Automatic steering wheel system for tractors

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Abstract

This project is part of a LeadEng collaboration with two other master programs, *Mechatronic Control Engineering* (MCE) and *Power Electronics And Drives* (PDE) and a specific motor therefore has to be incorporated.

The main objective of this project is designing and testing of the mechanical part of an automatic steering wheel system, where the design is the dominant and most comprehensive part of the project. The key motivation for doing this is utilizing obtained engineering expertise to create a new product, as well as attaining a greater experience in design and creation of a product.

In order to achieve the central goal of this project, firstly a problem analysis is conducted, where the problem that needs to be solved by the design is examined from different points of view. Secondly, based on that, the main focus is defined and delimitations are determined. Subsequently, concepts are generated and evaluated, as an effort to then choose a final solution, which will be designed. Afterwards, the design process is described.

Once this is completed, the design process is divided into sections, based on the function of each of them. The first section to be designed is regarding the gears, the gear ratios and the necessary forces transmitted by them for the subsequent designs. The succeeding section is the clutch mechanism, whose design is crucial for the section that follows, which is the shaft. The final designed part will be the fixture, which mounts and connects the components of the product together.

To conclude, the design is then validated in terms of structural integrity, using ANSYS simulations. Afterwards, economical and production aspects of the product are analyzed. Thereupon, the design, along with the developed prototype are discussed, where their flaws and strengths are examined. In conclusion, the design has been proven to posses most of the desired functionality, with the substantial flaws being the shifter mechanism and the provided motor.

Keywords: Tractor, Autopilot, Design, Steering Wheel, Farming, Electric Motors, Optimization

1. General Introduction

With the growing world population, the demand for food is increasing. In order to meet the future demand for food, it is estimated that farmers must increase their production by more than 70% in the next 50 years.[1] Moreover, agricultural farming is an exhausting task, and the long working hours, as well as monotonous work leave farmers fatigued. As the fatigue level of the farmers increases, the risk of injuries grows. [2] As a mean to increase the yield efficiency and reduce the strenuous work, more and more farmers are looking into automated solutions, where automatic and intelligent systems are used to ease the manual labour. [3] Technologies like self-steering systems that enable the tractor to be autonomous at certain levels have been developed to assist farmers when driving tractors, making the long working hours become less fatiguing. Furthermore, these technologies allow the farmers to operate efficiently in conditions with poor visibility and optimize the yield efficiency as the amount of overlap between crop lines can be minimized. If needed, such self-steering systems could be implemented with other automatic systems to make the tractor fully autonomous, which would allow farmers to further optimize the production. Based on this, the main focus of this project is stated as the following:

To design, optimize and test the mechanical part of an electrical self-steering system which can be implemented in different tractors.

2. Problem Analysis

In order to realize the mechanical design of an electrical self-steering system, it is necessary to understand the requirements of such a system.

Usage of a Self-Steering System

The main usage of the self-steering system is to allow the farmer to follow a predefined path as illustrated on Figure 1. The following assumptions a made about the usage of the system; it will not be in use when driving to and from a given field and for every day work, as this does not have predefined paths to follow.

Fig. 1 Illustration of possible paths needed drive to cover a complete field. The path is superimposed on a random field close to Aalborg city [4]

Market Analysis

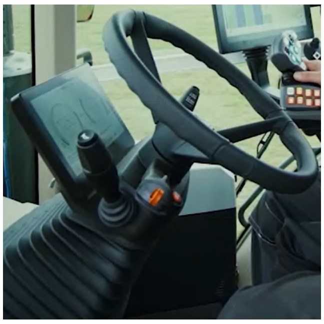
A study of steering wheels and steering wheel columns of different tractors is carried out, as these are fundamental in order to ensure the compatibility of the solution. Furthermore, different existing solutions are studied to get an understanding of the disadvantages of what is currently on the market, such that an improved solution can be designed.

Tractor Steering Wheels and Columns

The steering wheels and columns presented here are from modern agricultural tractors, without inbuilt automatic steering. From the examination it is clear that most common tractor steering wheels use a three spoke design, and that there is no direct access to the steering shaft. The spokes are all 120° apart and vary in thickness. A few of the examined steering wheels can be seen in Figure 2.

