramp height without exceeding the stall torque of the actuator, a pitch angle of 18.5° is needed for the ramp, hence the large angular displacement at the actuator. The fastest actuation time reached with an unworn clutch is achieved with a pitch angle of 14.5°. this however further increases the actuation times with a worn clutch.

Using a force limiting spring, like the AVL force limiter concept, reduces the actuation times significantly. The same ramp height is still needed since there is no displacement compensation but the pitch angle of the ramp can be increased to 35°. This results in the total angular displacement needed at the actuator being reduced to 46 radians. The actuation times with a worn clutch can be improved further by increasing the pitch angle to 50°. This of course also reduces the time to reach the required pressure plate force with a new clutch. The total sequence time however, is increased. Figure 4.2 illustrates the benefits of the force limiter.

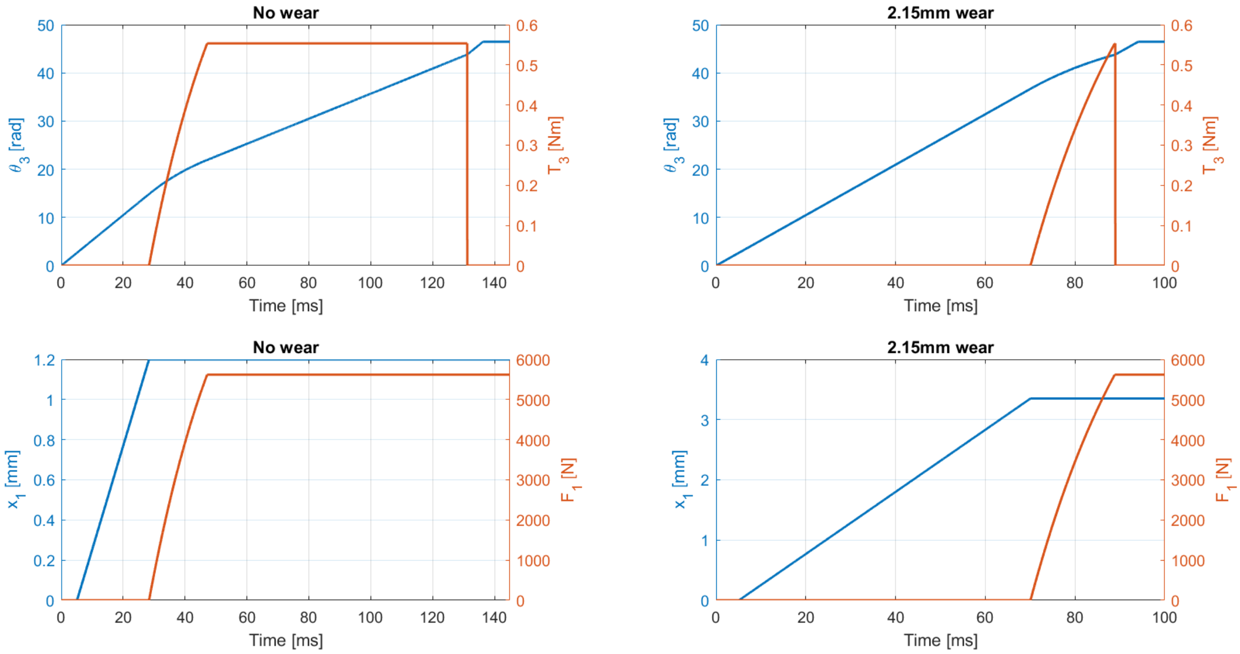


Figure 4.2: *Clutch actuation performance with force compensation.*

In Figure 4.3, the effects of displacement compensation for wear is shown. One important result here is that the ramp height can be decreased to 30.5 mm. With a pitch angle of 35°, the resulting actuator displacement angle is just under 25 radians. As can be seen, the actuation sequence is the same for the worn disc as for the unworn, with the pressure plate having a starting displacement of 2.15 mm relative to the actuator. Also in this case there is room for improvement by increasing the pitch angle to 50° but in order to accommodate for inertia and friction in the system, a pitch angle of 35° is used for the design process of the wear compensation concept. The resulting actuation times are compiled in table 4.1 and more graphs showing the different pitch angles can be found in appendix B. All results here are based on using the full power output available from the actuator.

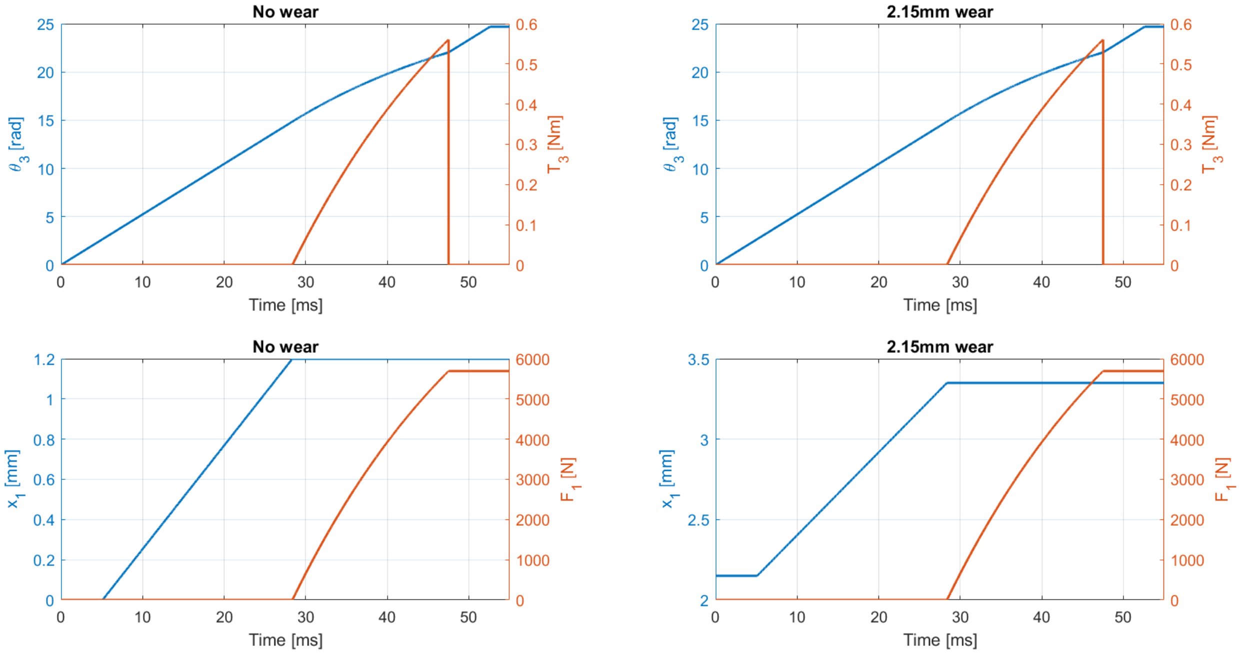


Figure 4.3: *Clutch actuation performance with displacement compensation.*

Table 4.1: Time performance comparison.

	Pitch angle	Unworn clutch	Worn clutch
Total actuation sequence time			
No compensation	14.5°	354 ms	227 ms
	18.5°	539 ms	179 ms
Force compensation	35°	136 ms	94 ms
	50°	213 ms	67 ms
Displacement compensation	35°	53 ms	53 ms
	50°	43 ms	43 ms
Time to reach required transferable torque			
No compensation	14.5°	109 ms	221 ms
	18.5°	86 ms	173 ms
Force compensation	35°	47 ms	89 ms
	50°	37 ms	61 ms
Displacement compensation	35°	47 ms	47 ms
	50°	37 ms	37 ms

4.1.2 Wear check cycle

The design of the actuation system does not allow most of the investigated concepts to ever perform a compensation automatically as wear increases. The reason for this is the design and utilization of the ramp, where the maximum displacement of the parts from the ramp to the pressure plate will be predetermined and will not be effected by wear of the friction discs. This being that the actuator, during all actuation sequences will move the ramp until the linkage rest on the plateau. Utilizing a ramp plateau like this is important in order to relieve the actuator from a high torque output after the sequence is completed.

In order to allow the wear compensation system to function as intended, a new actuation cycle needed to be added. This cycle, called a "wear check cycle" is possible by the addition of a second inclined part of the ramp. for this cycle, instead of performing a predefined displacement, a predefined torque will be applied to the actuator. The result of this will be an extra displacement that will depend on the state of clutch wear. This cycle is only meant to be performed occasionally, preferably when the vehicle is being turned off, since all parts then should be up to operating temperature.

4.1.3 Pressure plate levers

From evaluating the stress levels in the levers using the Ansys software, It was found that the first lever specifications lead to yielding. The biggest change made here to accommodate for the yielding was to increase the number of levers from 15 to 20. The rest of the parameters can be seen in table 4.2. It was also found that the preferred material for the levers is a high grade steel with a Young's modulus of 210 GPa. Figure 4.4 shows the resulting stress distribution on a lever from a load that corresponds to a throwout bearing force of 1800 N.

Table 4.2: Pressure plate levers specifications.

Parameter description	**Notation**	**Value**	**Unit**
Vertical distance from outer support to pressure plate contact point.	a_1	15	[mm]
Vertical distance from pressure plate contact point to throwout bearing.	b_1	45	[mm]
Lever beam width.		10	[mm]
Lever beam thickness.		1.8	[mm]
Number of levers.	n_{lever}	20	[pcs]
Lever ratio	i_{lever}	4	[-]

