Innovation in Motion

Potentials of powder metallurgical technology in clutch systems in the subsystem of the electrified drive train

Master Thesis

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ABSTRACT

In the wake of stricter exhaust gas regulations from the European Union, the automotive industry leads to the electrification of cars.

For a component supplier such as Miba Sinter Austria GmbH, this change is interesting because sintering technology can be used to produce new components for these electrified drive trains.

In the course of this master thesis, various hybrid and electrified drive concepts has been analyzed according to the criteria of the use of a clutch and the feasibility of sintering production. Clutches offer potential for powder metallurgic technology.

The results are components adapted to the powder metallurgic production and subjected to a finite element method calculation.

A chain wheel has been selected as the component with the highest potential. The verification of the part requires an endurance test with high torque.

Such a test bed is not yet available at Miba, and therefore a concept has to be developed and selected according to internal criteria.

A test bench has been designed, which requires several considerations. A proposal to carry out the test is suggested.

KURZFASSUNG

Im Zuge strengerer Abgasrestriktionen der Europäischen Union, geht in der Automobilindustrie der Trend in Richtung elektrifizierten Antrieb.

Für einen Komponentenlieferanten, wie die Firma Miba Sinter Austria GmbH, ist dieser Wandel interessant, da mithilfe der Sintertechnologie neue Komponenten für elektrifizierten Antriebsstränge produziert werden können.

Im Zuge dieser Masterarbeit wurden verschiedene Hybrid-, sowie elektrifizierte Antriebskonzepte analysiert und nach dem Kriterium der Verwendung einer Kupplung sowie nach der Möglichkeit der sintertechnischen Fertigung analysiert. Die Kupplungen bieten hier Potentiale für die Sintertechnik.

Das Ergebnis ist die Auflistung von Bauteilen, welche an die PM Fertigung angepasst wurden und wahlweise einer FEM-Berechnung unterzogen wurden.

Ein Kettenrad wurde als Bauteil mit höchstem Potential für die Sintertechnik ausgewählt. Die Verifizierung des Bauteils verlangt nach einer Dauerlaufprüfung mit hohen Momenten. Da im Hause Miba noch kein vergleichbarer Prüfstand vorhanden ist, wurde im Zuge dieser Masterarbeit ein Prüfstandskonzept nach innerbetrieblichen Kriterien entwickelt.

Zur Prüfung des Kettenrades ist ein Prüfstand konstruiert worden, welcher zahlreicher Überlegungen bedarf und es wird bis hin zum Vorschlag zur Durchführung der Versuche eingegangen.

CONTENT

1 INTRODUCTION

The change in the automotive industry towards alternative drives opens the possibility for component manufacturers to produce components for the electrified drivetrain. Miba Sinter Austria GmbH works in the powder metallurgic business and provides parts for cars. This change offers the option to create components apart the standard portfolio. The increasing production volume of hybrid and electric cars will make a PM production economic.
The object is to scan all these new drive trains and identify possible parts for the sintering technology and finally evaluate one of these parts.

1.1 INITIAL SITUATION

Miba has already carried out tests on alternative drives and purchased components for a precise analysis. The company has in-house expertise in sintering technology and a wide range of test benches for testing common sintered components. These test benches aim to verify the calculations and provide the testing option of their components.

1.2 TASK DEFINITION

The initial task definition of the Master thesis „ Potentials of PM technology in clutch systems in the subsystem of the electrified drive train" has been:

- Overview about electrified drive concepts (Hybrid, eAxles, transmission, DHT, All wheel connections, …)
    - What are the requirements and functions of these actuation and clutch systems?
    - Advantages and disadvantages of different Systems (installation space, number of components, complexity of the components, ….)
- Feasibility study
    - For which parts of clutches exists a potential for powder metallurgic technology (PM restrictions, geometry, height, number of plains, ...)?
    - What are the requirements with respect to precision and strength of the parts (if possible)?
    - Which geometrical adaptations of the components are necessary to be able to be represented in PM technology?
- Production of prototypes
    - Identification of a clutch component, which is suitable for PM technology
    - Support in the component calculation
    - Support in the prototype manufacturing
- Validation of the component
    - Conception of a test bench (dynamic tests)
    - Definition of a test procedure
    - If possible, execution and analysis of component test on prototype base

1.3 TIME SCHEDULE

timetable Master Thesis

Actual KW 13

28.03.2019

Group	Activity	start(KW)	duration (KW)	actual start	actual duration	percent concluded
Creating of an overview about the action and clutch systems	Research	37	1	37	1	100%
	Theoretical possibilties of hybrid configurations	37	1	37	1	100%
	Advantages and disadvantages of different systems	38	1	37	2	100%
	Conclusion of the task block and presentation	39	1	38	1	100%
Feasibility study for the PM production	Familiarization with PM technolgy in detail with PM design	39	2	39	0	100%
	Selection of parts with potential for the PM technology	39	3	39	0	100%
	Requirements of material strength and tolerances (if possible)	41	3	40	5	50%
	Adaption to PM production	44	2	45	2	100%
	Conclusion of the task block and presentation	45	1	45	2	100%
manufacturing of an component	Selection of one component	45	1	44	3	100%
	Support with the FE simulation	47	3	45	2	100%
	Support in production	49	3	45	1	100%
	Conclusion of the task block and presentation	51	1	46	1	100%
validation of the component	Creation of a test concept according to dynamic and static analysis	1	5	49	3	100%
	Definition of the test procedure	4	2	49	3	100%
	Creation of different test bench concepts	2	2	49	3	100%
	Evalation of the different variation and selection of one	3	2	3	2	100%
	construction and calculation of a test bench	2	6	3	10	100%
	If possible, execution and analysis of component test on prototype base	20	5	0	0	0%
	Conclusion of the task block and final presentation	8	1	14	1	100%

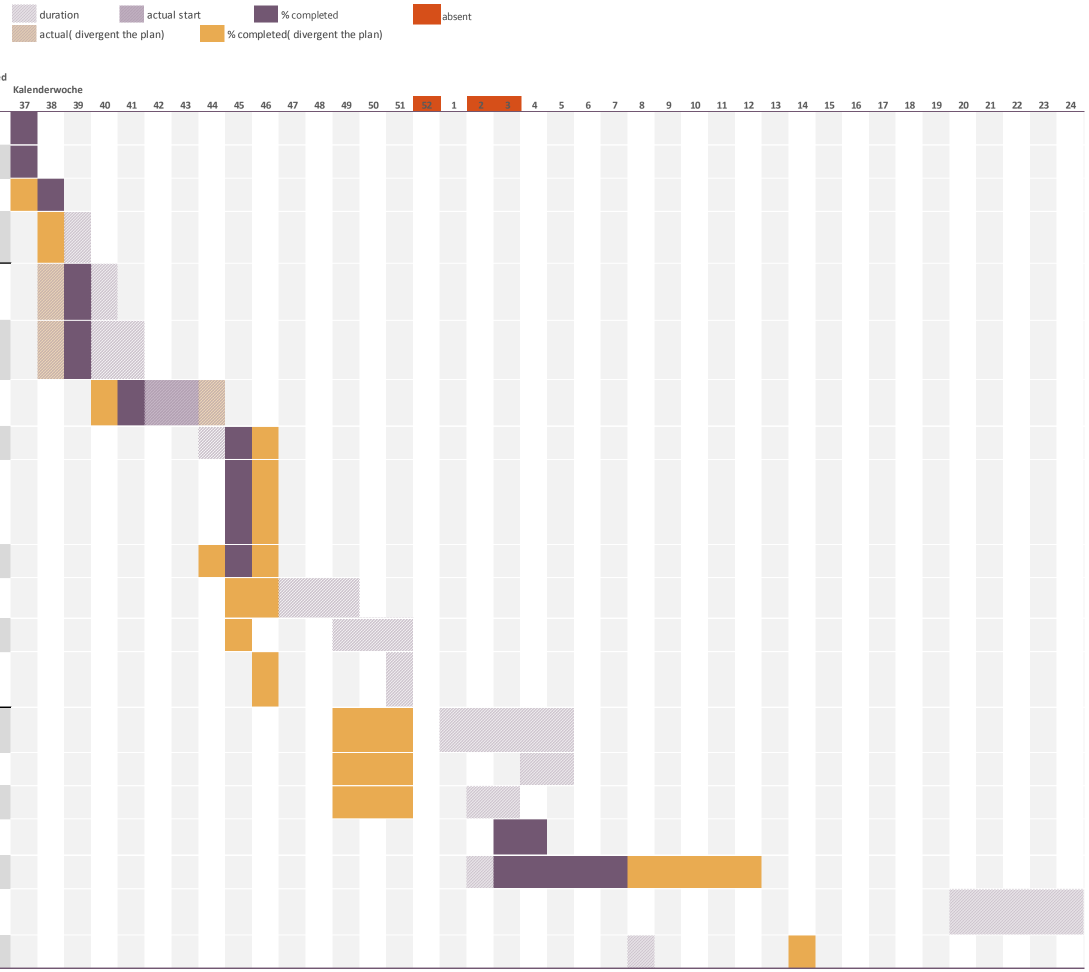

Figure 1: Timetable of the Master thesis

1.4 SHORTCUTS

PM→ Powder metallurgic

ICE→ Internal combustion engine

EM→ Electric machine

DHT→Dedicated hybrid transmission

AWD→All-wheel drive

CV shaft→ Constant velocity shaft

2M-concept→ two machines concept

1.5 SYMBOLS

Variable	*Description*	*Unit*
M	Torque	Nm
I_p	Polar moment of inertia	mm^4
φ	Angle	rad
G	Shear modulus	N/mm^2
ω	Angular velocity	rad/s
D	Diameter	mm
n	Rotational speed	rpm
β	Deflection angle	°

2 BASICS IN ELECTRIFICATED DRIVE TRAINS

2.1 THE HYBRID CAR

The aim of using a hybrid system is the prevention of CO_2 -emissions. There are some aims, which require some specifications for the hybrid system:

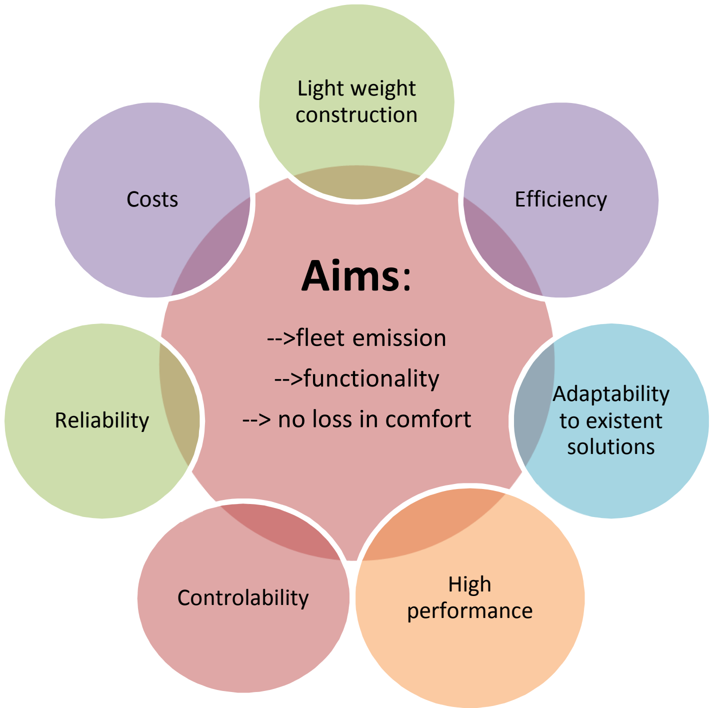

Figure 2: Aims and requirements [1]

All these requirements are needed to supply the user with an affordable hybrid car, without loss of functionality and comfort compared to a conventional ICE. The legislation for the fleet average of a brand is shown in the Figure 3.

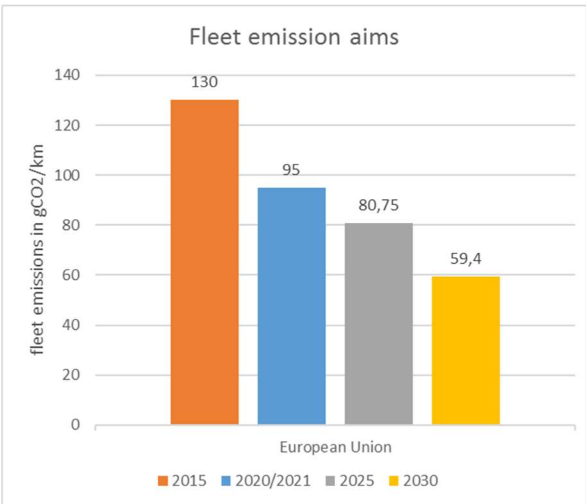

Figure 3: Fleet emission aims [2] [3]

According to the CO2 emissions of a car, these reductions are not reachable with conventual combustion engines. The solution to this problem is the electrification of the drive train. There are the options of fully electric vehicles and hybrid vehicles. In hybrid systems two drive units are needed, an ICE and an EM with the battery. This leads to a heavier drive train. Thus, there is a need to combine the advantages of both drives with a storage unit to create a successful concept. The focus is about the usage of the PM technology in these drives, but therefore it is important to know where a clutch could be implemented and, which functionality could be integrated. Therefore, hybrid cars are classified according to the two following ideas: [4]

- The architecture of the drivetrain
- The degree of electrification

2.1.1 Classification of the architecture of hybrid drive trains

There is the opportunity to differentiate a hybrid drive train in three ways. First of all, there is the opportunity to distinguish the structure of the combination for ICE and electric engine in the following three types. [4]

- Serial hybrid drive train
- Parallel hybrid drive train
- Power differentiating and combined hybrid drive trains.

2.1.1.1 The serial hybrid drive train

The serial hybrid uses the ICE for generating the electric energy, which can be stored in batteries. Further on this electricity supplies the EM to propel the wheels.