(a) Valtra

(b) New Holland T8 [5]



(c) Fendt Vario [6]

(d) John Deere 7R [7]

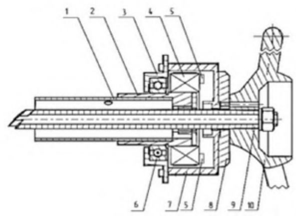
Fig. 2 Different steering wheels and columns

The following physical requirements to the layout of the system, are formulated:

- *Be able to fit steering wheels with three spokes separated by 120°*
- *Be able to fit a steering wheel with no access to the steering shaft*
- *Be able to fit steering wheels with varying spoke angles*
- *Be able to be mounted at different positions around the steering wheel to avoid lever and dashboard interference*

Existing Solutions

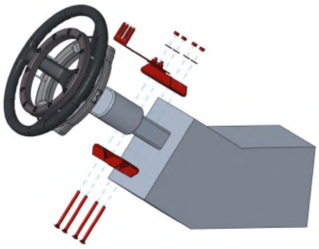
Different solutions exist and only the most relevant selection of these are presented. The solutions presented are chosen specifically due to the fact that their operating and mounting methods are different, providing a broad view of the possibilities when designing the system.



(a) Integrated rotor system [8]

(b) Belt driven solution [9]

(c) Friction drive system [10]



(d) Ag Leader [11]

Fig. 3 Existing solutions

Based on the information available to potential buyers and not from testing the products, seen in Figure 3, the disadvantages were found to be:

- *High price*
- *Not compatible with odd shaped steering wheel columns*
- *Incapable of or difficult to disengage*

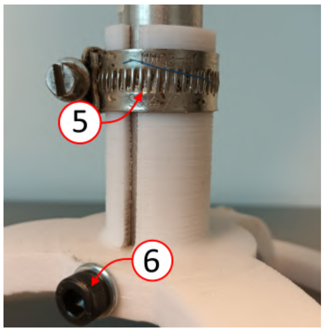
Experimental Analysis

An experiment was conducted to analyse the torque required to turn the steering wheel when driving in the field, such that the system can be designed to supply the required torque. The angle was also measured, to determine if the required torque would be different at different steering angles. To isolate the strains originating from the steering wheel torque, four strain gauges were used. These were mounted on opposite sides of the shaft and used in a full-bridge setup.

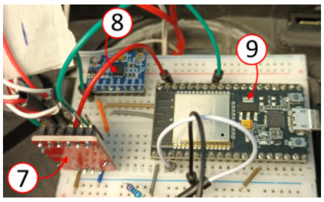
The test setup can be seen on Figure 4. It is composed of a secondary steering wheel (1), an aluminum shaft (2) and a 3D printed adapter (3) for the main steering wheel. The 3D printed adapter is bolted (4) to the tractor steering wheel. The strain gauges, two of which are noted with (10) & (11), are glued on to the aluminum shaft. In order to transfer torque from the secondary steering wheel to the main steering wheel, a bolt (6) is fitted through the adapter and the aluminum shaft. The 3D print and the aluminum shaft are further secured with a clamping strap (5) to restrict bending. The four strain gauges are wired in a full bridge and connected to the HX711 amplifier (7) which sends the amplified voltage signal to the ESP32 board (9). To measure the angle of the steering wheel, the MPU-6050 (8) is connected to the ESP32 micro-controller board [12].

All the tests were executed with the same tractor on the same field under cool and moist weather conditions.

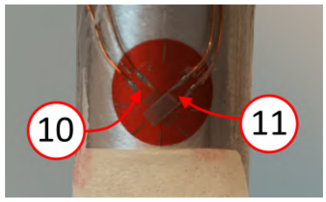
(a) Torque sensor mounted



(b) Mounting of the shaft



(c) Electronics of the sensor



(d) Two of the strain gauges

Fig. 4 Pictures of the test setup. (1) Secondary steering wheel, (2) aluminum shaft, (3) 3D printed adapter for the tractor steering wheel, (4) bolts to connect the two parts of the adapter, (5) clamping strap, (6) bolt to fix the shaft to the adapter, (7) HX711 amplifier, (8) MPU-6050, (9) ESP32 micro-controller board, (10) & (11) Strain gauges

The first test was turning the steering wheel through it's full range of motion, both slow and fast. The second test was driving in a straight line at respectively 3 $\frac{\mathrm{km}}{\mathrm{h}}$, 11 $\frac{\mathrm{km}}{\mathrm{h}}$ and 30 $\frac{\mathrm{km}}{\mathrm{h}}$. The third test was driving around the field in a random pattern, and at random velocities.

Final Results

As seen of Figure 5, the absolute max measured torque was found in test two to $T_{max,measure} = 2.39\,\mathrm{Nm}$. From the figure it can also be seen that there is no dependency between the torque and the angle, and that there is no difference when stationary and driving.