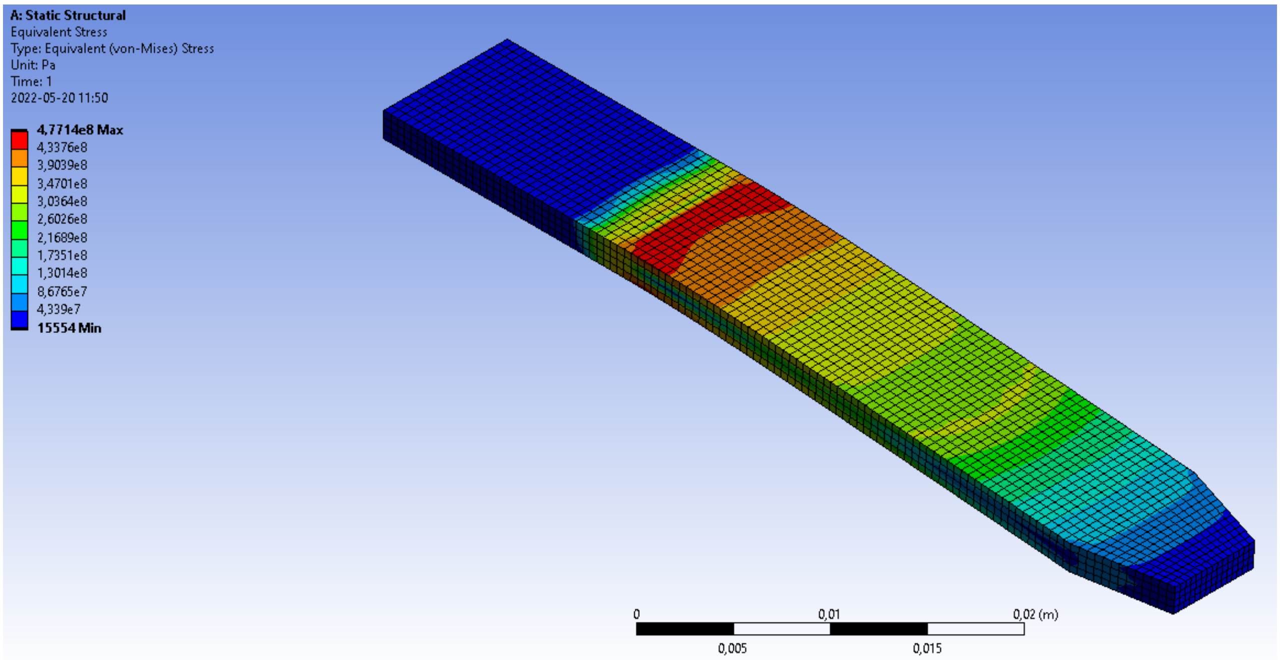


Figure 4.4: *Ansys calculation on stress distribution over the pressure plate lever.*

4.1.4 Clutch actuation fork

The fork has three interfaces with three other parts, being the wear compensation device, the throwout bearing and a pivot point. The connection to the wear compensation device consists of a wrist pin, held in place by two external circlips. Two tabs on the fork will provide a sliding contact with the throwout bearing which also allows for vertical movement to a small extent. The same applies for the pivot point but in this case the tab is a ball ended rod on a fixture that connects to the transmission housing. The fork is provided with an elongated groove that can move on this ball ended rod. The fork also is provided with a hole in which an output/input shaft can pass through. The design can be seen in Fig. 4.9 and the specifications can be seen in table 4.3.

Table 4.3: Clutch fork specifications.

Parameter description	Notation	Value	Unit
Vertical distance from lower support to throwout bearing contact point.	a_2	60	[mm]
Vertical distance from throwout bearing contact point to upper revolute joint.	b_2	180	[mm]
Horizontal meassurment from throwout bearing contact point to upper revolute joint.		60	[mm]
Fork weight.		0.27	[kg]
Lever ratio.	i_{fork}	4	[-]

4.1.5 Ramp specifications

As described in section 4.1.2, an extra section of the ramp was added. This section was given a lower inclination angle of 30° compared to the main part of the ramp with an inclination angle of 35°. This was made so that the extra displacement during the wear check cycle would not lead to an increased torque requirement for the actuator. The rest of the specifications of the ramp can be seen in table 4.4.

Table 4.4: Ramp specifications.

Parameter description	Notation	Value	Unit
Length of first and second plateau.		12	[mm]
Height of second plateau, relative the first.		30.5	[mm]
First inclination angle.	α_1	35	[deg]
Second inclination angle.	α_2	30	[deg]
Ramp weight.		0.18	[kg]

4.1.6 Layshaft gears and actuator shaft drive gear

The specifications for the gears from the first simulations described in section 3.3.3, did not need to be altered for the final design. The specifications can be seen in table 4.5 and the design can be seen in Fig. 4.8 & 4.9.

Table 4.5: Gear specifications.

Parameter description	Notation	Value	Unit
Actuator drive gear radius.	r_3	7.5	[mm]
Layshaft input gear radius.	r_2	50	[mm]
Layshaft output gear radius.	r_1	15	[mm]
Total layshaft weight.		0.57	[kg]
Total gear ratio (actuator to ramp).	i_{gears}	0.44	[rad/mm]

4.1.7 Requirement specification

Table 4.6 shows the compleate requirement specification with a summary of the actuation system specifications S1 and the resulting requirements for the actuator R2. In total, the actuation system adds 160 mm of radial measurement to the clutch disc package and 301 mm to the axial measurement. The wear compensation mechanism constitutes for 120 mm of the axial meassurement. The total mass of the system excluding the actuator was estimated to 2.03 kg using the CAD-model. This resulted in a weight requirement of 1.47 kg maximum for the actuator. The total force/displacement ratio results in 10.16 radians of actuator angle displacement per mm of pressure plate displacement, or 10.16 kN of pressure plate force per Nm of actuator torque. The resulting actuator torque requirement is 0.88 Nm with a safety factor of 1.5 included to accommodate for friction and inertia in the system. To meet the shift time requirement of 600 ms an actuator power output requirement of 37 W minimum is needed.

Table 4.6: Requirement specification.

	Req. ID	Requirement Statement	Value	Unit	Spec. ID	Specification statement	Value	Unit
L0 (clutch system)	**R0**				**S0**			
	R0.1	Transferable torque	> 700	[Nm]	S0.1	Outer diameter	170	[mm]
	R0.2	Shift time	< 600	[ms]	S0.2	Total thickness (disc pack)	45	[mm]
	R0.3	Total number of shifts	> 175000	[-]	S0.3	Number of friction discs	4	[pcs]
	R0.4	Wear management	> 2.15	[mm]	S0.4	Coefficient of friction	0.30	[-]
L1 (actuation system)	**R1**				**S1**			
	R1.1	Peak actuation force required upon pressure plate	5620	[N]	S1.1	Actuation system radial measurement**	245 (160)	[mm]
	R1.2	Pressure plate displacement*	> 3.35	[mm]	S1.2	Actuation system axial measurement**	305 (301)	[mm]
	R1.3	Number of shift cycles	> 175000	[-]	S1.3	Actuation system weight***	2.03	[kg]
	R1.4	Shift time	< 600	[ms]	S1.4	Total ratio between actuator and pressure plate	10.16	[rad/mm] [kN/Nm]
	R1.5	Additional radial space	< 165	[mm]				
	R1.6	Additional axial space	< 330	[mm]				
	R1.7	Added weight	< 3.50	[kg]				
L2 (actuator)	**R2**				**S2**			
	R2.1	Actuator torque output	> 0.88	[Nm]	S2.1	Actuator stall torque	1.10	[Nm]
	R2.2	Actuator power output	> 36.93	[W]	S2.2	Actuator no load speed	5000	[rpm]
	R2.3	Actuator weight	< 1.47	[kg]	S2.3	Actuator peak power output	144	[W]
	R2.4	Number of shift cycles	> 175000	[-]	S2.4	Actuator weight	1.00	[kg]

*R1.2 Itself only makes sure that all gaps between clutch discs can be closed at all states of wear.

**Added space within parentheses.

***Actuator weight excluded.

4.2 Wear compensation mechanism

The wear compensation suggested for this system is based on the ”rod and nut” concept. In order to adopt the concept for operation in a normally open clutch system and work together with the actuation linkage, the orientation of the push rod inside the nut and sleeve is reversed, see Figure 4.5. The housing is redesigned for an actuation force application onto the sleeve that allow for the simple design of the ramp. A ball bearing at the end of the sleeve allow for low friction force transfer from the ramp. The number of splines on the nut and inside the housing is reduced to three in order to increase robustness and get more rotation from one compensation event. The decrease of rod length with this design, during one compensation, is 0.67 mm. A smaller step than this was considered to risk compensation at every wear check cycle. This reduction of splines also enables the sleeve to only have three small inclined indentations to guide the nut and apply a small torque on it, which will make it turn when the splines goes past the housing.

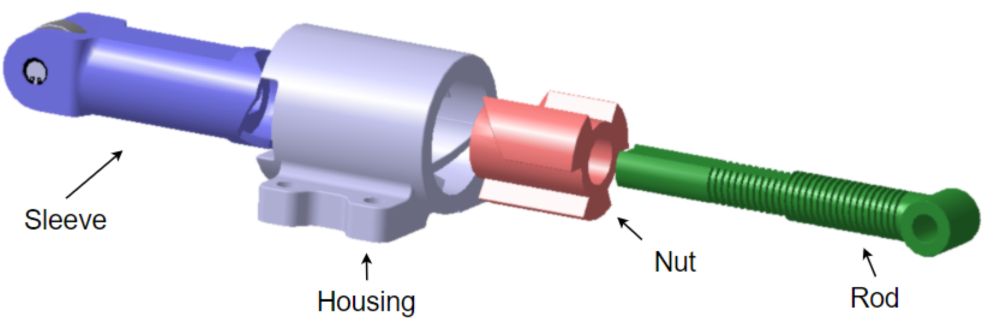


Figure 4.5: *Exploded view of the wear compensation mechanism.*

4.2.1 Sleeve

The sleeve is a cylindrical part with a fork structure at one end where a ball bearing is supported. The rest of the part is an open sleeve with an outer diameter matching the inner diameter of the internal splines of the housing, see Figure 4.6. The inner diameter of the open end matches the outer diameter of the nut (with the splines not included). The inner diameter of the sleeve matches the diameter of the rod and features two flat sides to constrain it from rotation inside the sleeve. Two similar flats on the outside of the sleeve constrain it from rotating inside the housing.