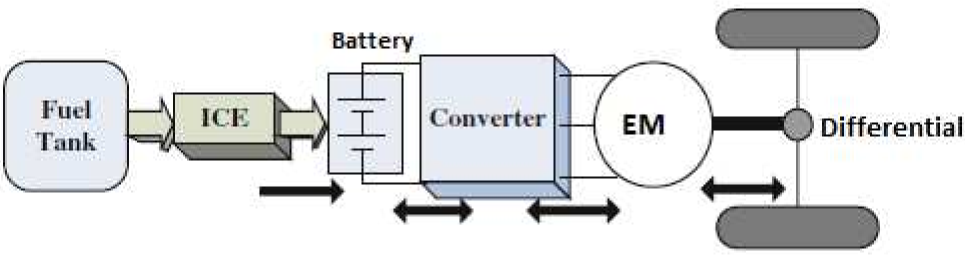

Figure 4: Serial hybrid drive train [5]

The big advantage of this system is the fact that the ICE works in the most efficient point of the engine characteristic. Using the engine at the optimal operation point saves fuel, but the conversion of rotational energy to electric energy, the saving in chemical energy and the conversion backward have several losses. This principle can operate more efficient than a direct connected ICE. Suboptimal configurations are often used in reality due to costs. A conventual engine can be adapted with a generator to use it for serial operation, but the engine is mostly not designed for this operation mode. Nevertheless, this concept is interesting for electric vehicles cars with a range extender, like the BMW I3. There is a twin cylinder engine of a motorcycle implemented. [4]

2.1.1.2 The parallel hybrid drive train

In the parallel drive train, the ICE and the EM work separately to propel the wheels. There are several options of parallel configurations which are named between P0 until P4. They are differed by the position of the EM.

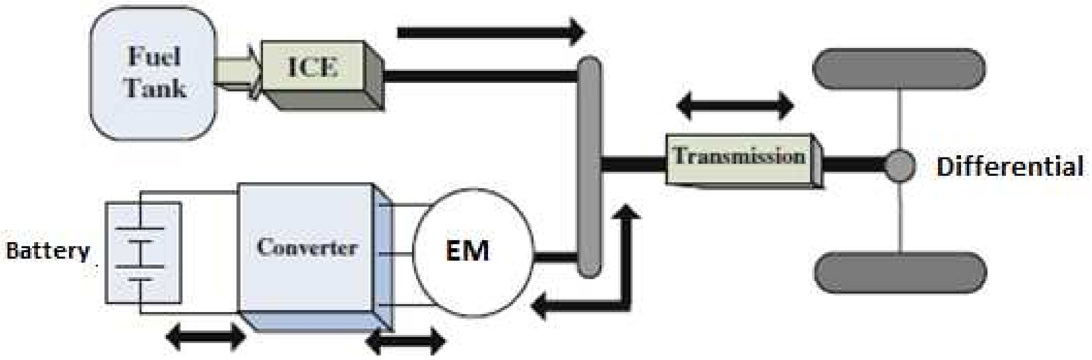

Figure 5: Parallel hybrid drive train [5]

P0 configuration

The P0 configuration has the EM at the front-end accessory drive of the ICE. It means the EM is connected with a pulley to the ICE. The EM is often also called belly starter/generator.

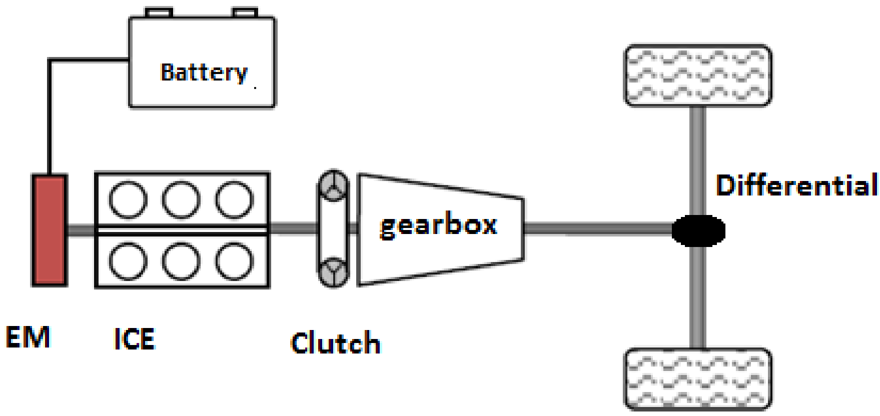

Figure 6: P0 configuration [6]

In this application, an EM with no freewheel option is used. Hence the electric machine can be used to propel and to generate. There is no disconnection from the belt. This configuration has no need for a disconnection clutch.

P1 configuration

This combination shows the EM in between the clutch and the ICE. Both are continuously connected together. This configuration doesn't need a clutch for disconnection.

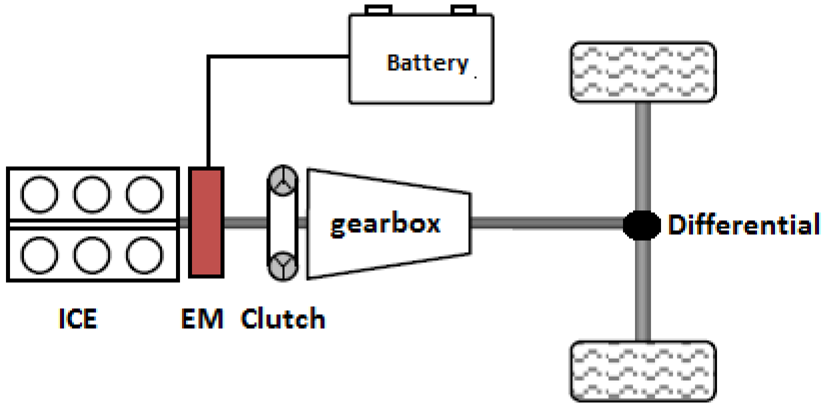

Figure 7: P1 configuration [6]

P2 configuration

Here, the EM is located between the clutch and the gearbox. The advantage is the fact that the option of disconnection of the ICE from the rest of the drivetrain is possible. That means no losses of the ICE occur, when the car is just rolling or just propelled with the EM. A sole driving with EM is possible.

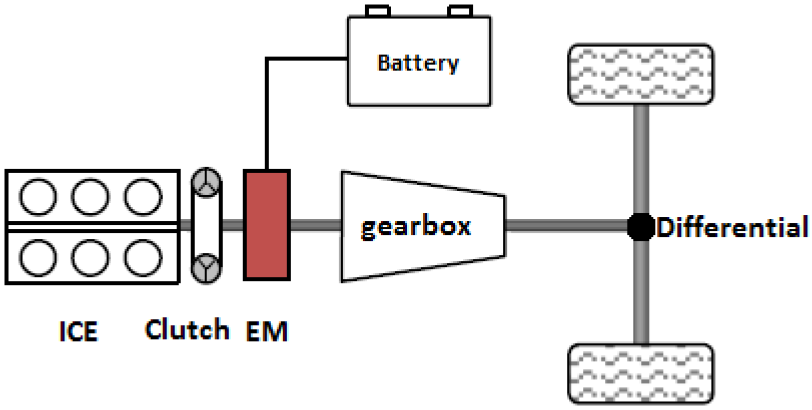

Figure 8: P2 configuration [6]

P3 configuration

In this configuration the EM is positioned between the differential and the gearbox.

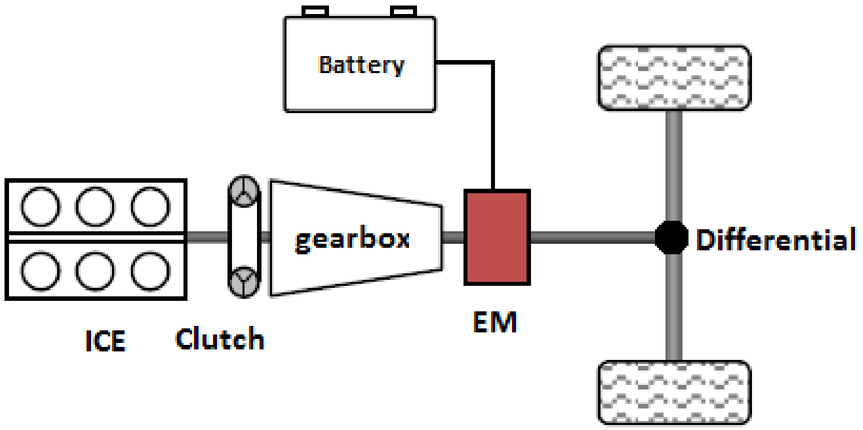

Figure 9: P3 configuration [6]

P4 configuration

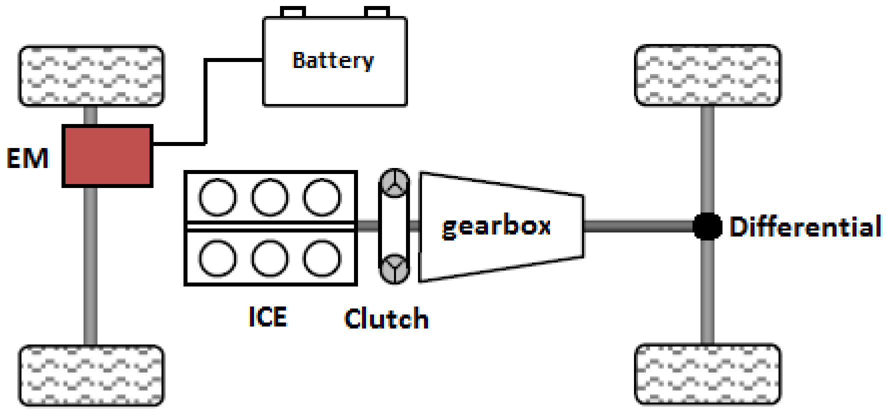

Figure 10: P4 configuration [6]

In this topology, the combustion engine propels one axle of the car. The other axle isn't mechanically connected with a shaft to the ICE. The EM drives the other axis of the vehicle.

2.1.1.3 Power differentiating and combined hybrid drive trains

Power differentiating and combined hybrid drive trains use dedicated hybrid transmission (DHT) and they are divided in two main types:

- Power split – DHT
- Multimode – DHT

Power split DHT

The Power split DHT concept consist of a power transmission with an electrical and a mechanical power path. A planetary gear set is the power distribution component. This concept offers the possibility to drive fully electrical and in a combination of the ICE and the EM.
The structure of a Power split DHT is shown in Figure 11. The main components are an ICE, two EM and the planetary gear set, but there is no need of a clutch. [1]

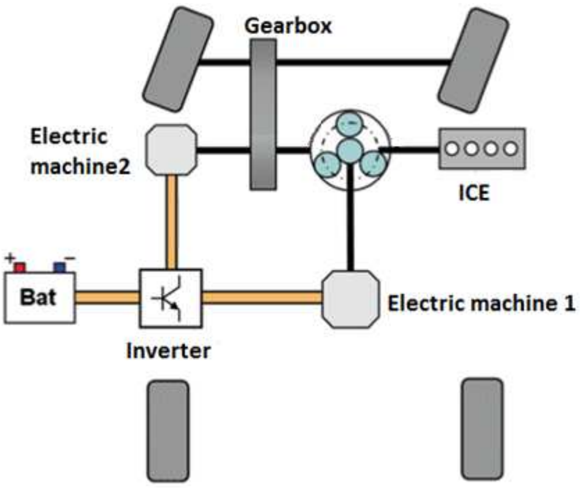

Figure 11: DHT-Power split components [1]

The Table 1 shows the main operation modes of this power split DHT. A control unit decides which combination of the drives is the best suitable for the current situation.

	ICE	*EM1*	*EM2*
Normal drive	Drive		
Slow acceleration/Start Stop/ slow driving			Drive
Fast acceleration	Drive	Drive	
Charging	Drive	Generate	
Recuperation			Generate

Table 1: Operation modes of a power split DHT [7]

Multimode DHT

This concept relies on the possibility to change between different operation modes of a gear box. A parallel, a serial mode and a pure electric mode is possible with this system.
The wheels are driven by an electric motor, an ICE or in combination of both. The ICE drives a generator and can be connected with a clutch to the wheels. The operating mode differs from the speed of the car, because there is no possibility to shift the gears. Figure 12 shows a GKN DHT-Multimode gearbox, which is already used in the Mitsubishi Outlander PHEV.

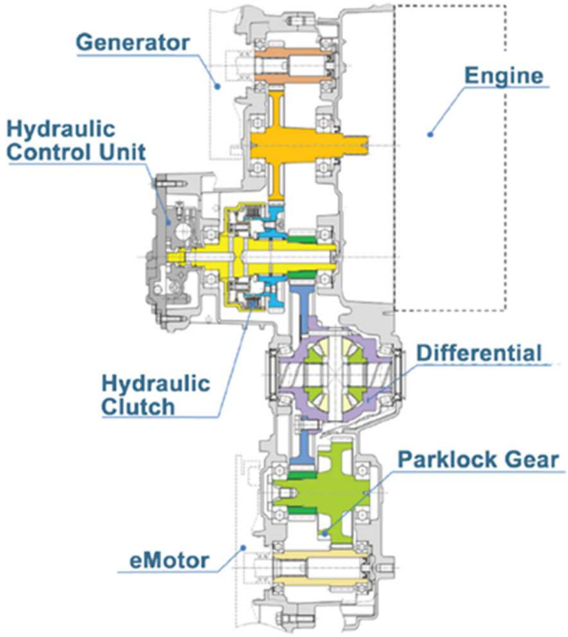

Figure 12: DHT-Multimode gearbox of GKN [8]

Operation mode	*EM*	*ICE*	*Driving situation*
Pure electric vehicle	Drive	Not used	City
Serial hybrid	Drive	Propel the Generator	Acceleration / Uphill
Parallel hybrid	Drive	Drive + Power generation	Highway

Table 2: Different operation modes according to the driving situation [8]

2.2 THE ELECTRIC VEHICLE

The electric vehicle is equipped with one or more electric machines and the corresponding power electronics and batteries. There are several concepts of electric cars, and the difference here is the quantity of EM and their position in the car body.