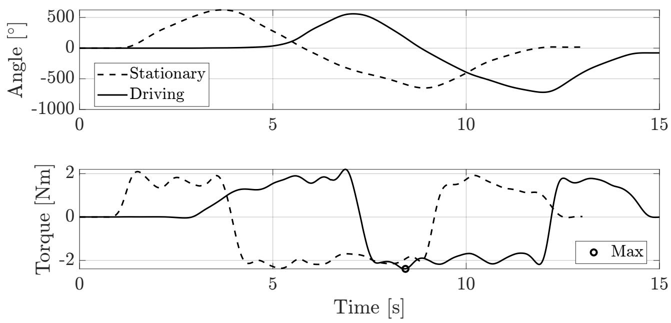


Fig. 5 Measurements from the first test

The influence of inexact mounting of the strain gauges was investigated by considering the theoretical strain measured in each strain gauge subject to a general load condition. The influence of the inexact mounting was estimated to be $\approx 7\%$ and therefore the maximum torque is corrected to $T_{max} = 2.56\,\mathrm{Nm}$. This value covers both the load torque and the torque needed to overcome the inertia of the system. The maximum angular velocity of the steering wheel was $\omega_{w,max} = 84.5\,\mathrm{rpm}$. This was obtained when turning the steering wheel as fast as possible, providing a good estimate of the upper rotational speed limit the steering wheel would have to follow.

These measurements were only done for a single tractor and may not be representative for other brands and models. To account for this a safety factor of 3 will be used as this is standard for models that approximately represent the system [13]. Thereby the steering wheel torque used for later dimensioning is: $T_{max} = 7.67\,\mathrm{Nm}$. All the test results can be seen in table I.

Tests	**No. 1**				**No. 2**						**No. 3**
	Stationary		Driving		3 kph		11 kph		30 kph		
	Fast	Slow	Fast	Slow	No-Hold	Hold	No-Hold	Hold	No-Hold	Hold	
Max	2.37	2.02	2.39	2.07	0.04	0.29	0.03	0.95	0.56	1.52	2.21
Mean	1.43	1.27	1.21	1.45	0.03	0.12	0.01	0.47	0.51	0.78	0.65

Tab. I Test results from the tests measured in Nm

Customer Analysis

In order to obtain useful information about the customers needs and wishes, a survey was conducted. Based on the answers, a House of Quality was constructed, to transform the needs and wishes into the design specifications shown in Table II.

3. Main Focus and Delimitations

Based on the market analysis, experimental tests and the customer analysis, the main focus defined in the introduction, is redefined as:

To design, optimize and test the mechanical part of an electrical self-steering system which can fulfill the physical requirements and the mechanical design specifications.

Delimitations

Due to the limited scope of the project it is not possible to design a fully implementable system. Therefore, the intended design will be made as a prototype where the proper manufacturing techniques and materials are not used. This will serve as a concept validation that will show whether the intended functionality is obtainable.

As the focus of the project is to design a mechanical solution incorporating an electrical motor, many of the design specifications are not relevant as they relate to the position control and the power electronics of the system. The relevant design specification can be seen in Table II.

Specification	**DoI**	**Target**	**Units**	**Weight [%]**
Installation time	$\downarrow$	1.5	Hours	2.7
Tools needed for installation	$\downarrow$	3	#	2.7
Amount of components needed to be replaced	$\downarrow$	0	#	5.4
Amount of required specialized tools	$\downarrow$	0	#	2.4
Installation operations	$\downarrow$	20	#	2.7
Manual deactivation		yes		7.8
Automatic deactivation		yes		6.1
Overall component cost	$\downarrow$	5.000	EUR	5.9
Dimensions of the mechanical system	$\downarrow$	20 x 20 x 20	cm x cm x cm	7.8
Minimum number of mounting positions	$\uparrow$	1	#	3.6
Torque at the steering wheel	$\uparrow$	2.39	Nm	3.3
Steering wheel angular velocity	$\uparrow$	84.5	rpm	2.2
Steering wheel movement smoothness	$\uparrow$	$4.4 \cdot 10^{-4}$	$\frac{rad}{s^3}^{-1}$	3.8
Weight of the mechanical system	$\downarrow$	1.5	kg	3.3

Tab. II Design specifications

4. Solution Generation and Evaluation

To find and chose the overall solution, different sub-concepts are generated and evaluated to find the most appropriate concepts. Based on this, a range of possible solutions is produced and evaluated to find the optimal one.

Concept Generation & Evaluation

The generation of concepts is based on the functions which must be fulfilled by the solution. The functions are shown in the concept tree in Figure 6.

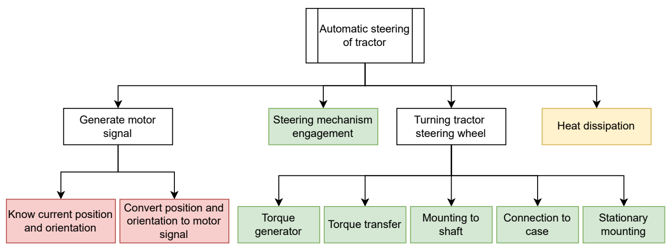


Fig. 6 Function tree

All the sub-functions are split in categories, where the green sub-functions indicate solutions which demand mechanical solutions, whilst the red functions are sensor and controller based. The yellow is outside the scope of the project. As the focus of the paper is to develop the mechanical solution only, the green sub-functions will be used, whilst the red sub-functions are handled by the other groups in the LeadEng collaboration. For each sub-function multiple possible solutions are generated by brainstorming. These include both concepts that are already seen in existing solutions along with innovative concepts.