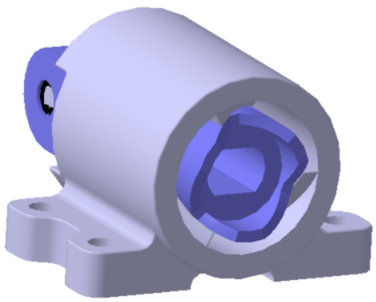


Figure 4.6: *A close up of the sleeve inside the housing.*

4.2.2 Housing

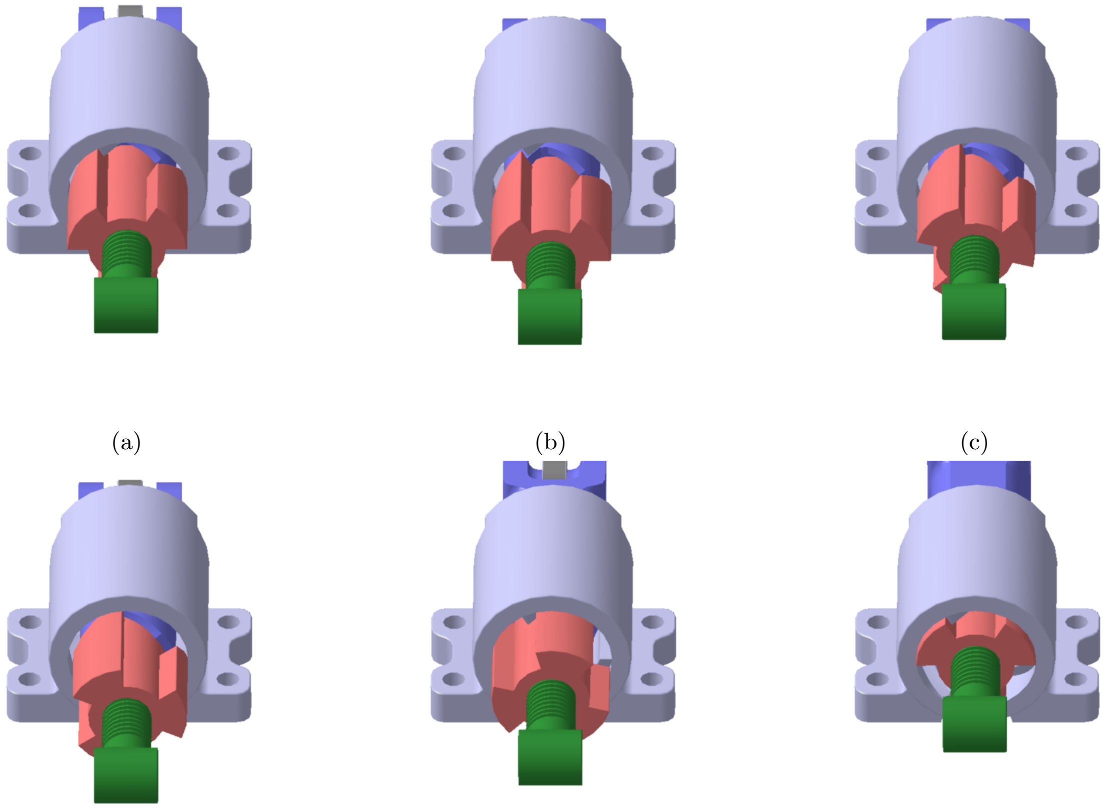
The main feature of the housing is to constrain five degrees of freedom for the sleeve and fixate the entire mechanism in space. For the latter, this particular design has a fixture with four bolt holes to enable mounting onto the transmission housing for instance. The front end of the housing seen in Figure 4.6, have three internal splines which constrain the nut from rotating within the housing but does not interfere with the sleeve.

4.2.3 Rod and Nut

The nut fits inside the end of the sleeve with the external splines constraining it from rotation inside the housing. The inclined ends of the splines interacts with the indentations in the sleeve and the clocking of the parts makes it so that the edges of the three splines are centered in these indentations, when within the housing. This forces the nut turn 15° when it travels past the splines of the housing. The inclined internal splines of the housing then forces the nut to turn another 105° on the return stroke. The rod as mentioned earlier is constrained from rotation. The axial movement of the rod relative to the nut, is constrained by a trapezoidal thread with a 2 mm thread pitch. The rod connects to the clutch actuation fork using a wrist pin, making a revolute joint between them. Due to the way that the fork is free to slide vertically on both its pivot point and on the throwout bearing, the rod does not need an extra revolute joint.

4.2.4 Operation

Figure 4.7 illustrates the compensation sequence in six steps. 4.7a is at 28 mm sleeve travel which is 5 mm before the nut is free to rotate. At 4.7b, the sleeve travel is 33 mm which allows the nut to turn. At this point the force F_3 acting on the rod and sleeve, forces the nut to turn 15° while the parts compress 3 mm and takes the position in 4.7c. On the return stroke of the sleeve, 4.7d - 4.7f, the inclined splines of the nut are forced to follow the inclined splines of the housing, making it turn another 105°. During this cycle, the rod will extend about 0.67 mm due to the thread. The compensation step corresponds to 0.04 mm at the pressure plate.



(a) (b) (c)

(d) (e) (f)

Figure 4.7: *Illustration of the wear compensation mechanism during a compensation.*

The relevant forces at different operation points are presented in Table 4.7. These are divided into three different stages of operation. Just before compensation refers to a stage when the wear is at a level that will lead to a compensation the next time a wear check cycle is performed. Just after compensation is a normal actuation when a compensation just have been performed. All values are peak values. One of the important results is the pressure plate force F_1 just before compensation, which does not fall short of the required 5620 N. One other important result is the throwout bearing force F_2 during the wear check cycle. As stated earlier, the yield limit of the fingers are reached at a stress that correspond to a throwout bearing force of about 1800 N. Also the actuator torque just after a compensation event is imported since this is the highest overall torque demand and thereby the reference point for the torque and power output requirement for the actuator.

Table 4.7: Acting forces at different states of wear.

Parameter description	Notation	Value	Unit
Just before compensation			
Force acting on the pressure plate.	F_1	5636	[N]
Force acting on the throwout bearing.	F_2	1409	[N]
Force acting on the wear compensation mechanism.	F_3	352	[N]
Actuator torque.	T_3	0.555	[Nm]
Just after compensation			
Force acting on the pressure plate.	F_1	5972	[N]
Force acting on the throwout bearing.	F_2	1493	[N]
Force acting on the wear compensation mechanism.	F_3	373	[N]
Actuator torque.	T_3	0.588	[Nm]
During compensation			
Force acting on the pressure plate.	F_1	6934	[N]
Force acting on the throwout bearing.	F_2	1733	[N]
Force acting on the wear compensation mechanism.	F_3	433	[N]
Actuator torque.	T_3	0.563	[Nm]

The following illustrations in figures 4.8 and 4.9, shows the wear compensation mechanism in connection with the rest of the actuation system. Here also a place holder for the clutch pack housing and an output/input shaft is included. It should be mentioned here that the parts in this actuation system is not optimized regarding strength nor weight and are only designed schematically.

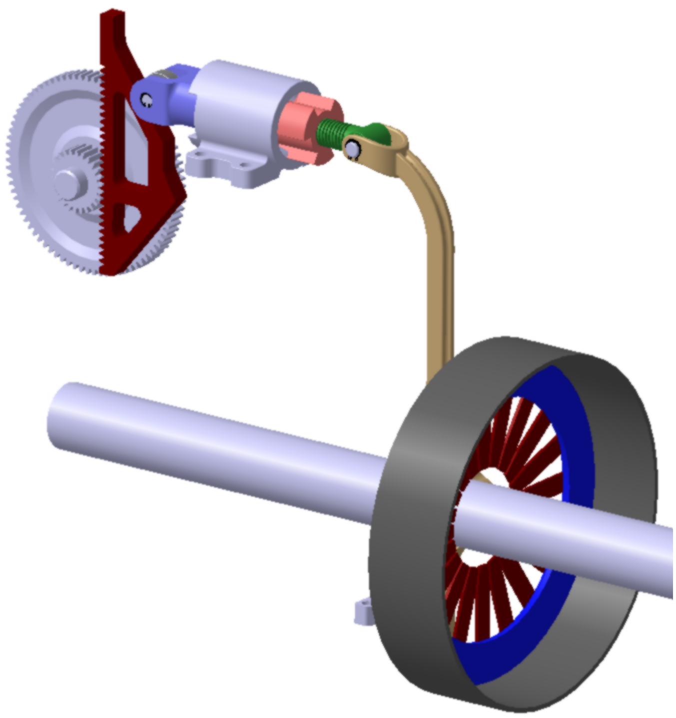


Figure 4.8: *Illustration of the wear compensation mechanism embedded into the actuation system.*

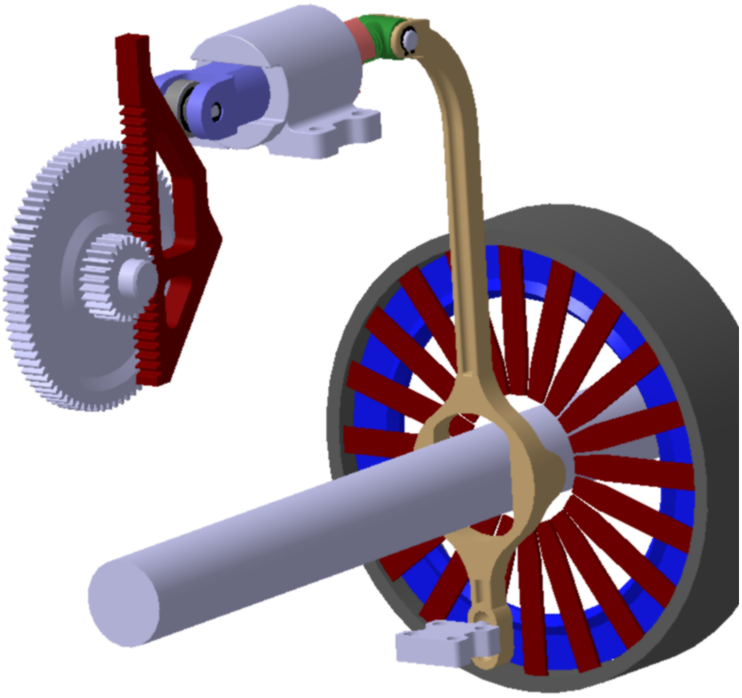


Figure 4.9: *Illustration of the wear compensation mechanism embedded into the actuation system.*

5 Discussion

In this chapter the discussions about the different parts of this project will be presented. Since some limitations and assumptions was used it is important to highlight these so no wrong conclusions are made. It is important to reflect about the conclusions found in the results and the methods used. This will result in better suggestions for the methodology and areas for future research.

5.1 Methodology

The methodology chapter introduced the flowchart showing the work flow used. The first part of this was the pre-study. To find relevant information and literature proved to be difficult. The theory and calculations behind the wear of the friction discs is based on a wide diversity of aspects. The wet clutch did not suffer from any significant wear, therefore only a dry clutch was used from this point forward. Using a reasonable wear rate, shift energy and number of actuations a reasonable total wear was calculated.

The pre-study also included searching for existing solutions. Several patents was found and after some sorting out these was presented in this chapter. The AVL concept was also added and described. The information and data about the patents proved to be of inconsistent quality. The descriptions of these concepts in the methodology chapter was written to briefly describe their functions respectively. These descriptions proved hard to simplify to increase the understanding for some readers. The sources for the complete patents however is available for interested readers.

The requirements and specifications was set on all three layers except for the one based on the results. These was filled in later when the final results was available.

A complete clutch system was designed using the requirements and specifications. The idea was to use a standardized system where the different concepts could be implemented. The two different locations where the wear compensation device is located, near the actuator and near the discs, was also decided upon. This whole system is simplified to make the reader easily understand the principles and to make a fair comparison easier to obtain.

The comparison between the concepts was presented and which of the concept that could function completely without or with some minor modifications.

The calculations and simulations were done and made it possible to fill in the requirements and specifications missing in the beginning. These steps, as declared in the flowchart, was an iterative process which was done to further improve the final results. Due to the time limitation the number of iterative processes was limited.