2.2.1 In wheel EM

The EM and the electronics are placed behind the wheel rim. Therefore, the unsprung mass increase and the so the damper spring constellation has to be changed. The installation space is here the criteria for dimensioning an EM. Often the wheel hub integrated the EM, brakes and the bearing in one part. There is usually a water-cooling system integrated for the transmission of the heat. The power supply is coming from the battery package which is mounted in the car chassis floor. Here is no need for a clutch, because the electric machine is directly mounted on the wheel rim. Hence, this kind of EV drive train is no subject of investigation in this thesis anymore. [9]

2.2.2 eAxle

The eAxle is positioned at the front or the rear the car. It is operating with just one EM for two propelled wheels, placed in between them. Basically, the electric machine drives the axle, but nowadays functions can be integrated in an eAxles. Following different components can be part of such an eAxle:

- Reduction gears
- Differential
- Park locks
- Disconnect clutches
- Power electronics

Figure 13: eAxle made by GKN [10]

These eAxles are also found in the P4 configuration as an All-Wheel-Drive, like in the Volvo XC90.

2.3 FUNCTIONALITY OF THE HYBRID AND ELECTRICAL SYSTEMS

	Parallel	*Serial*	*DHT*	*EV*
Start/Stop	dependent	yes	yes	no need
Boost	yes	yes	yes	no need
Recuperation	yes	yes	yes	yes
Load point shifting	dependent	yes	yes	no need
Electric driving	dependent	yes	yes	yes
sailing	dependent	yes	yes	yes
4WD operation	dependent	possible	no	dependent

Figure 14: Hybrid and EV functions [4]

The following functions can be available with an electrified drive train.

Start stop function

Prerequisites is the EM instead of the conventual combination independent starter and generator. The advantage of an EM is a higher power and can accelerate the ICE faster than a pinion starter, which is just able to start the ICE out of the still stand. The e-machine provide more power to start the ICE and can accelerate it to higher rotational speeds which allows a higher output after starting the engine. [4]

Boost of the engine in low rotational speeds

Due to the fact that the ICE has low torque at low rotational speeds, the electric engine supports the drive train with additional power. This support allows a faster dynamic response at low rotational speeds and improve the driving performance of the vehicle. [4]

Recuperation

The use of EMs allows one to use this part for the generation of energy in delay processes. This recuperation should feed the batteries with power for later use in the electric engine. In case of emergency brakes, there are big delays up to 10 m/s. This means significant electric machines (+200 kW) would be needed, so the field of recuperation is just interesting for slow delay processes, where more energy can be generated and saved. [4]

Load point shifting

This concept is interesting for the slow constant speed of a car, like in a city with speed limits of 30 or 50 km/h. The maximal power output of an ICE is several times higher than the actually needed power. If the ICE operates on low load points, the efficiency is low. If the engine is used in a higher operation point, the energy can be partly converted to direct current and saved in batteries to reuse the electricity again. The load shifting is useful if the increase in efficiency of the ICE is higher than the sum of the losses, which are occurring due to the storage and reuse process. [4]

Electric driving

Electric driving is relevant for constant slow speed according to the low energy consumption. For a middle-class vehicle at a speed of 30 km/h, an energy consumption of 2 kW and for a speed of 50 km/h, 4 kW are required. Acceleration of cars needs a higher amount of power and further on more powerful EM and batteries. [4]

AWD drive with an Electric engine:

The realization of an AWD is quite easy in the configuration of an ICE driven axis on the front axis. On the rear axis there is an electric engine, which intervene in the case of an traction issue. A cost advantage of this system is the loss of the cardan shaft and the middle differential. A disadvantage is the fact that the recuperation of the car happens over the rear wheels. With the switch from ICE mode to electric mode, there is a switch between the driven axels as well and it changes the driving characteristics of the car. [4]

2.4 THE CLUTCH

The focus of investigation lies on the components which are added due to electrification. That means conventual components are not parts of analysis. Furthermore, switchable clutches in the automotive industry are objects of investigation.

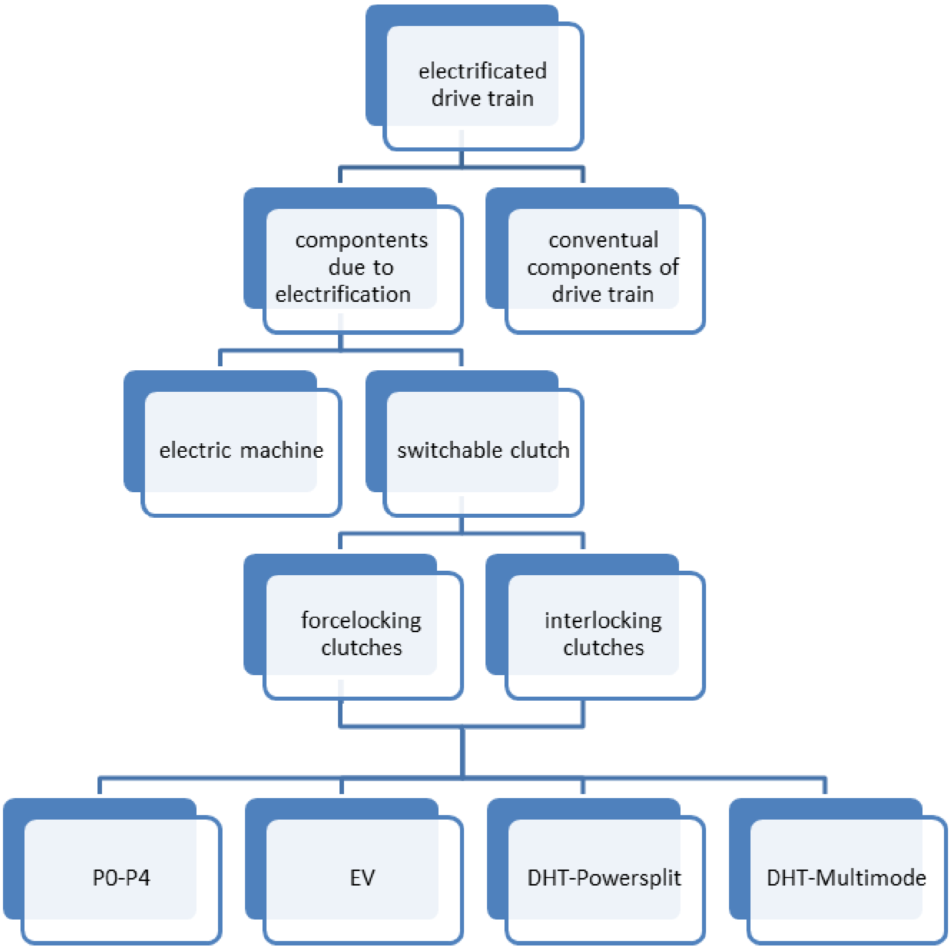

Figure 15: Overview of the types of clutches

2.4.1 Diversification of the clutch in following parts.

The clutch as an object is divided in these main parts. The following parts are object of the investigations to identify possible PM parts.

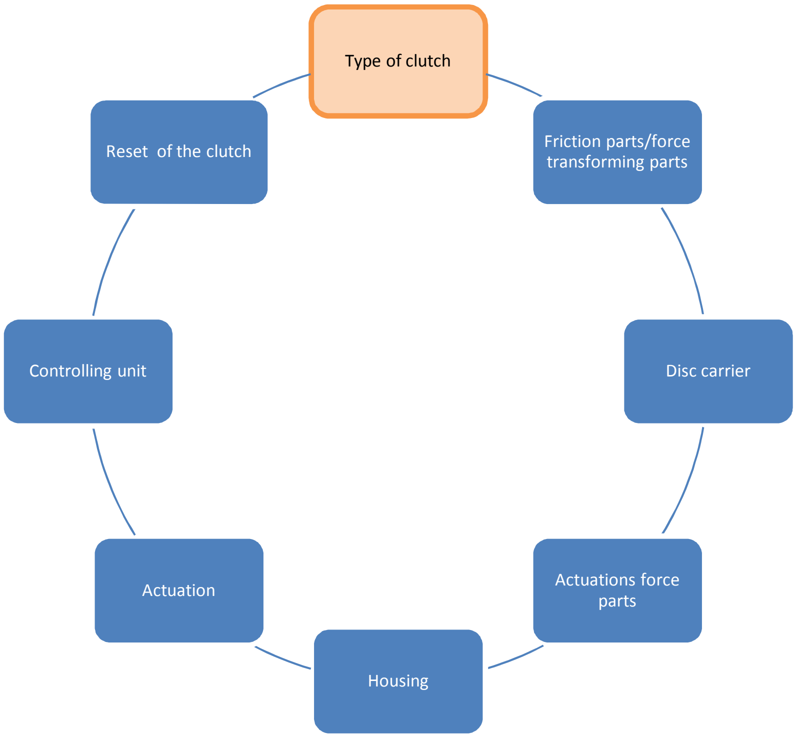

Figure 16: Main parts of a clutch

2.4.2 Types of clutches

The clutches are differed in the two ways of force transformation types.

2.4.2.1 Friction clutches

The first type uses friction discs to transfer torque over the clutch. This clutch connects two parts with different rotational speeds until their velocity is equal or it is just used for the connection between both components.

The principle behind the force transmission is based on a package of friction disc and the steel discs, which are arranged alternately, but at least one friction and one steel disc. Thus, every contact surface to each other produces a friction area. If there is contact between the friction plates a force in the tangential direction occurs, which allows transferring torsional moments over the clutch. Increasing friction pairs offer a transfer of higher torque.

Another application of a friction clutch is the control of the transferring torques through the clutch. The function is similar. Both clutches don’t need to reach the same speed. Different speeds of friction and steel discs are possible. The axial actuation force controls the transferred torque.

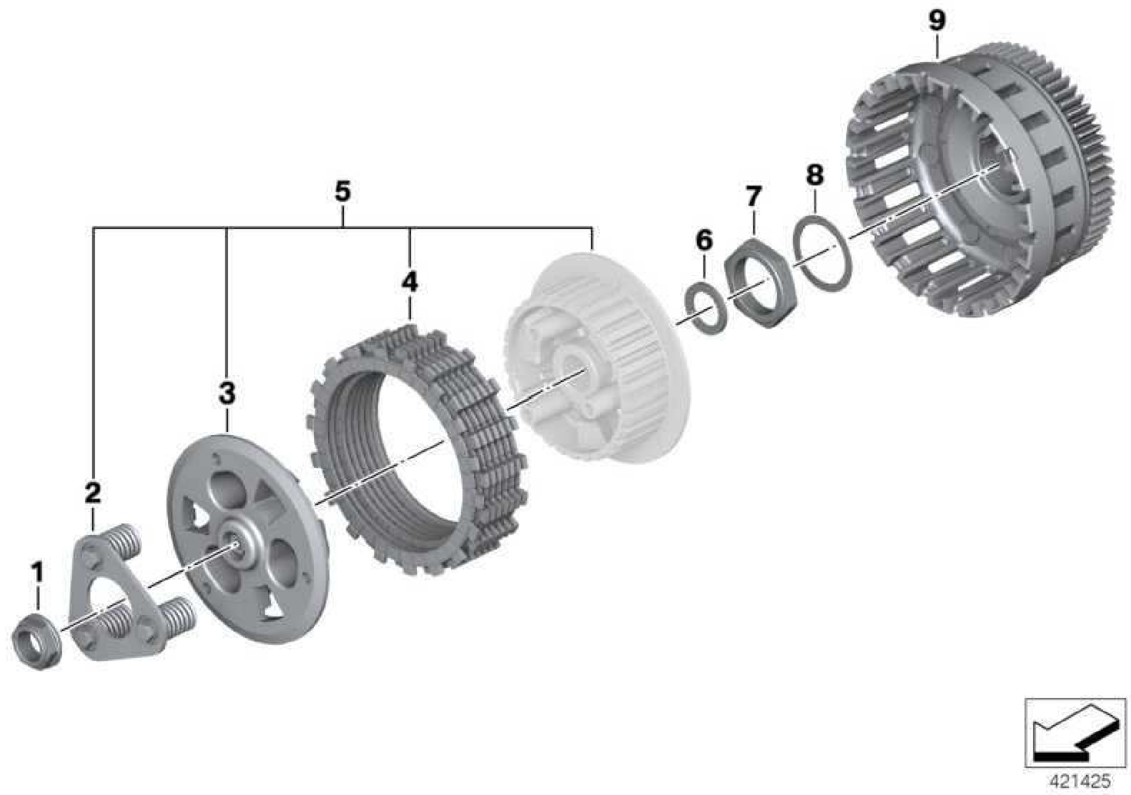

Figure 17: Exploded view of a friction clutch of an BMW GS [11]

2.4.2.2 Interlocking clutches

These clutches are made to connect two parts of the same speed and enable a rigid connection. The forces are transferred due to an interaction of teeth or other geometrical shapes.
There are just two possibilities:

- Connection
- Disconnection

2.5 The Powder Metallurgic Technology

The powder metallurgic technology uses fine powder of metals and this powder is formed under high pressure to a compact part, which is also called green compact. It is placed in a sinter furnace with a temperature below the melting temperature.