The generated concepts were evaluated based on feasibility and using the design specifications, to find the most suitable concepts.

Solution Generation & Evaluation
Based on the optimal concepts for each function an optimal solution can be defined, which is a combination of all the concepts. This however does not account for the feasibility of the combined concepts wherefore additional solutions are generated. This is done by choosing the best solution from each category and choosing the most sensible solutions from the remaining categories. The final solutions are evaluated based on their feasibility, reliability and ability to fulfil the system specifications. This leads to the optimal solutions which uses a combination of gears and a clutch to enable a disengageable torque transfer. The solution is mounted to the steering wheel by clamping around it, while the system is kept stationary by combining sheet metal parts with a strap.

5. Design

The designed solution can be seen in Figure 7 where the yellow part (1) is the stationary mount, the green part (2) is the connection to case and the red part (3) is the casing. Together these three parts make up the fixture. The purpose of the stationary mount is to fix the solution to the steering column, such that it is stationary relative to the column. The purpose of the connection to case, is to connect the casing to the stationary mount, such that the casing is also kept stationary. The purpose of the casing is to contain the motor and other rotating parts.

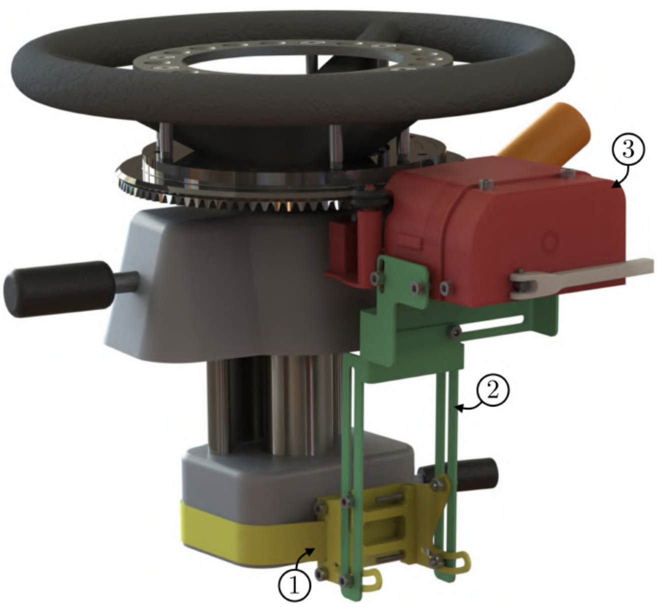


Fig. 7 Full assembly of the designed solution

To ensure proper meshing of the bevel gears, these are held together using v-shaped wheels. These are seen in Figure 15.

Gear Design
Gear design consists of determining the gear types, the gear tooth parameters and the sizes. Also, as part of it, the strength of the gears is determined. Comparing the motor parameters with the values obtained in the experimental analysis results in a required gearing of:

$$n_{full} = 33.26 \tag{1}$$

The most compact design is obtained using a combinations of helical and bevel gears seen in Figure 8.

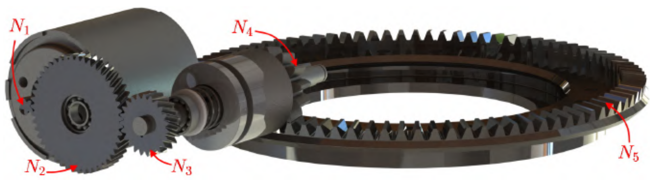


Fig. 8 The gearing

As the motor is capable of producing large torques, the gears must be manufactured using through hardened steel. The strength of the gears are validated by analysing the bending and surface stresses in fatigue. Assuming an expected lifetime of 5700 hours the safety factors of the gears are calculated, and the smallest safety factors in bending and surface fatigue are found in the helical gear N_2 and N_4, respectively. They are given as:

$$N_{b2} = 1.37 \tag{2}$$

$$N_{s4} = 1.50 \tag{3}$$

Clutch Mechanism
The purpose of the clutch mechanism, seen in Figure 9, is to provide a means to mechanically disengage the motor from the steering wheel, such that the tractor can still be used without relying on the solution. This is done by using a dog clutch mechanism, which uses interference as a way to transfer the torque.