5.2 Results

The simulation results show promising potentials of faster shift time performance when implementing wear compensation. The closely investigated normally open dry friction clutch system, proved to be challenging just regarding the actuation system design. The simulation results presented for the non compensated system, show that the forces reached leads to stresses far beyond the yielding limit for the pressure plate levers. The clutch actuation fork can be designed to be more rigid and withstand the high forces and bending moments. Making the pressure plate levers more rigid however changes the entire characteristics of the actuation system regarding force verses displacement. Implementing the force limiter thereby improves the possibilities tremendously, when it comes to actuation system design. As presented in section 4.1.1, it also leads to great improvements regarding the shift time performance, especially the time to reach the required transferable torque. These times matches the performance of the displacement compensating system when looking at an unworn clutch due to the ability to use the same ramp pitch angle. For the worn clutch, the times are improved $40 - 45\%$ using displacement compensation. Regarding the full actuation sequence time difference is even more pronounced and this is important since the actuator in a system like this often actuates another clutch e.g. a dog clutch, in order to complete the shift. The effects of friction and inertia is not accounted for in these comparisons and that will affect the displacement compensation more due to the added mass and number of movable parts. The time limitations did not allow for further investigations of these effects.

The friction is also a concern regarding the wear check cycle. In order for this cycle to work properly it is important that the torque required for a certain displacement is consistent throughout the life cycle. This is something that can be challenging since increased friction is something that is hard to avoid. It will require an encapsulation if the mechanism to keep grease within it clean or a more sophisticated pressurized oiling system. An alternative solution for the wear check cycle is to use kiss point learning in order to only perform this cycle when compensation is needed. By doing it this way, the applied torque for the actuator can be higher (or the second ramp pitch can be lower), so that the margin for the compensation nut to pass the splines of the housing, is increased. It will however also require a slightly higher rigidity of the pressure plate levers to avoid yielding.

As been mentioned earlier, the parts in this actuation system are mostly designed schematically. The resulting size and weight of the parts should thereby not be of high significance when reviewing the wear compensation concept. These results are only meant to give a rough reflection.

5.3 Contributions and Future Research

The final concept could be developed further to make sure it would work as intended. The future ambition of the automotive industry is to make transports as energy efficient as possible, without the product getting too expensive. Since there is and will be shortages of some rare materials used in electric motors, the need for these materials can be reduced when implementing a multi-speed gearbox. If you implement this type of gearbox, the requirements and size for the electric motor is reduced, resulting in a less expensive and material intensive electric motor. This type of gearboxes and clutches is of conventional materials and the most sensitive part is the actuation system. To make sure this sensitive actuation system is working as intended throughout the complete life cycle, some sort of wear compensation system needs to be implemented when using a dry clutch. This system could be of the same type as the final concept, or as one of the present wear compensation solutions. Further development and testing must be done to affirm that it will check all the requirements.

Since this work had some limitations and the literature of these wear compensation systems are limited, the potential for further development should be substantial. One solution that is thought of but was not included in the scope is using a hydraulic system instead, compensating the difference in displacement when the discs are worn by changing the position of the piston inside a hydraulic cylinder that is controlled by an actuator. Another system that is not considered is a more sophisticated control system for the actuator, including kiss point learning that will ensure that this point is optimal at all times. The drawback of this type of system is if there is unexpected additional friction which will produce a false optimal kiss point. Another drawback is that it could require more space inside the gearbox to allow for further travel if the actuator is using a lever to operate the throwout bearing.

The actual wear and material used in the friction clutch is also something that turned out to be hard to estimate. The relevant literature available were few and finding data from the industry proved to be hard. In a real life system the data for this would have to be much more precise and preferably come from specialized experiments.

The placement and type of wear compensation device is greatly influenced by the type of clutch, normally open clutch or normally closed clutch. Some modifications are needed in the majority of the present solutions to function as intended in each system. The type and size of gearbox would also need to be verified to be able to choose the best solution in term of function and packaging.

This type of wear compensation systems is not limited to passenger vehicles. The theory and principles is the same for other types of vehicles, including trucks and buses. The benefits will be the same when using an actuator to operate one or several dry clutches in multiple-speed transmissions. The main future usage for these multiple-speed gearboxes is most likely for heavy vehicles and it will be in this industry that these wear compensations systems will be primarily used.

6 Conclusion

In the wake of the movement to reduce global emissions the switch to battery electric vehicles is rapidly increasing, this result in a customer demand for longer range, lower cost and better performance. The majority of the fleet of battery electric vehicle are using a single-speed gearbox. This is often sufficient for most type of daily driving. The electric motor is using some rare materials that is expensive. One way to cut the cost of the vehicle is to use a smaller motor, requiring less materials. One way to do this is to implement a two-speed gearbox. When using two different gear ratios the performance could remain the same as for the single-speed gearbox with a larger motor. The efficiency or a specific performance, e.g. top speed could also be increased.

The use of a dry clutch in this gearbox will introduce some axial wear of the clutch discs throughout the vehicles life cycle. To make this electronically controlled clutch operation consistent, some kind of wear compensation have to be implemented. The different solutions found from the pre-study to this problem was presented and evaluated to investigate which of the concepts that was most likely to function with none or minor modifications. A complete clutch system was designed using set requirements and specifications. It was found that a wet clutch had negligible wear but the dry clutch however will experience wear during its life time. From this and the calculated total wear of 2.15 mm over its life time of a dry clutch with specified dimensions some calculations and simulations could be done. The questions regarding how clutch actuation is affected by wear, from the aim and research questions, could be answered based on the findings from these calculations.

A decision was made to focus on one of the concepts, which was the most likely to develop, given the time limitation. This featured a modified rod ant nut concept solution in a normally open clutch system. The rod could be lengthen if wear had occurred by 0.67 mm for each of the steps when turning the nut. This resulted in a displacement of 0.042 mm at the friction discs. This concept was not designed to work perfect, just to show the basic functions of the wear compensation system.

Further development of these type of systems could improve the performance, cost and packaging of different electric drivelines. The multiple-speed gearbox is predicted to be a main part of electrifying the next generation of heavy vehicles. The principle behind this system that was developed, even though it was originally designed for a passenger vehicle with a two-speed gearbox, will work the same way in larger scale applications as well. The automotive industry will always aim to achieve the best possible driveline based on efficiency, cost and performance. Implementing multiple-speed gearboxes is one way to achieve this goal. If a dry clutch is a part of this type of gearboxes, some type of wear compensation is beneficial or even necessary, even if it is of some drastically different design from the one presented here.

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Appendices

A Simulation graphs - unworn to worn

A.1 No compensation

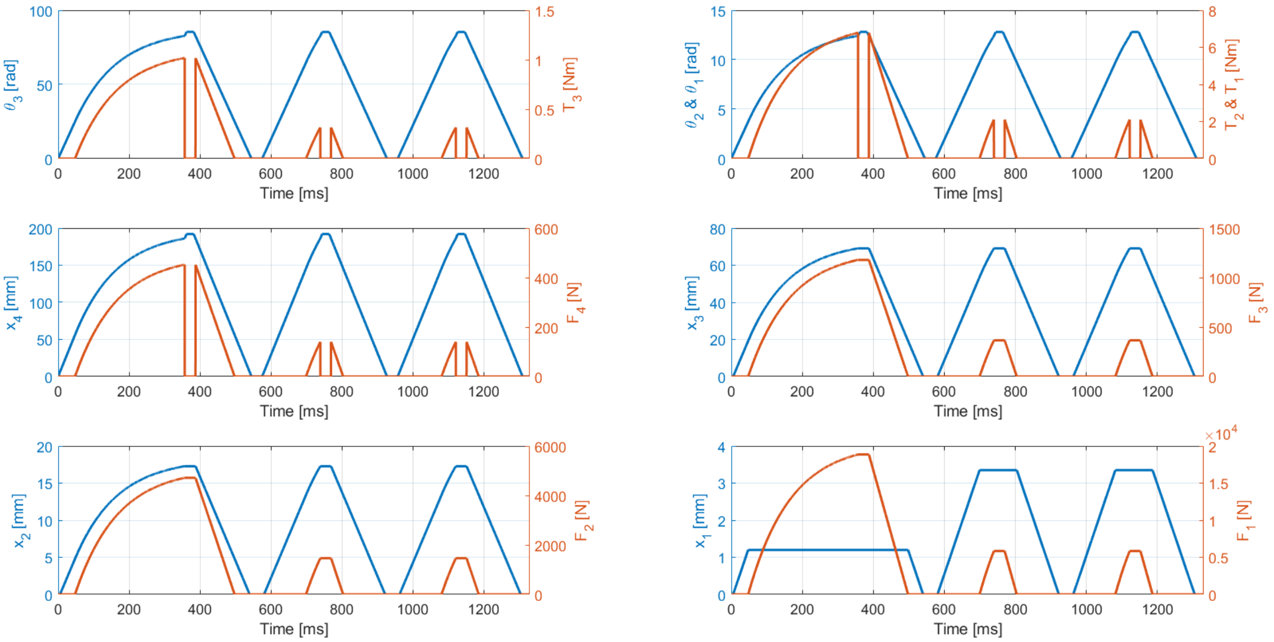

Figure A.1: *Force and displacement simulation of a normally open clutch system without wear compensation.*

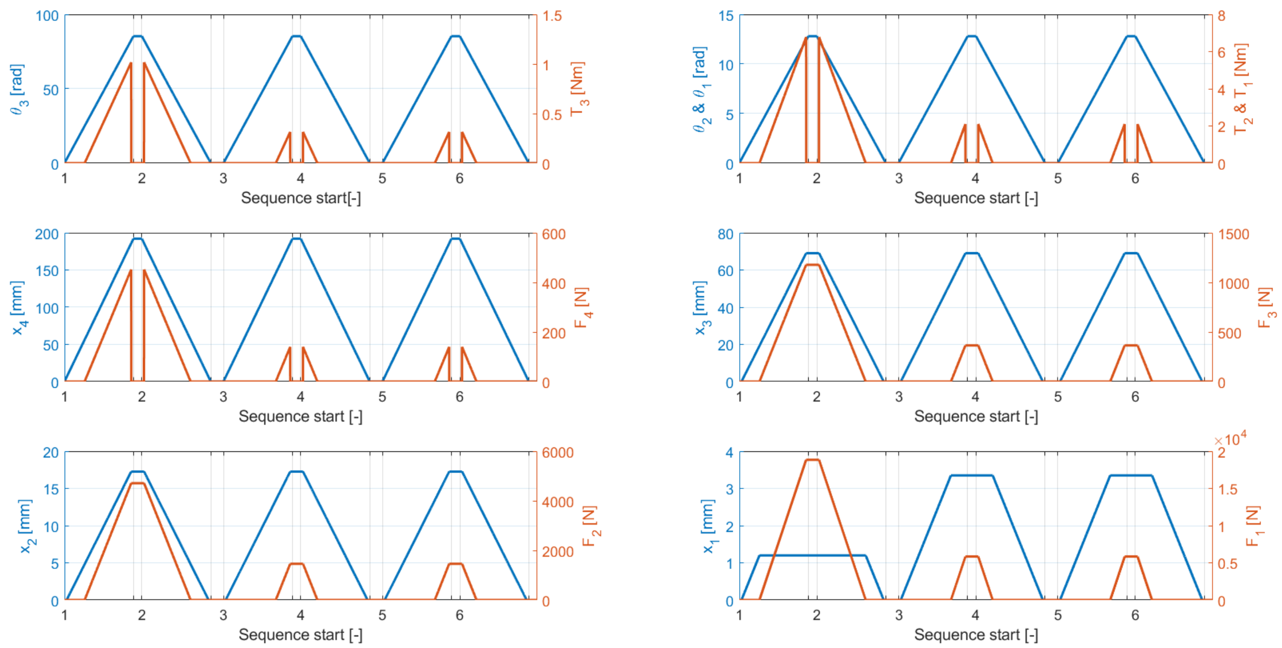

Figure A.2: *Force and displacement, time independent simulation of a normally open clutch system without wear compensation.*