The production process is divided in the following main process tasks:

- Mixing
    - Metal powders
    - Additives
        - Lubricants
        - Alloys
- Compaction
- Green machining(optional)
- Sintering
- Sizing (optional)
- Secondary machining(optional)
- Secondary finishing (optional)

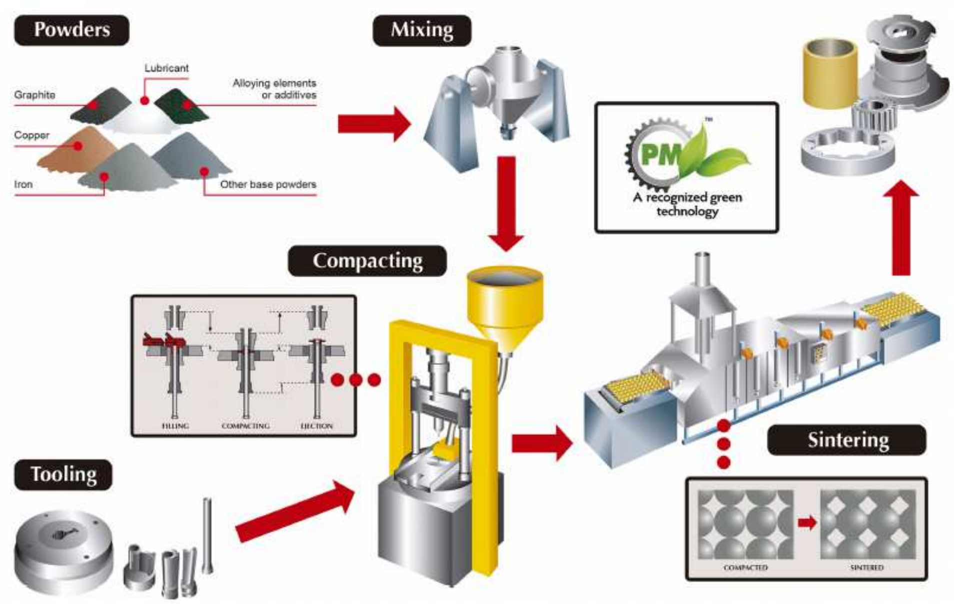

Figure 18: The sintering process [12]

The big advantages of sinter products are the low energy input during the production and a final shape close product after the sintering process and the possibility of individual geometry optimizations count to the benefits as well. A better NVH behavior count also to the advantages. A disadvantage of this process is the quite expensive sinter powder and the belonging allows. A further disadvantage is the lower strength of the material compared to conventual steel.

The relevant objectives of the thesis are the restrictions and options of the sintering process due to the compaction.

2.5.1 Restrictions of the powder metallurgic design

The design of parts made of metal powder follows several restrictions and principles, which should be kept in mind to design a part out of metal powder.

2.5.1.1 Equal density distribution

Density of solid steel parts is equal throughout the whole volume. In the powder metallurgy terms, there is no possibility of equal density distribution. One reason for unequal density distribution is the fact, that the compaction force in the middle of the volume is reduced through the friction force in the middle of the volume. This friction force depends on the height of the part. A so-called neutral zone can be created in the middle of the volume, where a smaller density occurs. Several other criteria like sharp edges, different height at one stamp can occur an unequal density distribution.
A difference in the density can produce distortion during the heating in the furnace and the subsequent cooling down time. An additional hardening process can cause disorientation as well. The aim is to reach an almost equal density distribution for high dimensional stability.
The density of the parts and the strength of the material are directly in relation. Therefore, a high density and equal distribution is the aim.

2.5.1.2 The projected area of the part

The projected area of a part is restricted due to the reason, that the compaction of the powder needs a certain pressure on the surface of the compacting element. This area is limited due to the pressure force of the presses.
The realization of high-density parts and the calibrated parts needs even higher press force on the component. This area depends on the material mix, process and the type of press. A rough estimation of the force per square centimeter is shown in the table underneath.

	N/cm²
Compaction pressure	50000 - 60000
Sizing pressure	60000 - 70000

Table 3: Needed compaction and sizing pressure

2.5.1.3 Number of plains on the part

The geometry of PM parts is limited with the functionality of the presses. In detail, every plain on the part needs one punch, thus the number of plains orthogonal to the compaction direction is limited. In case of the regarding Master Thesis, this limitation is set by 4 upper and lower punches. There is an exclusion of this restriction for high differences of one punch up to 2 mm.

2.5.1.4 Tightness against liquids

Powder metallurgic technology causes a porous structure, which can be detected by microscopes and this results in the fact that a sintered part without additional treatment is not able to seal liquids. The issue is the porous structure at the sealing surface. A densification of the sealing surface or a steam treatment make sintered parts useful for sealing tasks.

2.5.1.5 Minimal wall thickness of one punch

The force on the surface area is high and transmitted through stamps. The fact that these punches need stability against buckling restrains the minimal thickness to 3 mm.
Another aspect for thicker walls is the fact that the radial forces during the compaction cycle are quite high and these forces can displace the punch slightly.

2.5.1.6 Height limitation

The input height of the belt furnace in Miba Sinter Austria GmbH is limited to 60 mm.

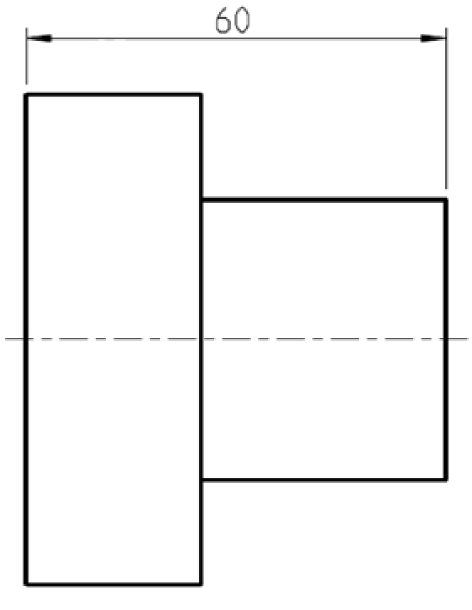

Figure 19: Restriction in the height of sinter components

2.5.2 Limited length due to the calibration process

The in-tool compaction of the already sintered component is a common way to increase the strength of the part. The reason for higher strength is an increase of the density of the part. A density increase is possible up to about 7.4 g/mm^2. The mass of the components is not allowed to increase during this process, so the parts shrink in the high due to the compaction die.

2.5.2.1 Limited geometry orthogonal the compaction direction

The compaction process doesn't allow the creation of geometry orthogonal the compaction direction. There is no option to create holes, grooves, and undercuts. These geometries would need machining.

2.5.2.2 Economic Aspects of the Sinter process

The big advantage of the powder metallurgic technology is the efficient production of components with complex geometry for large quantities with a high repeatability accuracy.
Every part, which is made of metallurgic powder, needs its tool for compaction, and this tool can't be reused for other components and has to have small tolerances, high surface qualities, and high strength. These high requirements for tools led to high cost and therefore a quantity of at about 100 000 pieces per year is recommended for efficient production. A high amount of machining makes them more and more uneconomical.

3 SELECTION OF POSSIBLE PM PARTS

The approach is to identify possible PM parts by examining the different concepts according to three criteria. The investigations for one approach stop if one criterion is evaluated negatively. The aspect of the clutch means there is a clutch in the electrified powertrain which could be partly manufactured out of sinter.

For the selection of the part the following criterion are chosen in the cause of this Master Thesis:

- A necessity of these powertrains is to have a clutch for the electrified power train.
- The diameter of the clutch parts is limited with 150 mm.
- The quantity must be higher than approximately 100000 pieces/annual.

	Additional Clutch	*Size of the clutch components*	*quantity*
P0	-		
P1	-		
P2	-		
P3	depending	unknown	unknown
P4	+	+	+
Serial hybrid	-		
DHT Power split	-		
DHT Multimode	+	+	unknown
Electric Vehicle	+	+	+

Table 4: Classification of concepts according requirements for a sinter process

Table 4 shows us the results of the different concepts. The explanation of the symbols is simple, but every minus immediately causes the exclusion for further investigations.

The concept of P0, P1, P2 serial hybrid, and DHT Power split concepts work without a additional clutch.

The P3 configuration has the opportunity for using a clutch between EM and gearbox, but it is not necessary. Furthermore, this drive has been mentioned several times in the literature, but the investigations have shown no integration in a mass production car so far. This concept won't be examined any more in this Master thesis.

All drivetrains of P0 until P3 have the option of an additional AWD with an additional transfer case, which is already state of the art.

The P4 configuration often integrates a dog clutch to disconnect the EM from the differential. The quantity can be assumed higher than 100000 pieces because this clutch is mounted in all twin-engine series of Volvo XC90, V90, and S90 as well as BMW 225xe and the official announcement is about 300000 already produced electrical AWD units. [13]

The DHT Multimode concept is interesting, because there is a clutch unit integrated in the gearbox with components for PM technology. The only doubt is the number of produced pieces a year.

The dedicated hybrid transmission, the electric vehicle and the transfer case for an AWD of the P0 - P3 concepts are chosen for further investigation.

3.1 PARTS OF THE DHT MULTIMODE CONFIGURATION

This dedicated hybrid multimode transmission is used in the Mitsubishi Outlander PHEV. There are three different modes to change between an electrical, internal combustion engine and parallel drive. The change between them is realized through a hydraulic clutch, which has been investigated.

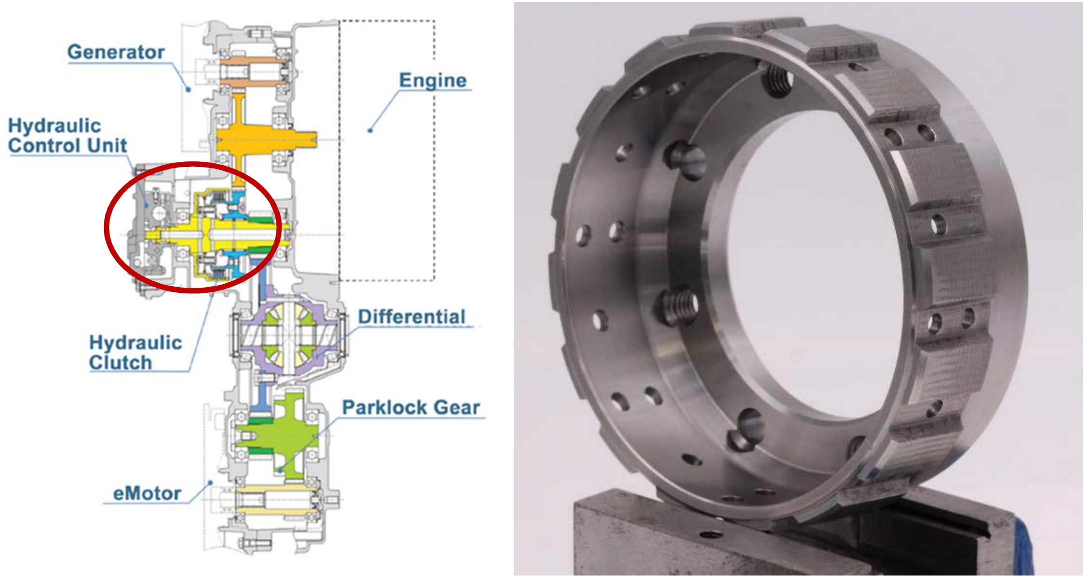

Figure 20: Overview of the system [8] and original inner disc carrier (milled steel part)

Figure 20 shows an inner disc carrier which would be a typical PM - Part. On the outer diameter, there is a toothing which could be manufactured with the sintering process. The radial orientated holes for the oil distribution and the wrenches in axial direction must be either manufactured on the green or already sintered component.

Slightly reconstructing for a sinter optimized part results in the design in Figure 21. This redesign doesn’t influence the functionality of the component or system. The only critical aspect of the geometry is the smaller radius at the end of the part, which is highlighted in brown. This smaller radius is an oil barrier, which forces the oil to flow two the radial holes.

The following figure shows geometry with an adaption for the sintering process.

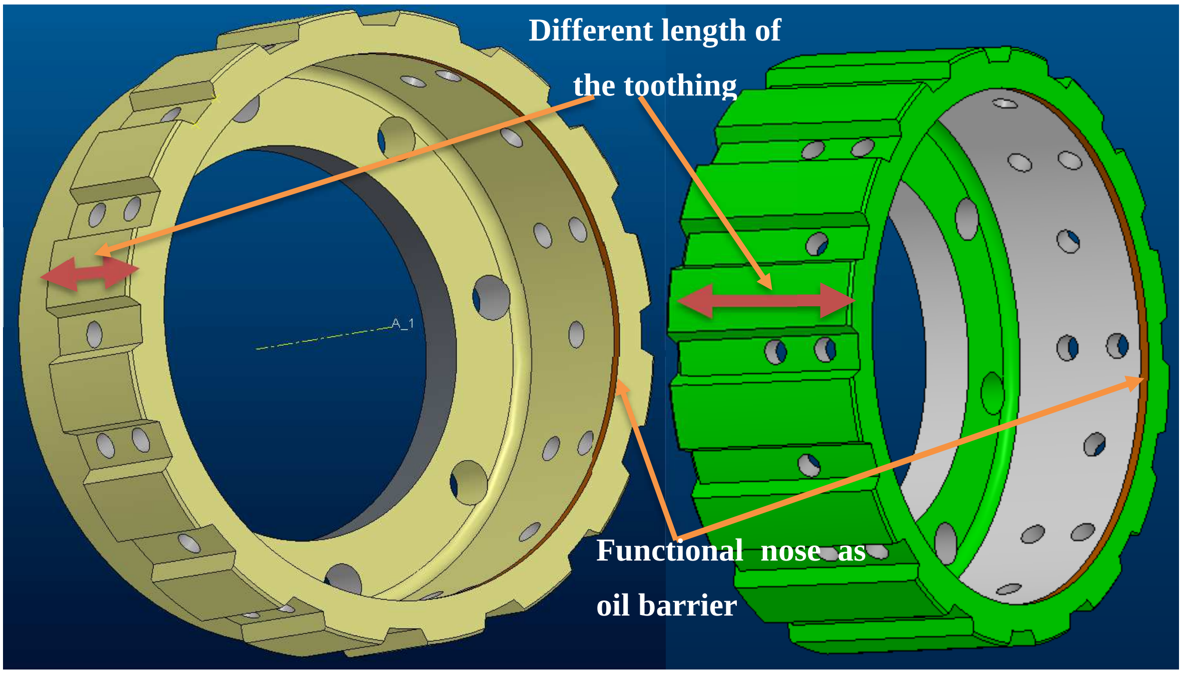

Figure 21: Original inner disc carrier (left) and modified sinter inner disc carrier (right)

A request to the transmission producer concerning the number of built transmissions results that production with PM technology would not have an advantage due to the high initial cost of the tools.