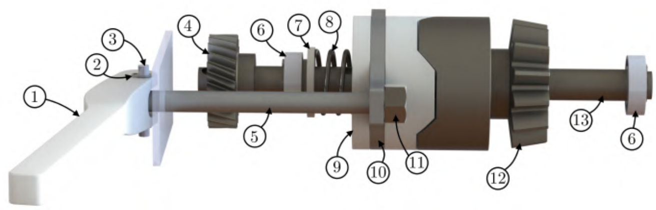


Fig. 9 The clutch mechanism. (1) Handle, (2) Snap ring, (3) Pin, (4) Gear, (5) Shifter rod, (6) Bearing, (7) Spring plate, (8) Spring, (9) Sliding clutch, (10) Shifter, (11) Nut, (12) Rotating clutch/Bevel pinion, (13) Shaft

The automatic disengagement works by making the teeth of the clutch tapered at an angle, β, such that when the motor reaches the nominal torque, the clutch disengages. Because the torque affecting the teeth reduces with increasing radius, the taper angle should change with radius. It is assumed that the change in angle from the inner radius to the outer, is linear. Therefore, only the angle at these two positions are calculated based of the equilibrium equations seen in Equation 4.

$$\begin{cases} \mu F_{N,full} - \frac{T}{r}\sin(\beta) + \left(\frac{T}{r_{key}}\mu + k_s(x_0 + 5\,\mathrm{mm})\right)\cos(\beta) & = 0 \\ F_{N,full} - \frac{T}{r}\cos(\beta) - \left(\frac{T}{r_{key}}\mu + k_s(x_0 + 5\,\mathrm{mm})\right)\sin(\beta) & = 0 \\ \mu F_{N,half} - \frac{T}{2r}\sin(\beta) + \left(\frac{T}{2r_{key}}\mu + k_s(x_0)\right)\cos(\beta) & = 0 \\ F_{N,half} - \frac{T}{2r}\cos(\beta) - \left(\frac{T}{2r_{key}}\mu + k_s(x_0)\right)\sin(\beta) & = 0 \end{cases} \tag{4}$$

The angles are found to:

$$\beta_i = 32.33^\circ \qquad \beta_o = 51.97^\circ \tag{5}$$

The manual disengagement works by using a shifter that fits in the groove of the sliding clutch, as seen in Figure 10. By using a groove it is possible to have the sliding clutch rotate, whilst keeping the shifter stationary.

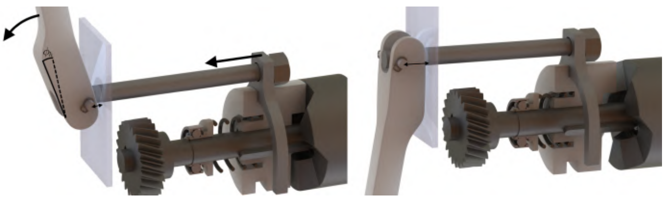


Fig. 10 Shifter mechanism

Shaft Design

In Figure 11 the disposition of the elements on the shaft and their functionality can be seen. The torque is transmitted by the shaft gear (12) to the shaft (1), this provokes the rotation of the sliding clutch (9). The sliding clutch will engage or disengage with the bevel pinion/rotary clutch (6) depending on the compression of the spring (13).

To hold the shaft and keep it aligned, two bearings will be used, which are the bearings nr. 4 and 1, that correspond to the figure numbers as (3) and (10). Another two bearings are used in the design, that are bearing nr. 3 and bearing nr. 2, which correspond in the figure with the (5) and (7), respectively. These bearings are used for allowing the bevel pinion (6) to remain static when it is not engaged and transmit torque when engaged by the sliding clutch (8). For transmitting the

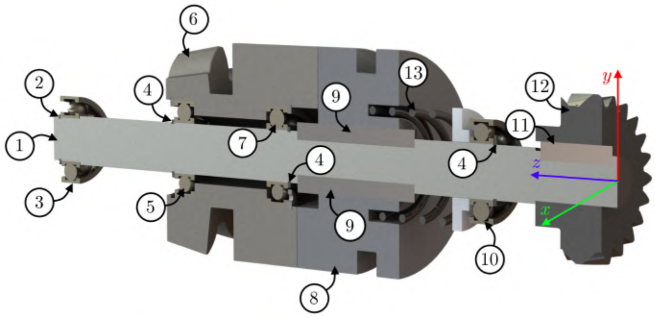


Fig. 11 Render of the shaft. (1) Shaft, (2) External snap ring, (3) Bearing, (4) External snap ring, (5) Bearing, (6) Bevel pinion, (7) Bearing, (8) Sliding clutch, (9) Key, (10) Bearing, (11) Key, (12) Shaft gear, (13) Spring

torque with both the shaft gear and the sliding clutch, two keys are used, that have the corresponding numbers (11) and (9). Finally, for keeping the bearing nr. 4 (3) and nr. 3 (5) in place, two snap rings are added. The rest of the bearings, are held in place thanks to the changes of section of the shaft, as well as the slots of the casing where they are inserted.