A.2 Force compensation

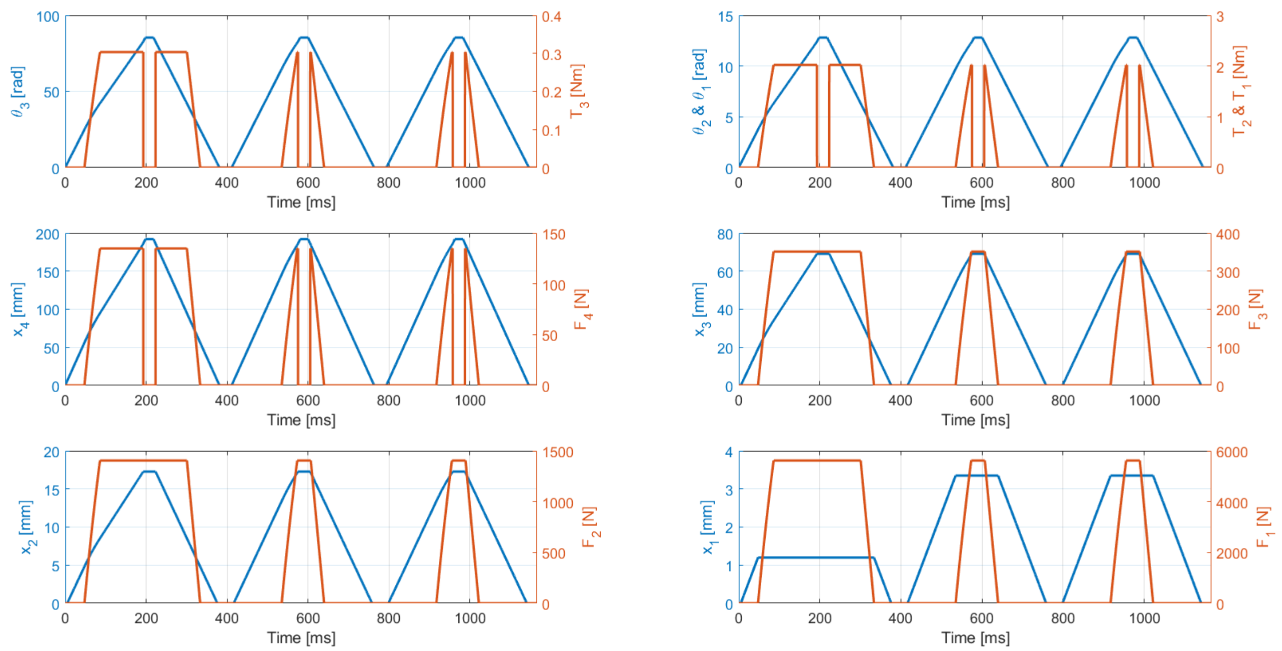

Figure A.3: *Force and displacement simulation of a normally open clutch system with the AVL force limiter.*

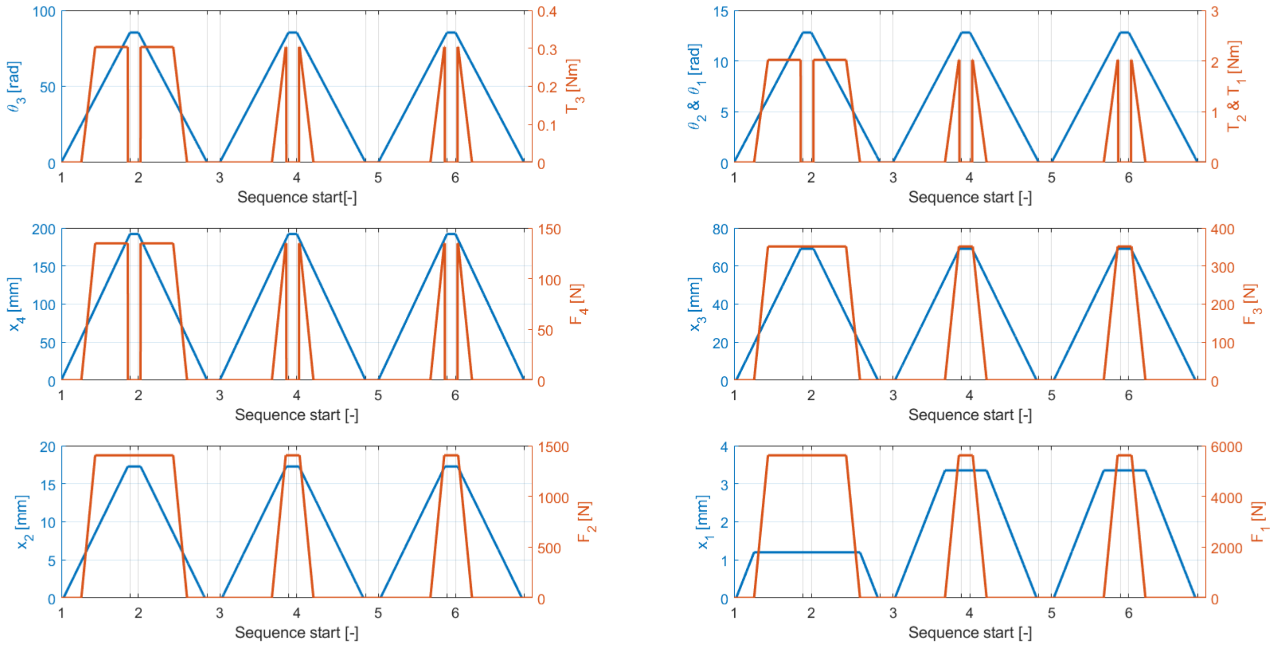

Figure A.4: *Force and displacement, time independent simulation of a normally open clutch system with the AVL force limiter.*

A.3 Displacement compensation near discs

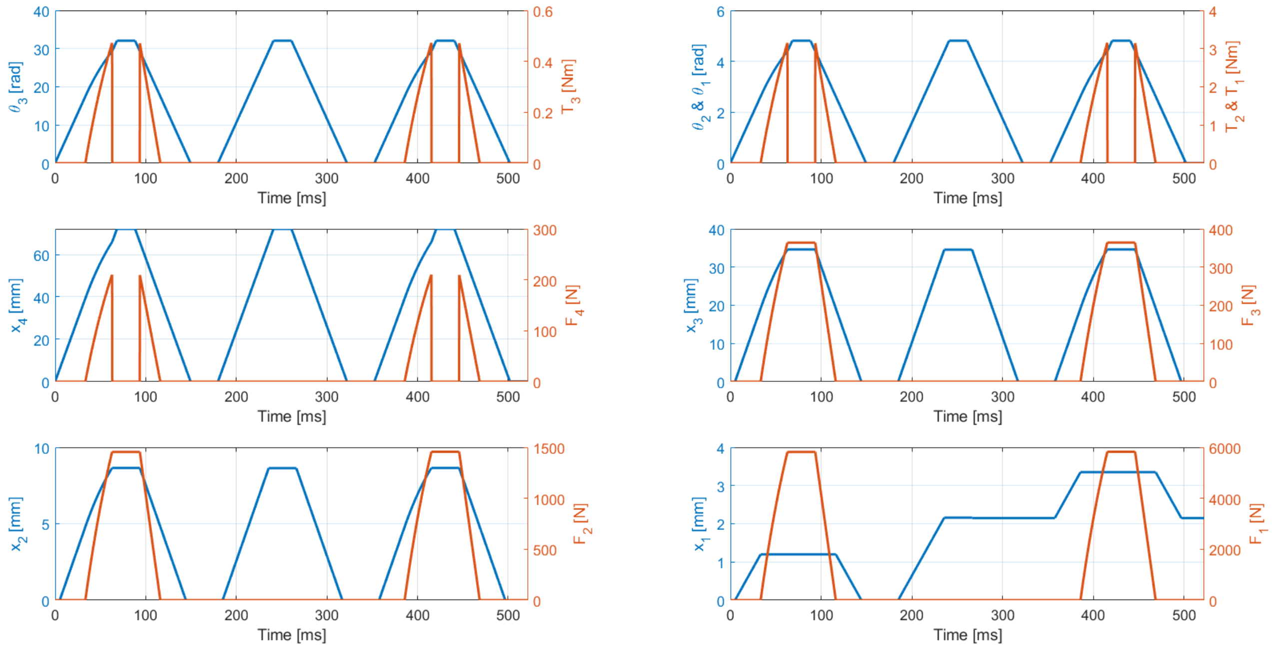

Figure A.5: *Force and displacement simulation of a normally open clutch system wit wear compensation near discs.*