3.2 PARTS OF THE P4 CONFIGURATION

A typical parallel configuration with an ICE drive on one axle and an electric machine on the other one is used in the current AWD vehicles of Volvo and the BMW 225xe.

The cars use an electric engine, which can disconnect the electric machine if there is no need for it. The aim is to reduce the inertia and losses. A dog clutch is integrated into the differential gears.

Figure 22: Disconnecting differential (left) and disconnecting disc (right)

The disconnected disc of the dog clutch is quite an interesting component to manufacture with powder metallurgic. The right figure below shows the construction adapted for PM technology. Additional wavy bumps (not shown) on the back could be used for a better density distribution in the parts, they reduce the density gradient in the feet radius of the interfering teeth. A snap-in connection is integrated in the original component and it could be manufactured in the sintered option as well, therefore the snap-in connection has to be positioned slightly on another position.

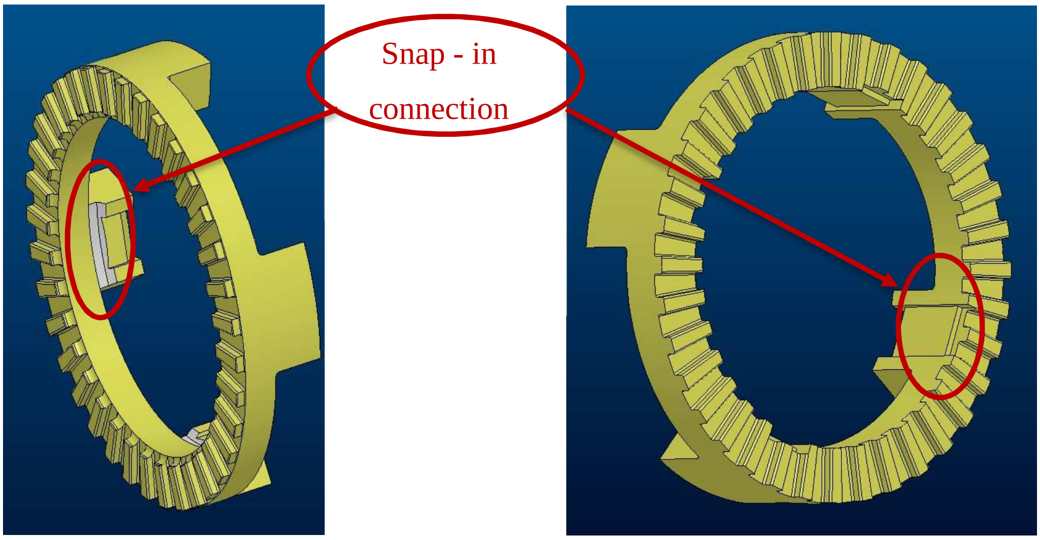

Figure 23: Original disconnect disc (left) and the adapted PM-option (right)

Currently, this disc is a forging part and the information of the power over the clutch is used to create a FEM simulation for the first indication about the stresses.

The transmitted torque over the same disconnection disc is 2000 Nm (BMW 225xe) and 2400 Nm (Volvo XC90) [13].

The claws tolerances in shape and position are needed for assuming a realistic contact relation between the claws, which are not available. The initial simulation is based on optimal interactions of the claws and the load case of BMW.

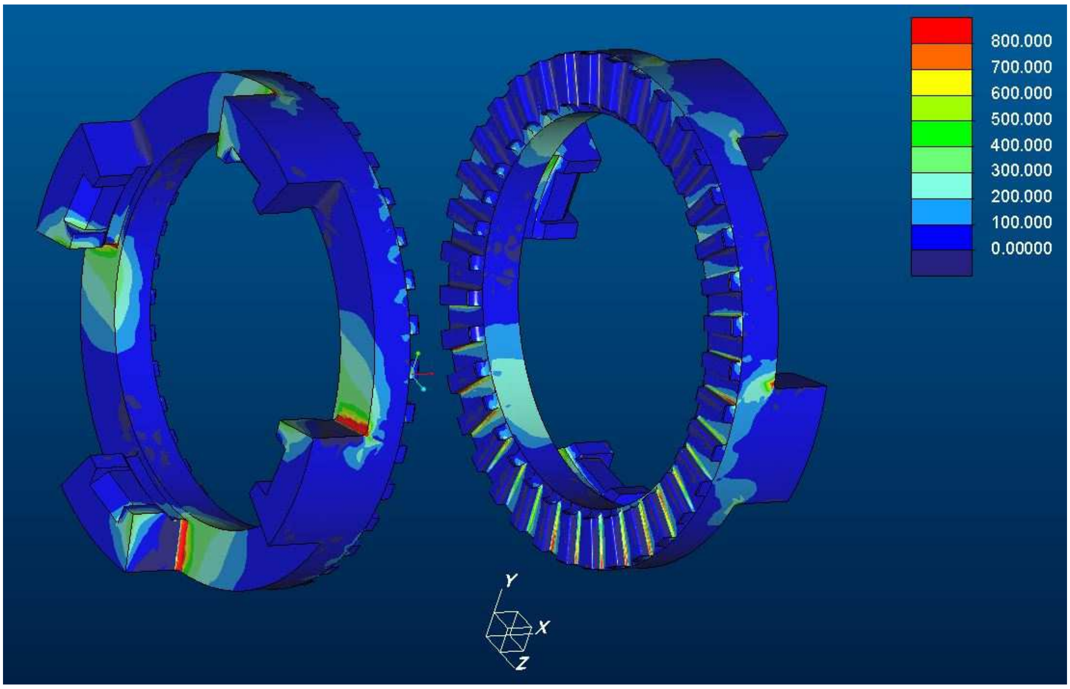

Figure 24: FEM of the disconnect clutch for 2000 Nm

The Finite Element solution shows quite high peaks of tension in the foot circle radius and in the radius of the four transfer elements.

PM technology is critical here due to the result of the max stresses, because the optimal contact situation combined with the lower load already has problems in the area of the transition radius. This part is discarded due to the critical stresses for even the best interaction situation and the lowest load, but if there would been an intensive design study on this part with possible changes in the other components as well, there could be potential for the sinter technology again.

3.3 PARTS IN AN ELECTRIC MACHINE DRIVE

The concept with an electric engine is almost similar to the P4 idea. There is just no ICE included on the other axis. A single speed gear set is used in the most cases and in combination with a double speed gearbox there is no clutch needed. Due to the similarities to the P4 concept no further investigations have been done on that.

3.4 PARTS OF THE TRANSFER CASE FOR P0 UNTIL P3

A current customer request has drawn the attention to this part. The transfer case uses a clutch for adjustment of the torque distribution between the front and rear axis.

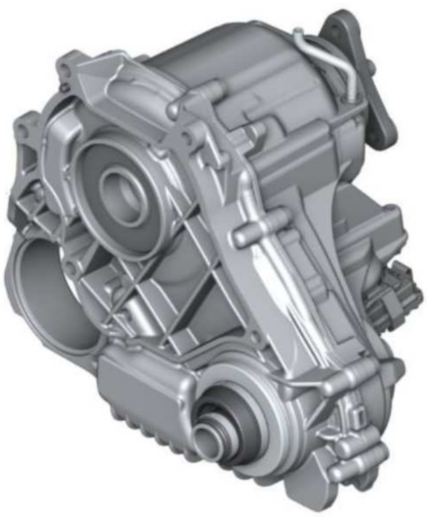

Figure 25: Transfer case [14]

A friction clutch is implementd in this transfer case. This clutch part is split with a plug in toothing into a sprocket and the outer disc carrier.

Figure 26: Inner and outer disc carrier ATC 450 [15]

The outer disc carrier is a stamped sheet metal part, but the sprocket requires small tolerances for the toothing of the tooth chain. The PM technology is suitable for the production of an outer toothing of the sprocket with high requirements on accuracy and surface hardness, which is required. In addition, the compact size fits good for the PM technology.

3.5 CHOSEN PART AND POSSIBILITIES FOR PM TECHNOLOGY IN THIS PART

The sprocket of the transfer case fits best and it is analyzed in detail. The outer diameter is about 95 mm and the inner one is about 50 mm. The projected area is about 51 cm^2. A 500 tons press is able to reach the needed compaction of the part even for the calibration process.
The total height is also less than the 60mm, hence it is possible to compact and sinter this component.

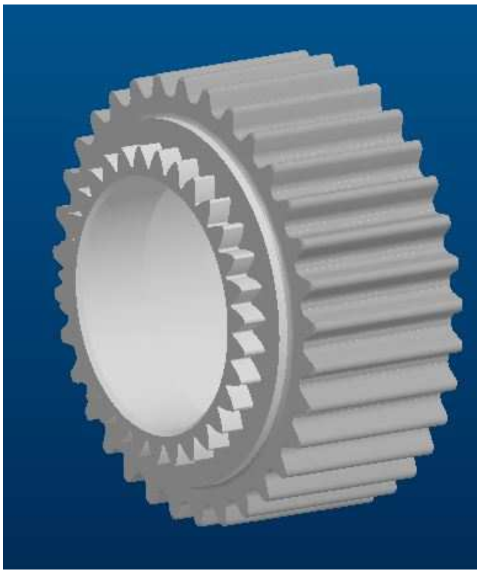

Figure 27: Chosen sprocket for further investigation

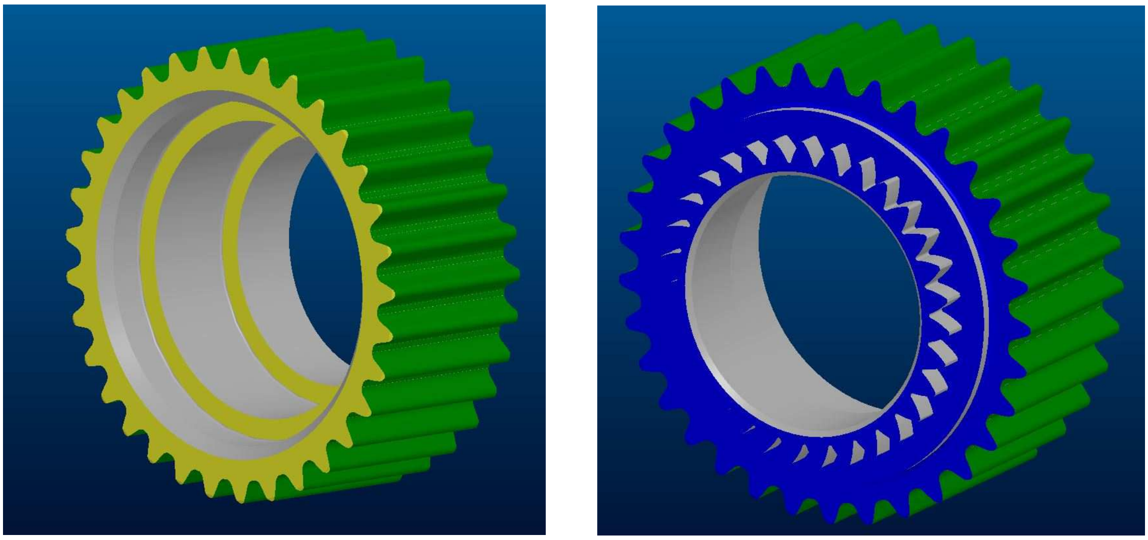

Figure 28: Sprocket with the highlighted plains shown from both sides

In Figure 28 shows the blue and yellow highlighted plains from both sides. Each of these plains need a separate punch for the compaction. The part exhibit in total three main planes from each side. The size of the planes is also big enough for punches.

The typical advantage of the sinter process is given as well, because the complex outer geometry of the sprocket and the geometry of the plug-in toothing are cost intensive in the production with conventional technologies, therefore it is the perfect part for the PM technology.

4 LOAD SCENARIO AND FINITE ELEMENT METHOD

4.1 FINITE ELEMENT ANALYSIS

A colleague has analyzed the part for given loads with a finite element analysis program. The result of the simulation shows that the component can withstand these loads for the given load spectrum.

5 Concept for the validation of the part

5.1 Validation of the parts

The concept of validation of the part is the major part of this Master Thesis. Several tests are used to prove the material's strength. The component should withstand a certain number of samples without relevant damages. Following three types of criteria describe the test of the part:

- Static load on the sprocket
- Dynamic load of the chain teeth's
- Testing of the interacting system in a chain drive

The first two test criteria can be tested with already existing testing machines, for the third methodology, the development of a new test bench is necessary.

5.2 Testing of the interacting system

The testing of the interacting system should deliver a statement about the fatigue strength and the types of damage in the interaction zone, like pitting. Therefore, the whole lifetime cycle has been shortened to a collective load of 1000 Nm over one million cycles. The system has to turn 1 million rotations and the sprocket should not show significant damages. Another criterion is the stationary chain speed which should be higher than 1500 rpm to reproduce the interaction between the chain links and the sprocket teeth's depending on the speed.

Figure 29: Symbol picture of a teeth chain drive [16]

The transmitted power of the chain drive is correlated to the momentum and the speed according to Equation 1:

$$P = M * \omega = \frac{n * M}{30} * \pi = 157\ kW$$

Equation 1: Initial power of the drive chain

The transferred power over the Sprocket is high and so it is the most important criteria for concept creation of a test bench.

5.2.1 The initial specification for the test bench concept

Subject	**amount**
Torsional moment	1000 Nm
Minimal stationary rotational speed	1500 rpm
Test duration	maximum 1 week
Oil temperature	maximum 100 °C
Pretension of the chain	determined by the center distance
Centre distance between sprockets	160 – 200 mm
Angular position	± 40 °

Table 5: Testing criteria

Additionally, factors like easy handling and exchange of testing parts are necessary for the test bench. Variability for different load situation and position differences should be covered as well for further testing.