The required shaft dimensions are determined using three different failure theories. These determine the shaft diameter based on the maximum normal stress, the maximum shear stress and the maximum equivalent Von Mises stress, respectively as seen in Equations 6, 7 and 8.

$$D_{Nstress} = \sqrt[3]{\frac{(16 \cdot (M_{tot} + \sqrt{(M_{tot}^2 + T_{tot}^2}))}{(\pi \cdot \sigma_{Amax})}} \tag{6}$$

$$D_{Sstress} = \sqrt[3]{\frac{(16 \cdot \sqrt{(M_{tot}^2 + T_{tot}^2)}}{(\pi \cdot \tau_{Amax})}} \tag{7}$$

$$D_{VonMises} = \sqrt[3]{\frac{32 \cdot \sqrt{(M_{tot}^2 + \frac{3}{4} \cdot T_{tot}}}{(\pi \cdot \sigma_{Amax})}} \tag{8}$$

Using the different theories leads to the different diameters seen in Figure 12.

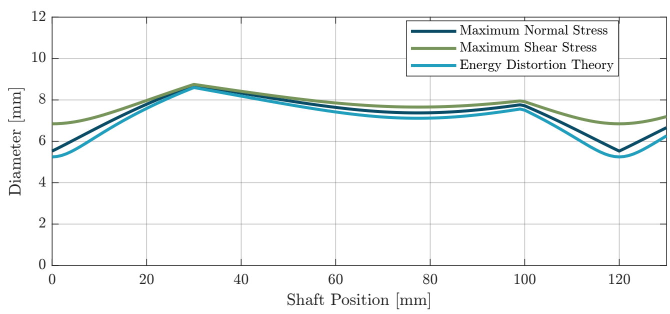


Fig. 12 Shaft diameters according to different theories

To be able to use readily available bearings, the shaft is manufactured in $8\,\mathrm{mm}$ and $9\,\mathrm{mm}$ segments.

Casing and Stationary Mount

The function of the stationary mount is to provide a stationary reference for the casing. Without this reference the casing would rotate around the steering wheel instead of turning it. The stationary mount is designed to clamp around the steering column using straps, as this allows for a compact mount that fits multiple different sizes and shapes of steering columns. The stationary part can be seen in Figure 13.

The stationary mount is constituted by the adjustment plate (1) which is a bent sheet metal part. The hole and curved slots allow for angular adjustment, allowing the fixture to fit different steering columns. The adjustment plate is attached to the connection by bolts (4). The strap (7) is wrapped around the steering column and is connected to the support pin (2) and the tightening pin (5). The strap support pin is not fixed to the adjustment plate and is only kept in place when the strap has tension. This allows for easy removal, if the strap needs replacement. The adjustment plate has a slot through which the belt passes in the side, where the support pin sits. The tightening pin is tightened by adjustment of the tightening bolts (6). The support bracket (3) sits in between the sides of the adjustment plate and functions as reinforcement for this. The bracket is mounted to the adjustment plate on the leftmost side by bolts, whilst the threaded holes in the rightmost side allow the tightening bolts to be tightened.

In order to mount the stationary part to the casing, the connection is designed. The connection is made such that it is be able to adjust to different relative positions of the stationary mount and the casing. The connection is shown on Figure 14.

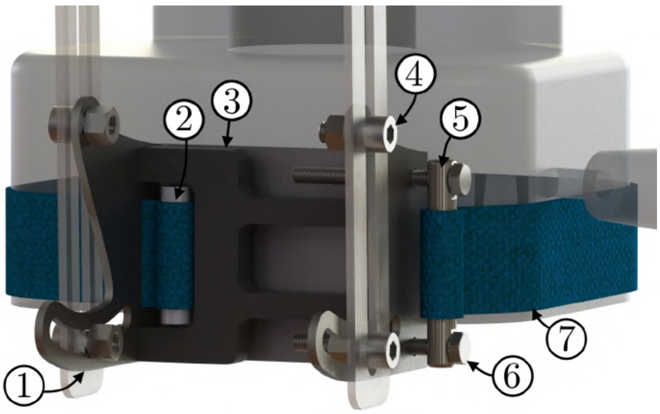


Fig. 13 Stationary mount. (1) Adjustment plate, (2) Strap support pin, (3) Support bracket, (4) Bolts for mount adjustment, (5) Tightening pin, (6) Bolts for strap tightening, (7) Strap