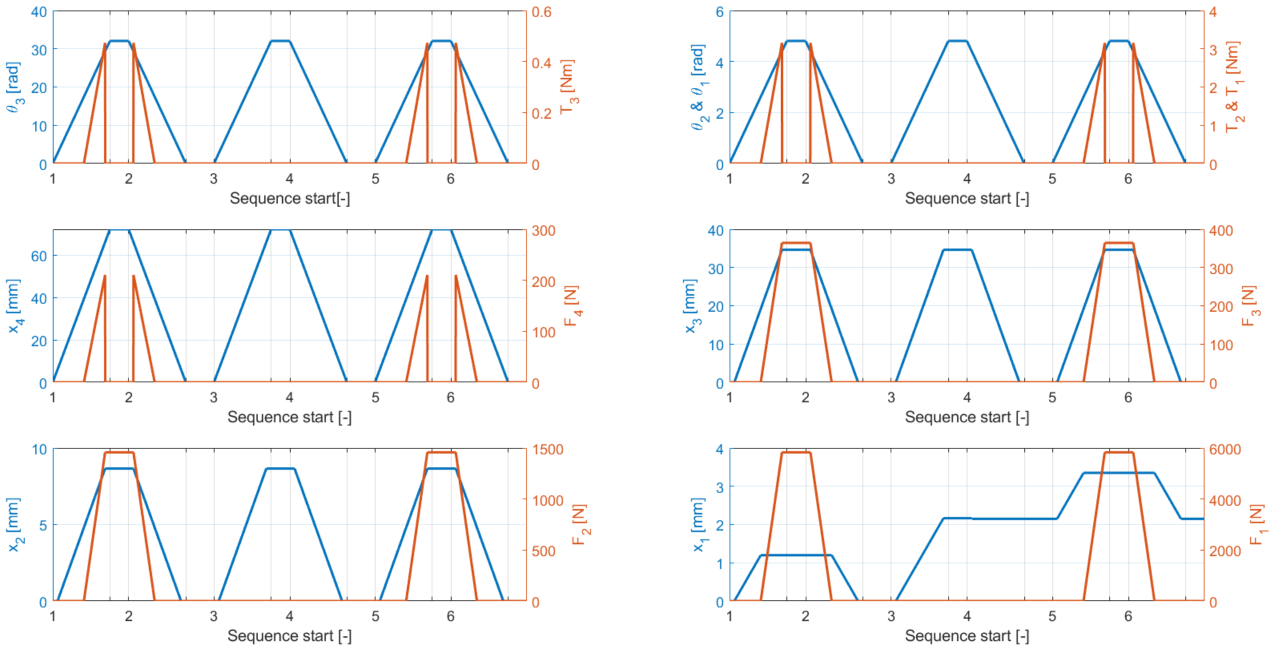

Figure A.6: *Force and displacement, time independent simulation of a normally open clutch system with wear compensation near discs.*

A.4 Displacement compensation near actuator

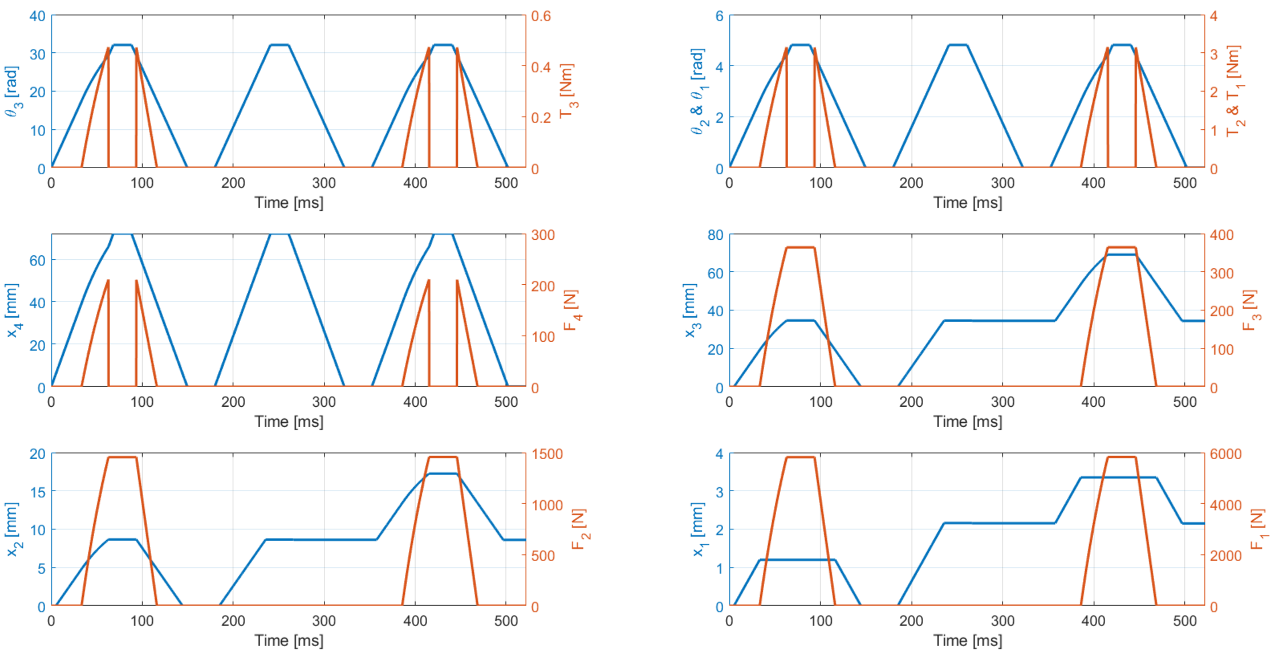

Figure A.7: *Force and displacement simulation of a normally open clutch system with wear compensation near actuator.*

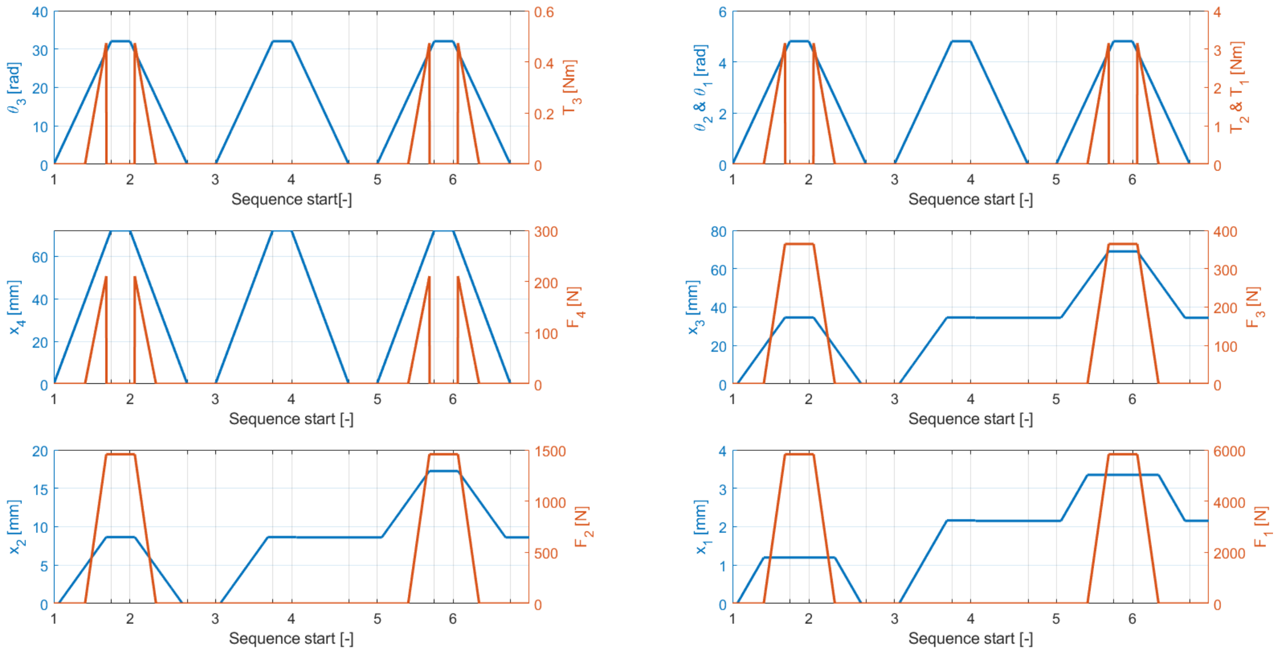

Figure A.8: *Force and displacement, time independent simulation of a normally open clutch system with wear compensation near actuator.*

B Simulation graphs illustrating actuation time performances

B.1 No compensation

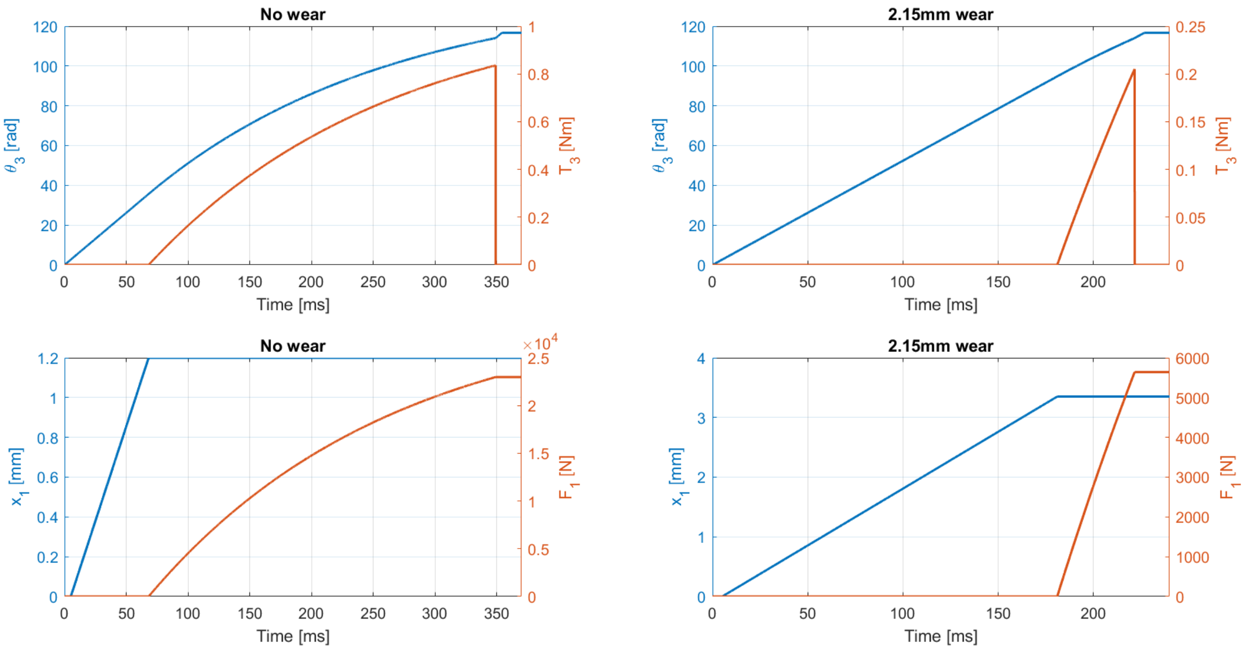

Figure B.1: *Clutch actuation performance with a ramp pitch angle of 14.5° and no wear compensation*