A criterion which is also quite essential is the fact to have almost the same conditions as in real usage. These parameters should be equal for a chain drive:

- The deflection and angular position of the shaft
- The lubrication
- The tension of the chain
- The center distance
- The rotation direction

The chain drive is a part of a transfer case which is mounted after the gearbox to realize an AWD in cars. This test bench is used to get an outcome about the strength and the wear of the sprockets of the chain drive. The aim is to reach the most similar result than in real operation just without dynamics.

5.3 CONCEPTS FOR TEST BENCHES

There are several possible concepts to fit the load specifications, but all of them have to provide this load situation for the sprockets.

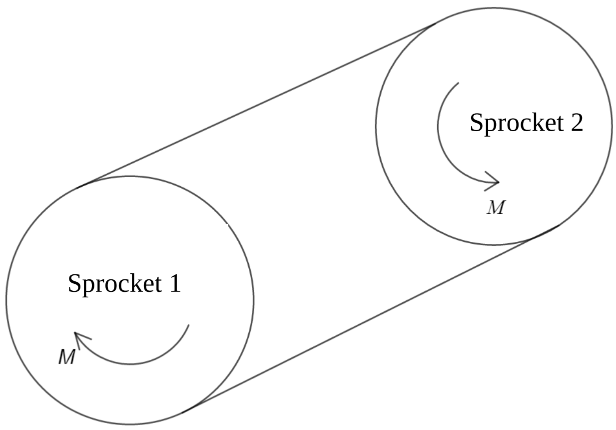

Figure 30: Load situation of the sprocket during a fatigue test

The following figure gives an overview of the possible types of test rigs, which could provide the specifications.

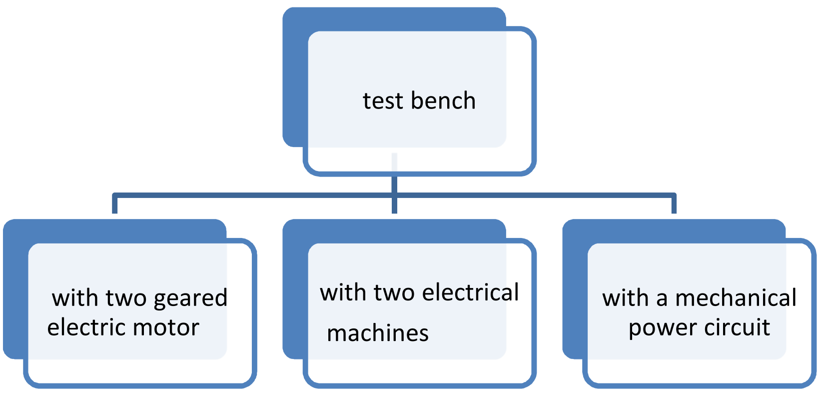

Figure 31: Possibilities for a test bench

5.3.1 Test bench with two geared electrical engines

Two electrical engines with gearboxes create the load for the system. In this case, one of these engines works as an engine and the other one as a generator.

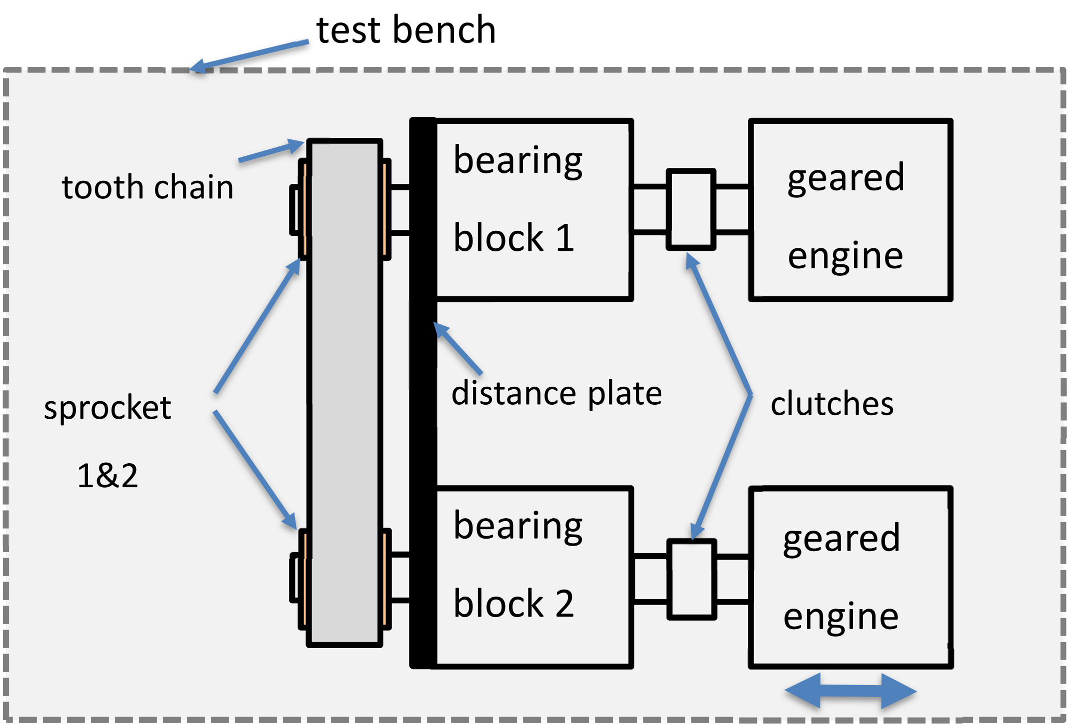

Figure 32: Concept 1 with geared electric machines

In Figure 32, two engines which are connected with flexible clutches to the bearing blocks are shown. The radial forces are transmitted with the tooth chain between the sprockets, and they are supported with the two bearing blocks and the distance plate in between. This distance plate offers an easy and stiff design and a force flux over it. This plate is responsible for the positioning of the bearing blocks. Variability of this concept is given through the exchange of the distance plate and the adaption of the sprocket holders.

The advantage of small, fast rotating engines with a high ratio of the gearbox is that the initial costs for the required infrastructure, the testbench and the electric supply are low.

This system meets the initial requirements, but further investigation has shown that a chain drive with a stationary rotational speed still have some dynamics in the chain and therefore the chain speed is relevant. We have set the minimum chain speed to 1500 rpm. This speed limit is not reachable with the geared engine, which are still small enough for this test bed.

5.3.2 Test bench with two electric machines- 2M concept

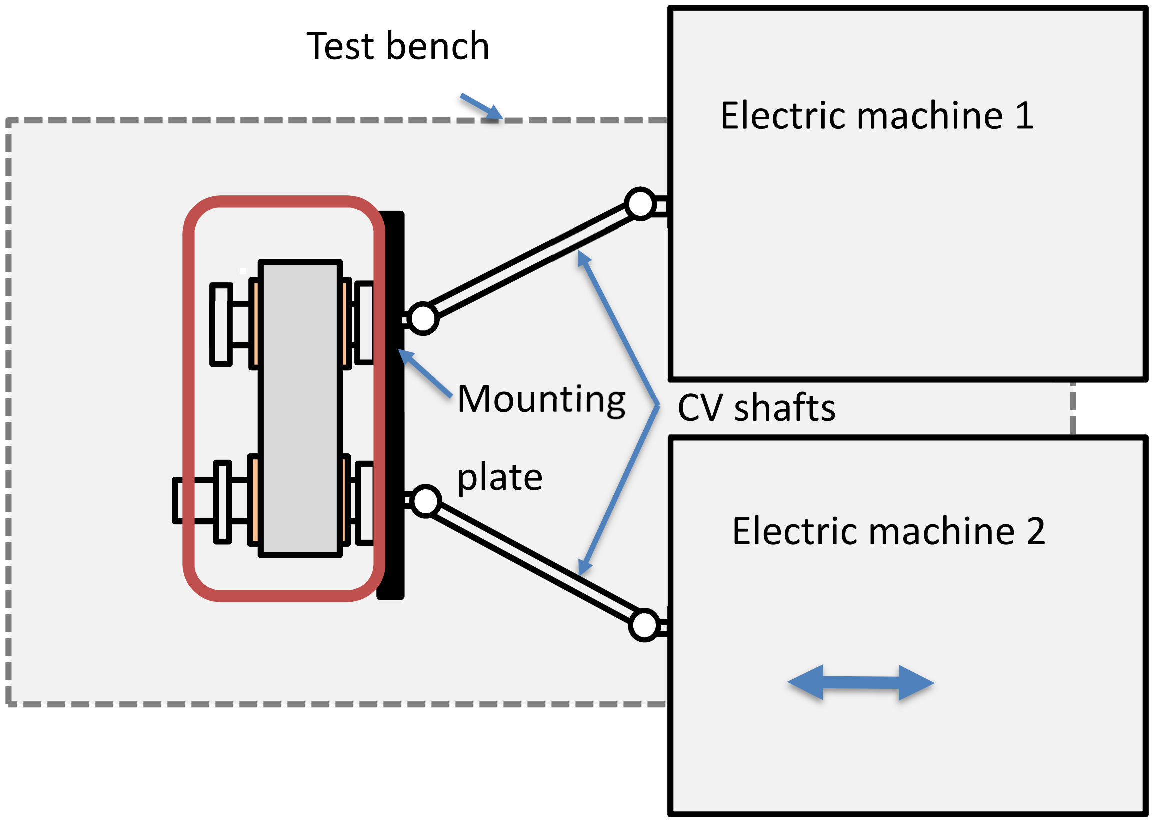

Figure 33: Concept 2 with two electric machines

This concept relies on two powerful engines with more than 157 kW continuous power. This system works on the principle that one of these machines works as generator and the other one as propelling machine. It provides a big advantage of possible dynamic testing as well. A further advantage is the fact that these engines can be used for different test benches and the electrical machines can be configured easily.

The significant disadvantages of these engines are the large dimensions as well as the high mass. The engines for this application have roughly 1,5 tons. The limited space where the test bench should be positioned has not the required foundation neither the required lifting equipment for these heavy machines. The economic aspect is to consider as well because every engine needs the power unit and an optional cooling periphery too. This concept has discarded without any request for prices and delivery time.

5.3.3 Torsion test rig

This concept relies on a test rig with a mechanical power circuit. It means that there are two shafts. One of them is a torsion spring, made of spring steel. The test rig is pretensioned in a circuit. The necessary torsional moment for the test is produced due to pretension. An electric engine speed up the system and the produced power at constant velocity is equal to the losses in the system due to friction. This principle is adopted from FZG gear test rigs which are common in gear testing. [17]

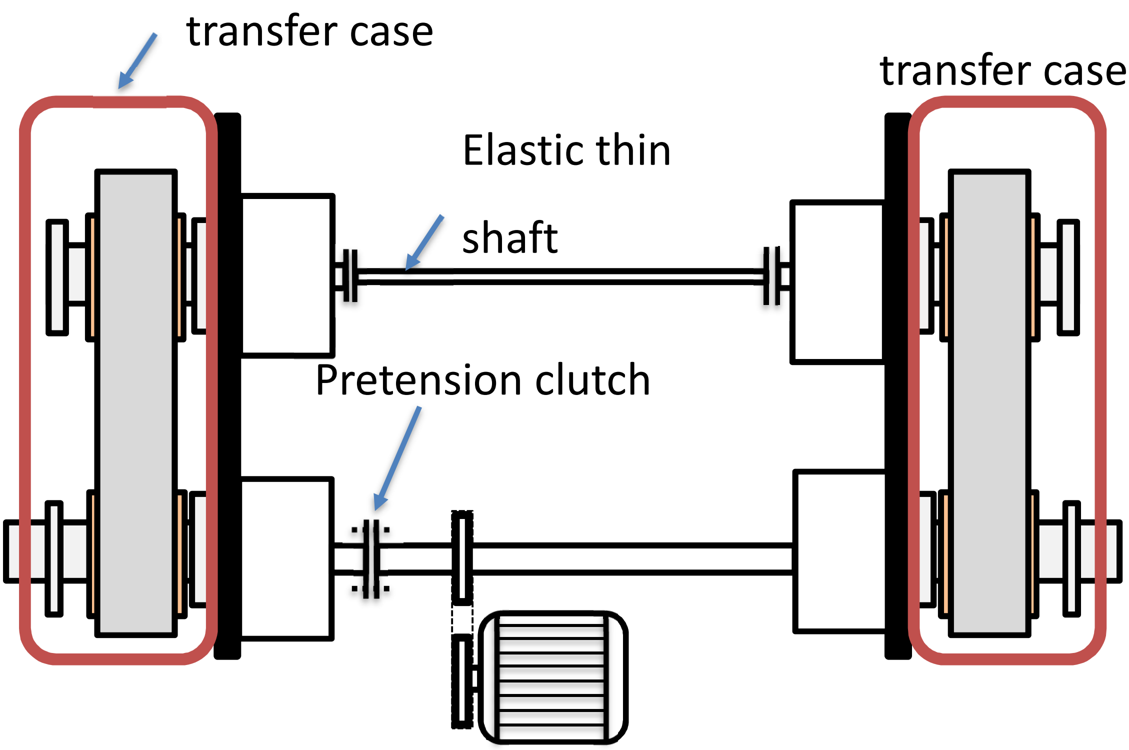

Figure 34:Test bench with transfer case at both sides

This system is similar to the original FZG test bench concept with two gears sets, but instead of gears, two chain drives are used. The advantages of the torsion test are the small needed electric power supply and the compact design size compared with the 2M-concept.

Additional aspects for drive chains

The mounting position of the chain drives should be like the original use in the transfer case. It means that there is the need for variability of the mounting angle and center distance of the test bench for different drive chains. The variability of chain drives in a horizontal and vertical direction is necessary.

An elongation of the chain occurs during its lifetime. This elongation reduces the pretension in the system.

Deflection, axial and angle errors of the two sprockets

The positioning of sprockets strongly influences the lifetime of a chain and the sprockets.