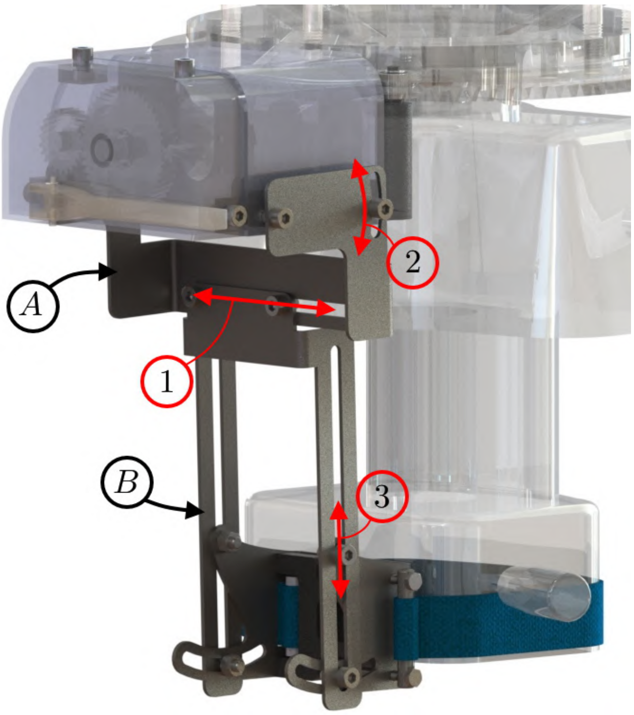


Fig. 14 Depiction of the connection and its degrees of freedom. The annotations mean the following: (A) connection to the casing, (B) connection to the stationary part, (1) relative lateral movement, (2) relative rotational movement, (3) relative longitudinal movement

The casing, as seen on Figure 15, encompasses the most essential components of the system and allows for transmission of mechanical power from the motor to the

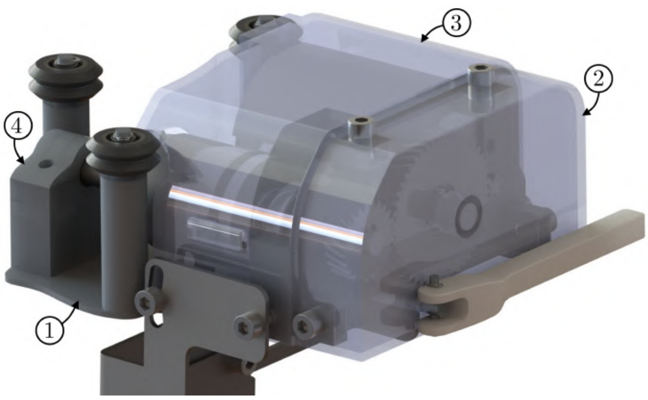


Fig. 15 Portrayal of the casing, and its individual parts. (1) platform, (2) gear cover, (3) main cover, (4) bearing house

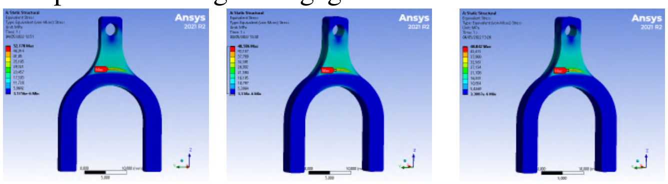
crown gear, thus also to the steering wheel.

6. Model Validation

In order make sure that some of the more complex part will not fail due to static overload, the shifter, the clutch and the clutch's key slot are analysed using the Finite Element program ANSYS.

Shifter Analysis

The shifter is analysed using three different materials; PLA, POM and aluminium. The maximum load on the shifter is $40\,\mathrm{N}$, which occur when the shifter spring is compressed during disengagement.



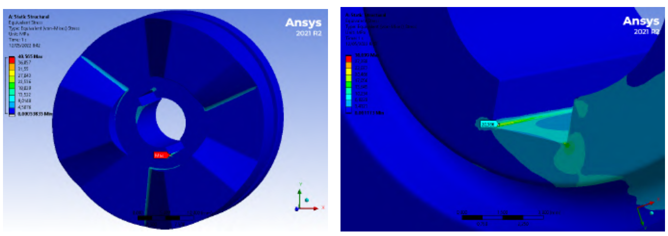
(a) Aluminium (b) POM (c) PLA

The three analyses show a similar equivalent stress of approximately $50\,\mathrm{MPa}$ in magnitude. Considering the yield strength of the 3 materials, $162.5\,\mathrm{MPa}$ for aluminium, $64\,\mathrm{MPa}$ for POM, and $52\,\mathrm{MPa}$) for PLA, the shifter will have the following safety factors:

$$n_{alu} = 3 \qquad n_{POM} = 1.3 \qquad n_{PLA} = 1.06$$

The safety factor for the aluminum is unnecessarily large and the safety factor for PLA is theoretically good enough, but it is deemed too low, as the safety margin is only 6%. POM is therefore chosen as the material, as it gives a 30% safety margin and is a cheap and is easy to manufacture.