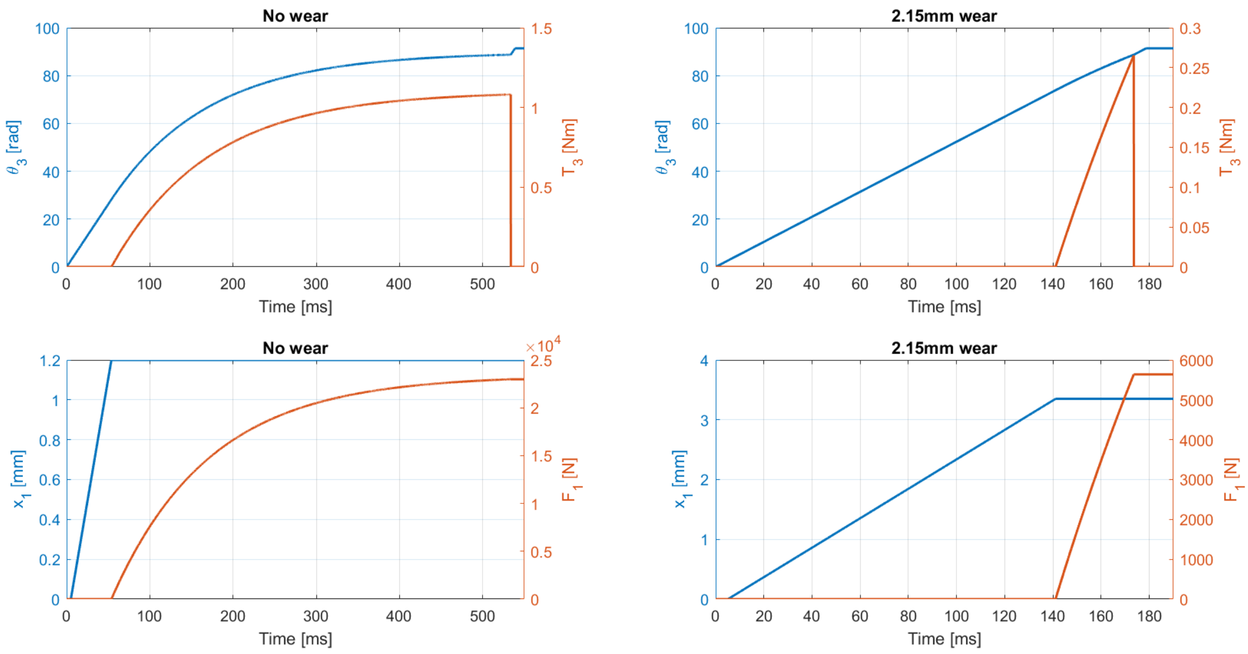

Figure B.2: *Clutch actuation performance with a ramp pitch angle of 18.5° and no wear compensation.*

B.2 Force compensation

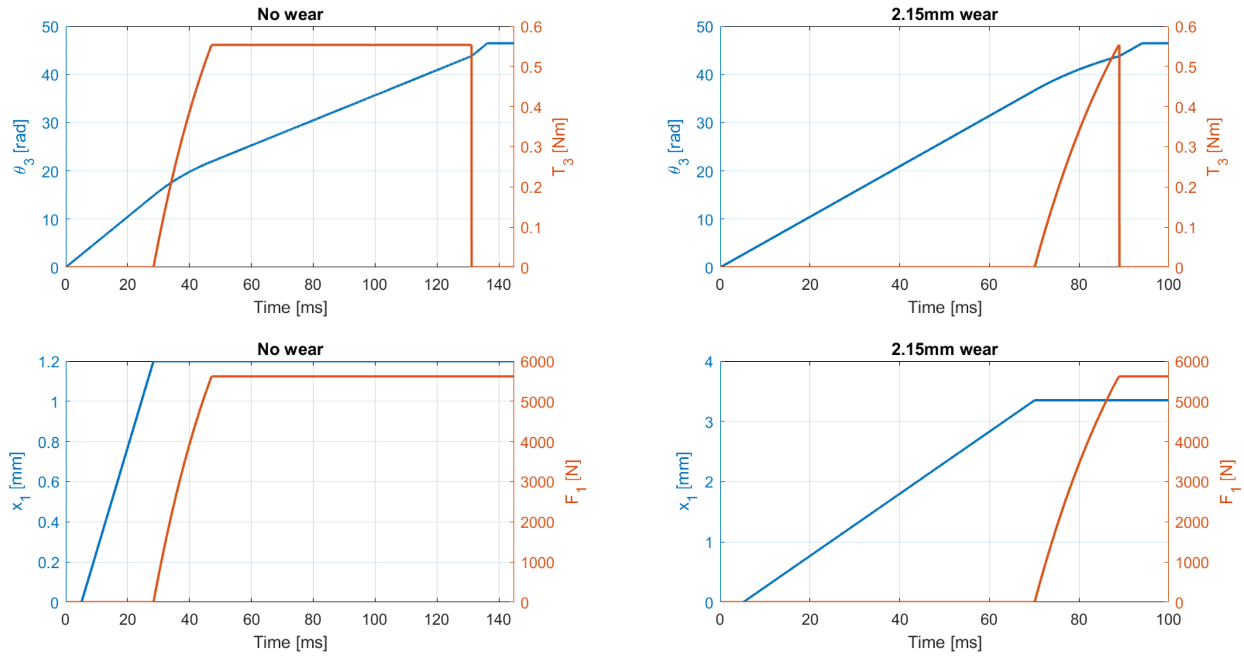

Figure B.3: *Clutch actuation performance with a ramp pitch angle of 35° and force compensation.*

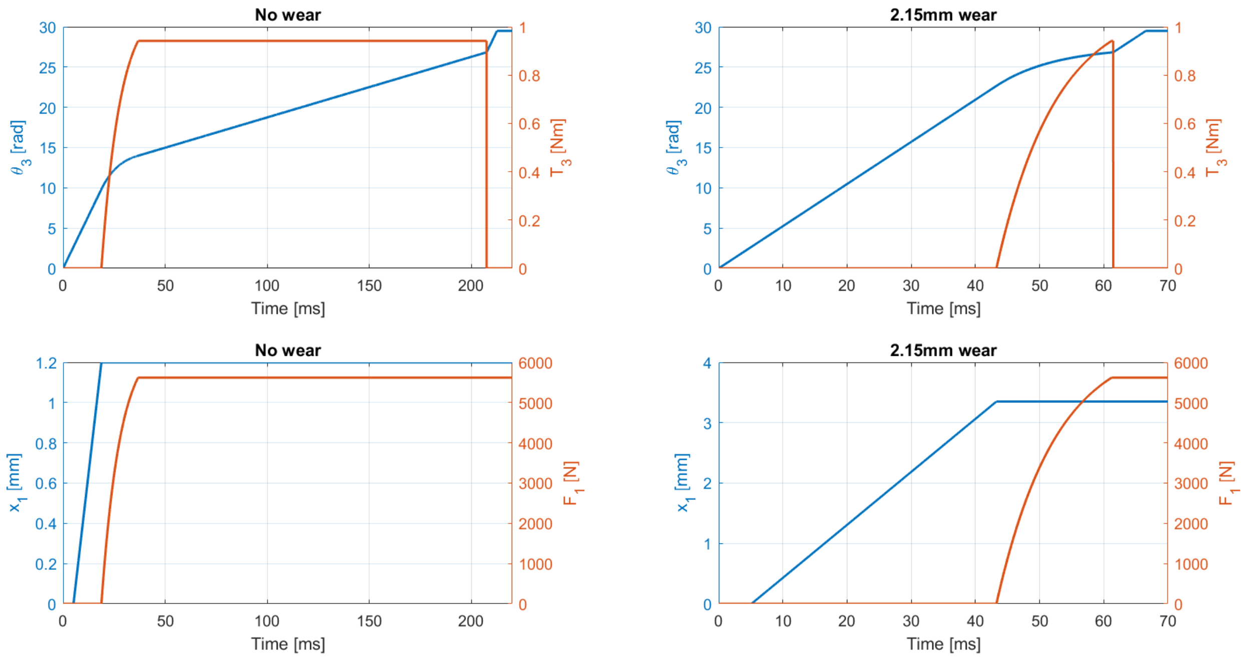

Figure B.4: *Clutch actuation performance with a ramp pitch angle of 50° and force compensation.*

B.3 Displacement compensation

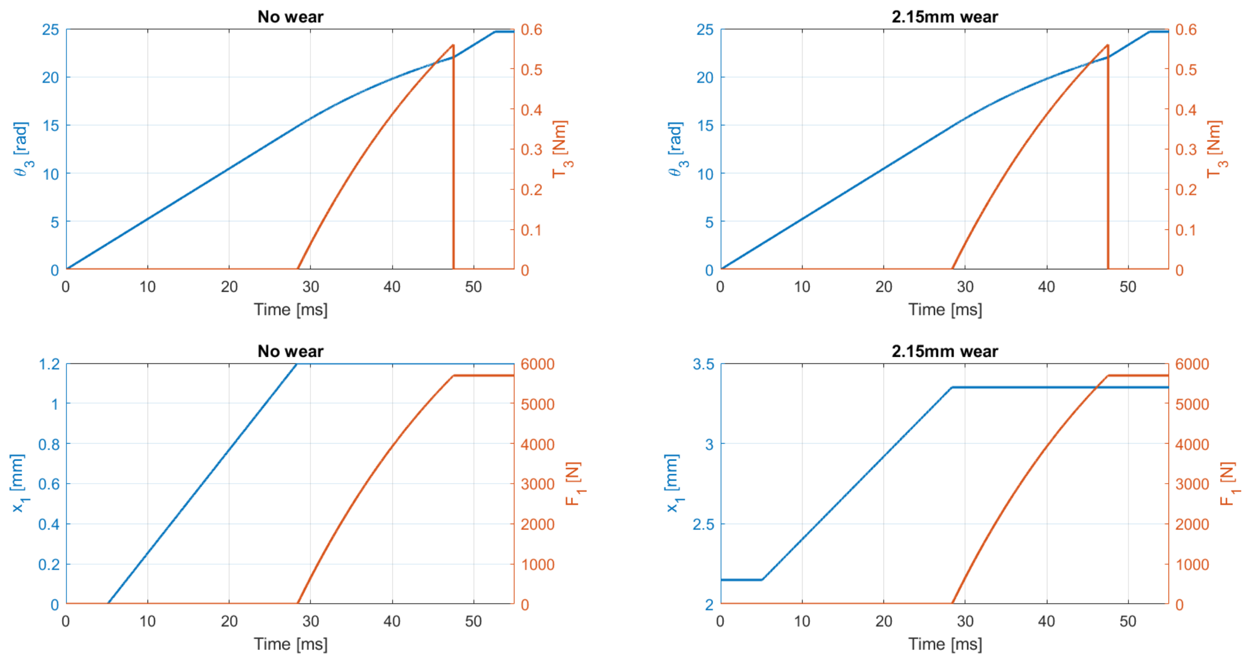

Figure B.5: *Clutch actuation performance with a ramp pitch angle of 35° and displacement compensation.*