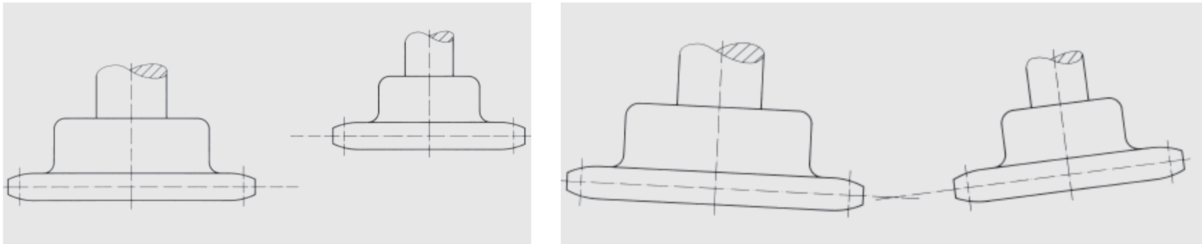

Figure 35: Errors in axial position (left) and angle (right) [18]

Errors in the angle or the axial position of the sprocket cause a non-ideal load distribution or additional force. The pull force occurs bending of the two sprockets. The effect depends on the layout of the bearing and sprocket combination.

In the case of the cantilevering shaft, there is an angle between both sprocket axes and an axle displacement. The other case the sprocket is on the axis in the middle of both bearings. There is no angle in between in the case of two symmetric shafts. A change of the center distance of both sprocket still occurs. Nevertheless, the situation differs always from the real operation situation, load, geometry and the material. [18]

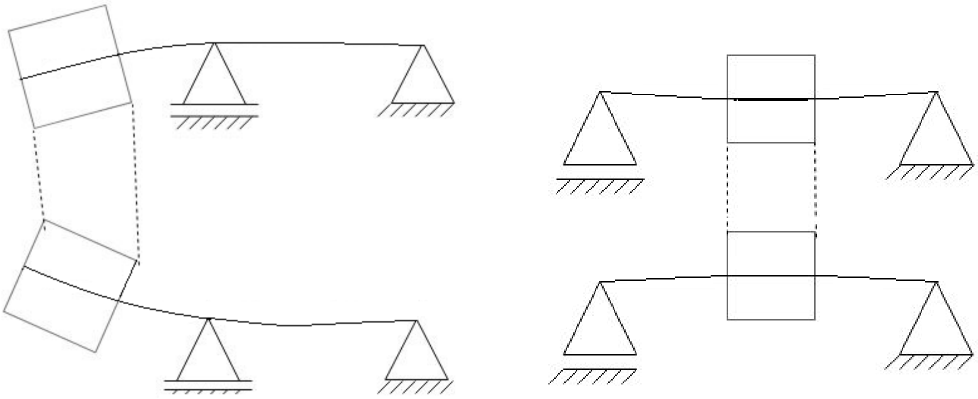

Figure 36: Problems with cantilevering shafts and optimal solution

The real situation is best tested in real housing with the same type of bearings. The real lubrication is ensured in the original housing as well. Therefore, the decision is made to test the sprocket inside the original transfer case.

6 TORSION TEST RIG

6.1 SPECIFICATIONS OF THE TEST BENCH:

The requirements of the test bench have set upwards to 2000 Nm and 3000 rpm. This test could be done for a big variety of car transfer cases, which are limited in the center distance and the angle.

	requirement
Maximum torsional moment	2000 Nm
Operation range	± 25 Nm around the nominal torque value
Range of rotational speed	0 - 3000 rpm
Minimal center distance	100 mm
Maximal center distance	300 mm
Upper angle of slope	-60 °
Lower angle of slope	+60 °
Pretension of the chain	Set due to center distance
Adaption of connection to the transfer case	All types of plug-in toothings
Lubrication of transfer case	Like in the transfer case
Maintenance interval	500 h
Measurement range of torque	± 2500 Nm
Accuracy of the torque measurement	± 5 Nm
Rotational speed measurement	± 3500 rpm

Table 6: Requirements for the test bench

Additional aspects for the torsion test rig are a stiff design, and the whole test bench should be mounted on a cast iron plate with the dimension of 1x2,5 meters. The mounting occurs over several T-groove. No flux of force should occur over this cast iron plate except from the static load due to gravity. Due to the high potential of the test bench the maximum rotational speed and the maximum torque are doubled. The maximum power in the mechanical power circuit calculated according to the following equation.

$$P = \varpi * M = \frac{2 * \pi * 3000}{60} * 2000 = 628.3\ kW$$

Equation 2: Mechanical power in the circuit

6.2 Detailed concept of the test bench

At the end of the Master Thesis, the concept of the test bench is fixed and the detailed concept of the construction is shown in the Figure 37.

Some FE-simulations of the shaft are in progress. Therefore, there is still change until the final design. The safety coverages are not designed or documented yet.

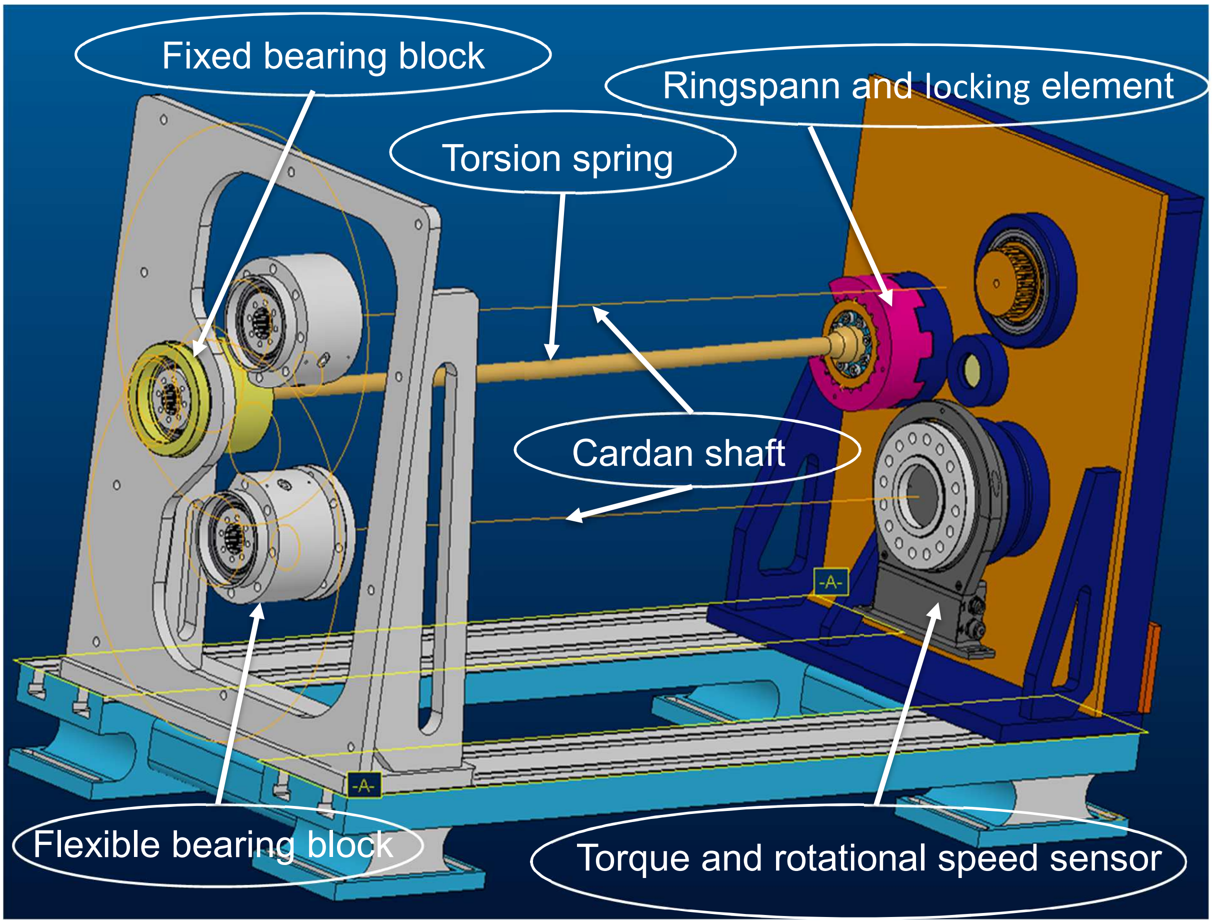

Figure 37: Screenshot of the construction at the end of the thesis

The Figure 37 shows the test bench which is divided into two main parts and they are connected with a torsional shaft and cardan shaft. The left side is made to mount the unit under test and on the right side there is a gearbox to close the mechanical power circuit.

The testing transfer case is to mount on the left side. An individual mounting plate is the adapter between the test bench and the different types of transfer cases. This plate implements the fixing holes for the test bench and the transfer case and it positions the two bearing blocks with drilling holes according to the center distance of the transfer case. The flexible bearing block is connected to a cardan shaft, which is just sketched in Figure 37. Further on, this side is called mounting side.

The flexibility is given through a mounting plate which is to manufacture individually according the test transfer case. The center distance and the angle can be changed with the exchange a mounting plate.

A gearbox for tensioning is shown on the right side of Figure 37. The task of this test bench is to transfer the pretensioned torque from the torsional shaft to the cardan shaft. Additionally, a HBM measurement system measure the torque in the system and the rotational speed. The pink highlighted pretension clutch is to fix the system during the pretensioning and to remove afterwards. The Ringspann clutch make it possible to open and close the mechanical power circuit. Further on, this side is called gearbox side.

During the master thesis all these parts are constructed, excluding the blue highlighted under construction in Figure 37, which is done by a college. For the initial test, the torsion test rig will be mounted on an existing cast iron test bed and the electric power supply will be realized with existing electric machines. The following chapter explain all the thoughts behind the construction.

6.3 ASPECTS TO CONSIDER IN TORSION TEST RIG

6.3.1 Elongation of the chain

The elongation over lifetime is maximum 0.5 % of the length, which is defined out of experience. The initial length is 641 mm.

The creation of the torsional moment in the power circuit relies on a pretensioned torsion bar. Big twisting of the long torsional spring is preferred because then the influence of elongation is minor. Therefore, the length is chosen with 1000 mm. The increasing length influences the variable φ.

$$M(\varphi) = \frac{\varphi * I_p * G}{l}$$

Equation 3: Torsional moment in dependency of the twisting angle φ

The elongation of the chain is known in total out, but the course of the curve until its final value is not known. Therefore, a linear curve is assumed. The real course of elongation will be known during the first tests.

The following figure shows the loss of the torsional moment with a shaft designed for 1000 Nm applications with two chains drives. In this case, the shaft is twisted 30.85 ° in total. The change of the twisting angle caused by the elongation is 7.85 °.

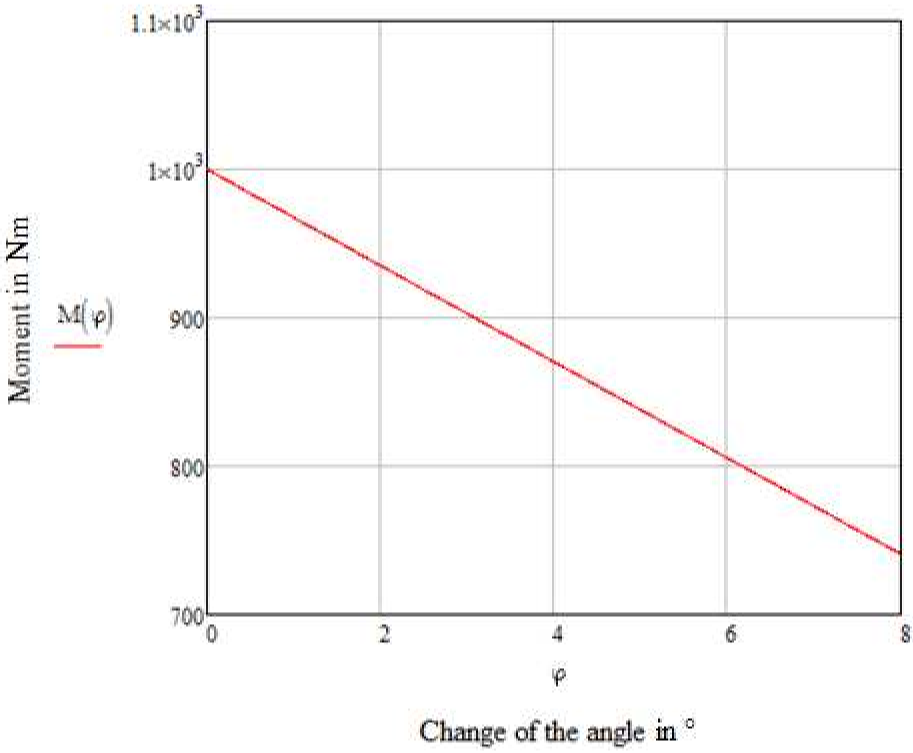

Figure 38: Loss of the torsional moment in the mechanical power circuit in case of two chain drives (1000 Nm)

Figure 38 shows that the torsional moment can't be held on the same level. A retension of the system is needed to keep the torque in a certain range.