Clutch Analysis



(a) Maximum stress on teeth (b) Maximum stress on key

Fig. 17 Von Mises stresses in sliding clutch

The analysis of the clutch was made with the teeth as fixed supports, and the force applied to opposing key slot faces. The value applied, was obtained by converting the maximum torque in the shaft, to a force at the diameter of where the key is located. Figure 17 shows the stress distribution over the clutch surface and teeth. The location of the maximum stress can be seen in Figure 17 to be in the key slot. The stress has an approximate value of $40.5\,\mathrm{MPa}$, which gives the clutch a safety factor of:

$$n_{clutch} = 1.53$$

7. Discussion

Production and Economics

To estimate the cost of the solution, the price of the readily available components is summed up with the estimated price of manufactured components, along with their intended production methods. The final estimations of the cost of different components can be seen in Table III, where it can be seen that the gearing accounts the majority of the total cost.

Components	Price [DKK]
Stationary Mount	221
Connection	64
Gearing	3073
Shaft	349
Clutch	259
Casing	407
Total	4373

Tab. III Prices of particular components of the system, as well as the total cost of the system

Prototype

To test whether the system works as intended, a prototype of the system is constructed. The prototype is intended to highlight the functionality to determine whether the system is feasible or some parts of it should be changed. Due to limited production facilities, few of the components can be manufactured as intended. Thereby the maximum permissible stresses, stiffness and the tolerances of the parts for the prototype are lower than for the intended solution. It is therefore not possible to validate whether the system is capable of withstanding the intended load. Furthermore, the deformation and fitting of the elements is likely to differ from intended. Pictures of the prototype mounted in a tractor can be seen in Figure 18.

Through testing the functionality of the prototype is proven to be mostly valid. The torque transfer happens as intended, with some friction in the helical gears, likely due to bad tolerances. The manual clutch disengagement is the major flaw of the design, as the shifter is not able to disengage the clutch by pulling the

Fig. 18 Picture of the prototype mounted to a tractor steering wheel column

rod using the handle, as this creates a bending moment around the length of the shaft, increasing the friction.

As for the bevel gear assembly, the v-shaped wheels manage to align the bevel gear and pinion gear correctly, as well as to provide a rigid connection between the casing and the crown gear. The fixture also worked as intended, since it is possible to mount it to different steering wheel column layouts. Furthermore, the fixture achieves the task of connecting the casing and the stationary part.

Requirements and Specifications

Table IV shows fulfillment of physical requirements and design specifications. Fulfillment can be either fulfilled, not fulfilled, partially fulfilled and undetermined, whereas 'undetermined' means that it was not possible to test whether the requirement or specification was fulfilled.

Requirement/Specification	Fulfillment
Be able to fit steering wheels with three spokes separated by 120°	Fulfilled
Be able to fit a steering wheel with no access to the steering shaft	Partially fulfilled
Be able to fit steering wheels with varying out of plane spoke angles	Theoretically fulfilled
Be able to fit steering wheels with varying thickness	Theoretically fulfilled
Be able to be avoid lever and dashboard interference	Partially fulfilled
Installation Time	Fulfilled
Tools Needed for Installation	Partially fulfilled
Amount of Required Specialized Tools	Fulfilled
Amount of Components Needed to be Replaced	Theoretically fulfilled
Installation Operations	Not fulfilled
Manual Deactivation	Partially fulfilled
Automatic Deactivation	Fulfilled
Overall Component Cost	Fulfilled
Dimensions of the Mechanical System	Partially fulfilled
Minimum Number of Mounting Positions	Partially fulfilled
Torque at the Steering Wheel	Undetermined
Steering Wheel Angular Velocity	Undetermined
Steering Wheel Movement Smoothness	Undetermined
Weight of the Mechanical System	Not fulfilled

Tab. IV Table showing fulfillment of physical requirement and design specifications

Motor

As the project was limited to a specific motor due to the LeadEng collaboration, most of the components were designed to withstand the higher than necessary loads, produced by this. Had a more suitable motor been chosen it would have been possible to design a more compact solution using smaller components manufactured using lighter materials.

8. Conclusion

A prototype has been manufactured and tested, which served as a way to examine the functionality and feasibility of the design. It has been demonstrated that most of the system functionality are valid, although some tests were limited. The major flaws of the system are the shifter mechanism and the provided motor. In the end, it can be concluded that the objective has been mostly achieved, as the design has been proven to be feasible to a large extent, except for the aforementioned flaws and other minor shortcomings.

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