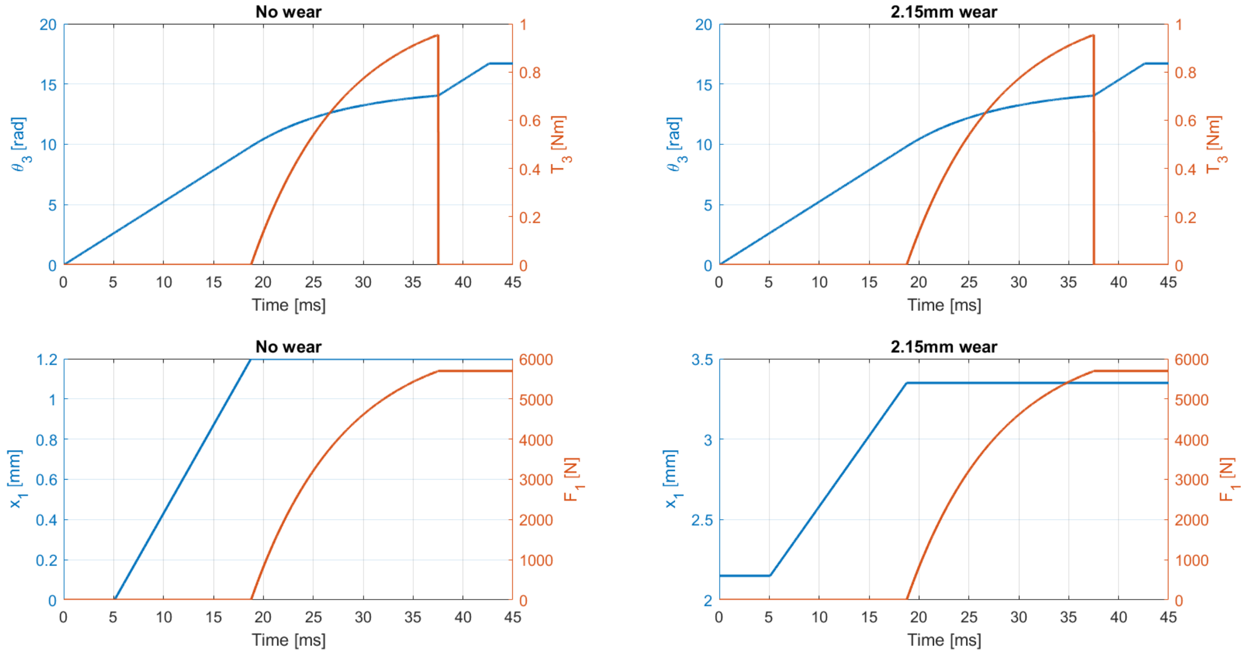

Figure B.6: *Clutch actuation performance with a ramp pitch angle of 50° and displacement compensation.*

C Compiled result tables

C.1 Actuation system specifications

Table C.1: Pressure plate levers specifications.

Parameter description	Notation	Value	Unit
Vertical distance from outer support to pressure plate contact point.	a_1	15	[mm]
Vertical distance from pressure plate contact point to throwout bearing.	b_1	45	[mm]
Lever beam width.		10	[mm]
Lever beam thickness.		1.8	[mm]
Number of levers.	n_{lever}	20	[pcs]
Lever ratio	i_{lever}	4	[-]

Table C.2: Clutch fork specifications.

Parameter description	Notation	Value	Unit
Vertical distance from lower support to throwout bearing contact point.	a_2	60	[mm]
Vertical distance from throwout bearing contact point to upper revolute joint.	b_2	180	[mm]
Horizontal meassurment from throwout bearing contact point to upper revolute joint.		60	[mm]
Fork weight.		0.27	[kg]
Lever ratio.	i_{fork}	4	[-]

Table C.3: Ramp specifications.

Parameter description	Notation	Value	Unit
Length of first and second plateau.		12	[mm]
Height of second plateau, relative the first.		30.5	[mm]
First inclination angle.	α_1	35	[deg]
Second inclination angle.	α_2	30	[deg]
Ramp weight.		0.18	[kg]

Table C.4: Gear specifications.

Parameter description	Notation	Value	Unit
Actuator drive gear radius.	r_3	7.5	[mm]
Layshaft input gear radius.	r_2	50	[mm]
Layshaft output gear radius.	r_1	15	[mm]
Total layshaft weight.		0.57	[kg]
Total gear ratio (actuator to ramp).	i_{gears}	0.44	[rad/mm]

Table C.5: Requirement specification.

	Req. ID	Requirement Statement	Value	Unit	Spec. ID	Specification statement	Value	Unit
L0 (clutch system)	**R0**				**S0**			
	R0.1	Transferable torque	> 700	[Nm]	S0.1	Outer diameter	170	[mm]
	R0.2	Shift time	< 600	[ms]	S0.2	Total thickness (disc pack)	45	[mm]
	R0.3	Total number of shifts	> 175000	[-]	S0.3	Number of friction discs	4	[pcs]
	R0.4	Wear management	> 2.15	[mm]	S0.4	Coefficient of friction	0.30	[-]
L1 (actuation system)	**R1**				**S1**			
	R1.1	Peak actuation force required upon pressure plate	5620	[N]	S1.1	Actuation system radial measurement**	245 (160)	[mm]
	R1.2	Pressure plate displacement*	> 3.35	[mm]	S1.2	Actuation system axial measurement**	305 (301)	[mm]
	R1.3	Number of shift cycles	> 175000	[-]	S1.3	Actuation system weight***	2.03	[kg]
	R1.4	Shift time	< 600	[ms]	S1.4	Total ratio between actuator and pressure plate	10.16	[rad/mm] [kN/Nm]
	R1.5	Additional radial space	< 165	[mm]				
	R1.6	Additional axial space	< 330	[mm]				
	R1.7	Added weight	< 3.50	[kg]				
L2 (actuator)	**R2**				**S2**			
	R2.1	Actuator torque output	> 0.88	[Nm]	S2.1	Actuator stall torque	1.10	[Nm]
	R2.2	Actuator power output	> 36.93	[W]	S2.2	Actuator no load speed	5000	[rpm]
	R2.3	Actuator weight	< 1.47	[kg]	S2.3	Actuator peak power output	144	[W]
	R2.4	Number of shift cycles	> 175000	[-]	S2.4	Actuator weight	1.00	[kg]

*R1.2 Itself only makes sure that all gaps between clutch discs can be closed at all states of wear.

**Added space within parentheses.

***Actuator weight excluded.

C.2 Actuation system performance

Table C.6: Time performance comparison.

	Pitch angle	Unworn clutch	Worn clutch
Total actuation sequence time			
No compensation	14.5°	354 ms	227 ms
	18.5°	539 ms	179 ms
Force compensation	35°	136 ms	94 ms
	50°	213 ms	67 ms
Displacement compensation	35°	53 ms	53 ms
	50°	43 ms	43 ms
Time to reach required transferable torque			
No compensation	14.5°	109 ms	221 ms
	18.5°	86 ms	173 ms
Force compensation	35°	47 ms	89 ms
	50°	37 ms	61 ms
Displacement compensation	35°	47 ms	47 ms
	50°	37 ms	37 ms

Table C.7: Acting forces at different states of wear.

Parameter description	Notation	Value	Unit
Just before compensation			
Force acting on the pressure plate.	F_1	5636	[N]
Force acting on the throwout bearing.	F_2	1409	[N]
Force acting on the wear compensation mechanism.	F_3	352	[N]
Actuator torque.	T_3	0.555	[Nm]
Just after compensation			
Force acting on the pressure plate.	F_1	5972	[N]
Force acting on the throwout bearing.	F_2	1493	[N]
Force acting on the wear compensation mechanism.	F_3	373	[N]
Actuator torque.	T_3	0.588	[Nm]
During compensation			
Force acting on the pressure plate.	F_1	6934	[N]
Force acting on the throwout bearing.	F_2	1733	[N]
Force acting on the wear compensation mechanism.	F_3	433	[N]
Actuator torque.	T_3	0.563	[Nm]

D Rendered concept illustration

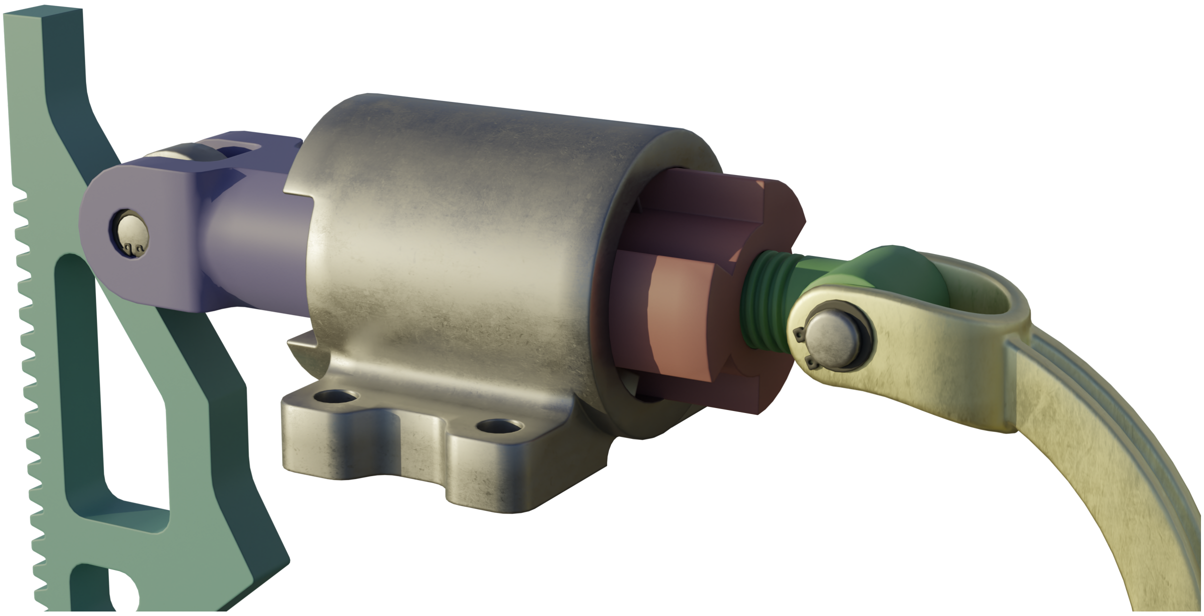

Figure D.1: *Picture of the wear compensation device rendered using Blender.*

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