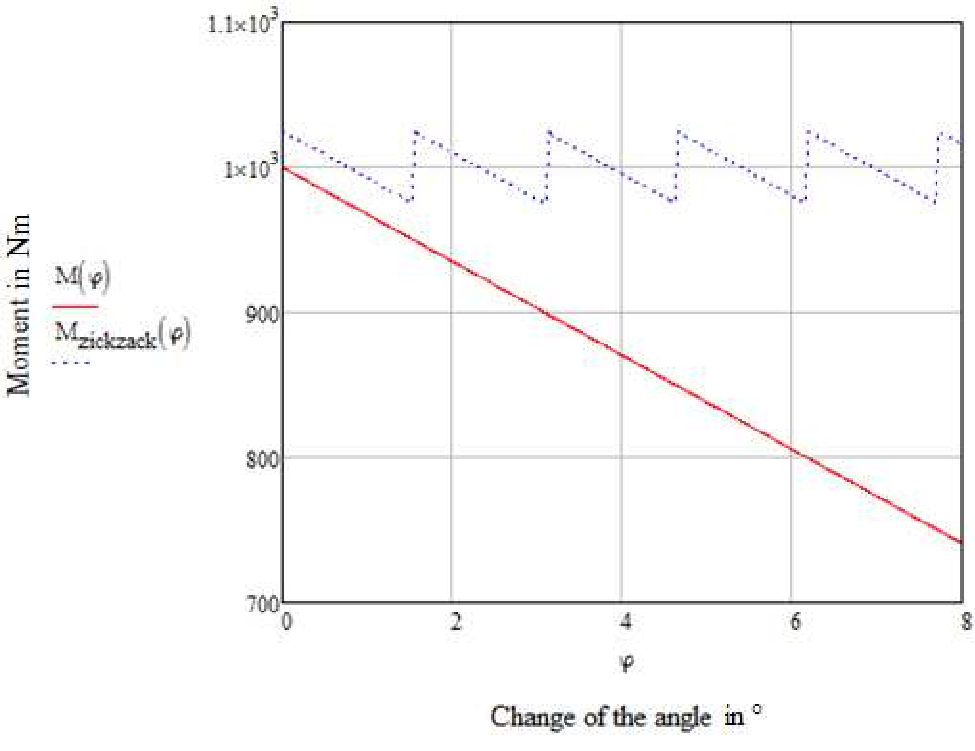

Figure 39:Retension of the mechanical power circuit

The Zick-Zack line shows the torsional moment loss and the reestablishing of the pretension in the system. This process of reestablishing the pretension in the system is quite time intensive, which has to done 5 times in the test shown in Figure 39. Probably this retension could take 20 minutes twice a day.

An exchange of the second drive chain against a gearbox reduces the elongation by almost the half, because the gears have several times less wear than a tooth chain. The change of loss of twisting is now just 3.92 ° in total. The operation area of the test bench until a retension is set to ± 25 Nm of the nominal value.

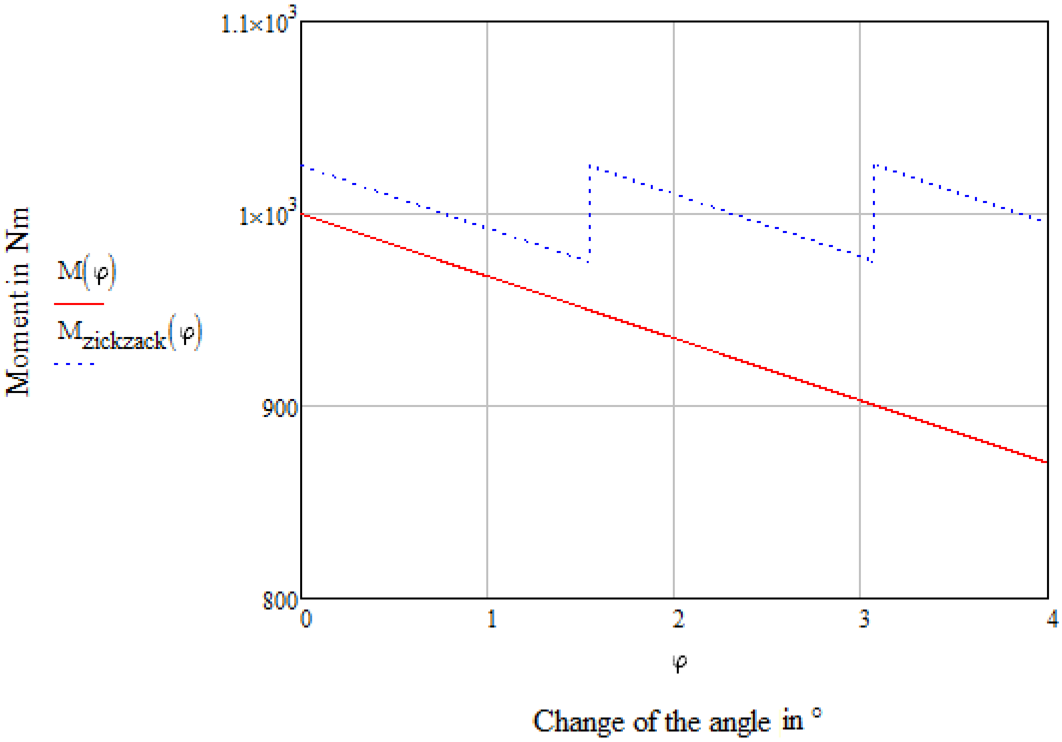

Figure 40: Loss of the moment with just one chain drives

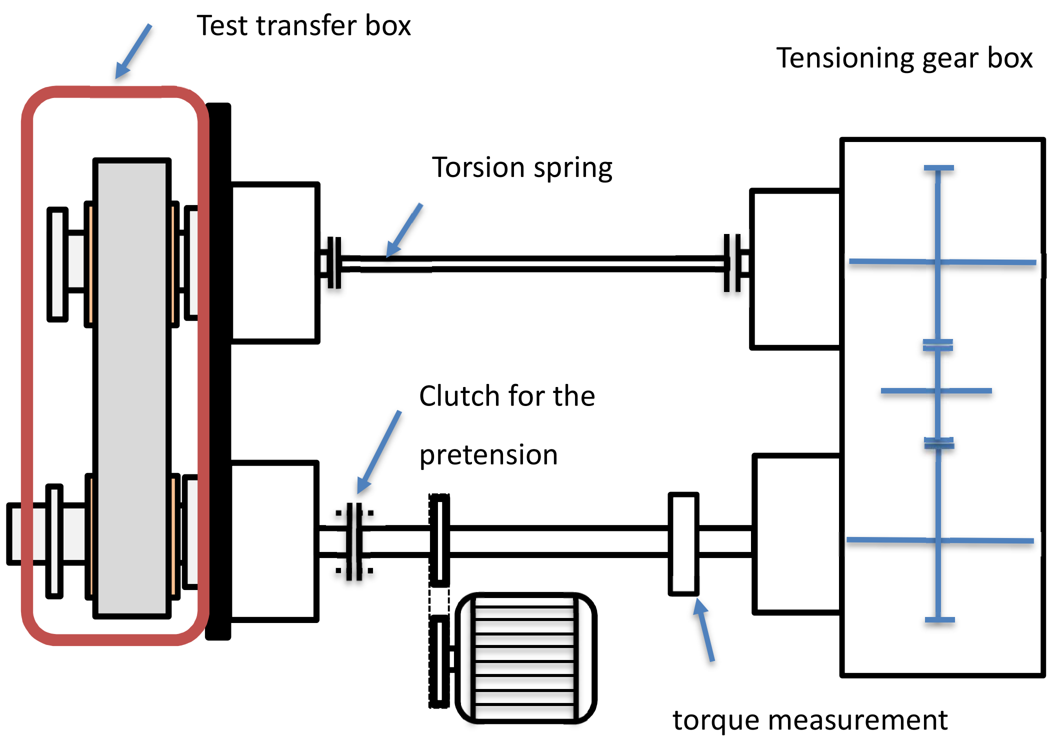

Figure 41: Concept with a gearbox on one side

Figure 41 shows a modification of the typical FZG tension test bench. It shows on the left side the test transfer box and on the right side a gearbox, which is responsible for transferring the pretension forces.

The difference between a chain drive and gear stage is the varying output rotational direction of them. The gearbox has three gears inside to create the same rotational direction. Additionally, a ratio between the shaft is not allowed. Otherwise, the system is self-blocking.

This concept has a torque measuring flange, which has the task to measure the torque and interrupt the test cycle if the torque falls below a limit. The second task of it is the measuring of the initial tension.

6.3.2 Different torsional moment classes

The possibility of mounting different transfer cases or gearboxes needs a variability of torque. Not all stiffnesses of the other components like the cardan shaft, the gears, the measurement flange and the shaft hub connection are known and they are usually several times stiffer than the torsional spring. Therefore, it is assumed, that the biggest twisting will occur in the torsional shaft and the other twistings of the components are neglectable.

The elongation of the chain requires torsional shafts relating to the applied torque. The twisting angle due to pretension has to be high, that the influence of the chain is minimal. The polar moment of inertia, the torque, the shear modulus and the length of the shaft are the influence parameter. Different diameters of the shaft result in different polar moments for the varying torque classes.

$$I_p = \frac{(\frac{D}{2})^4}{2} * \pi$$

Equation 4: Polar moment of inertia

The expected use in the future requires three shafts according to the following table.

	Torsional moment	**Diameter**	**Length of the plug-in toothing**
Shaft 1	1000 Nm	22 mm	25 mm
Shaft 2	1500 Nm	25 mm	30 mm
Shaft 3	2000 Nm	28 mm	35 mm

Table 7: Shafts for different torsional moment classes

The calculation of the shafts bases on the only external load, the torque and is done for the material 51V4, which is a spring steel.

The stress, caused by this torque, is compared with the permitted pulsating torsional stress of the component. The safety between calculated torque and permitted pulsating torque is for all three types of shafts bigger than 1.15.

The plug-in toothing has been calculated against pressure. The safety of the plug-in toothings is at least 1.45 for all three types of shafts.

6.3.3 Exchange and pretension of the torsional shafts

The torsional shafts with different diameters are made with the same connection parameters. On one side there is a clamping element of the brand “Ringspann” which is designed for up to 3400 Nm. There is a plug-in gear to connect the torsional shaft with a bearing pedestal on the other end of the shaft. The length of the plug-in toothing depends on the torque class of the torsional shaft.

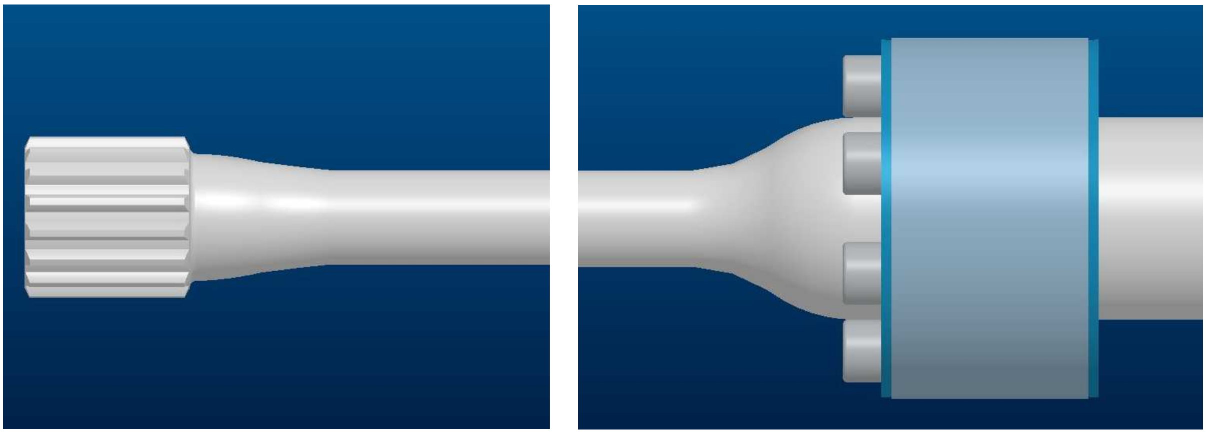

Table 8:Connecting ends of the torsional shaft; left side shows the plug in toothing and right side shows a cylindrical clamping element

The pretension of the system works on the principle that the tensioning gearbox gets locked and the screws of the clamping element get opened. One option is that the operator plugs the leaver with the right adapter on the shaft and apply a load on the leaver according to the required pretensioned torsional moment. The pretension can be measured by the output of the torque measurement system and if the right pretension is reached the operator fix the screws of the “Ringspann” element.

The maximal pretension torque of 2000 Nm would require a long leaver or high forces on the leaver, but the system has to be pretensioned with a tool which is easy to handle and manipulate. Therefore, another option with a mechanical torque multiplier is more interesting. The testbench is adapted to use this pretension system, which functionality base upon a planet gear set. The torque multiplier is supported on the red highlighted part of the housing.

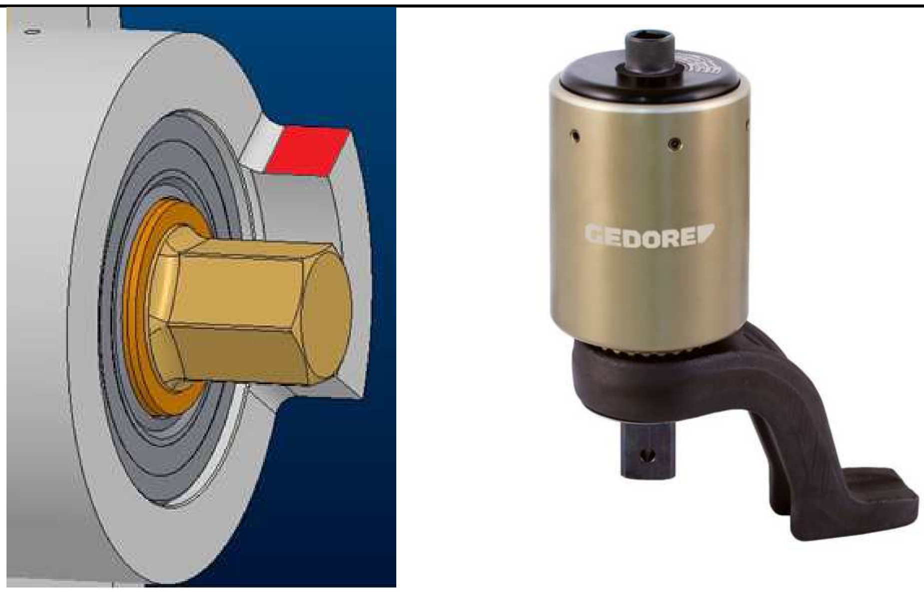

Figure 42: Torsional shaft with hexagon and housing (left) and torque multiplier (right) [19]

Another aspect for this concept is a higher safety of the pretension process, due to about 16 times smaller loads on the leaver.

6.3.4 Measurement of systems torque and speed

The significant two main parameters, which should be recorded, are the torque level and the number of revolutions. The nominal torque is 2 kNm, but due to the interaction of tooth's and chain, there is a dynamic in the torque value. Natural frequencies occur in the run up. Hence the choice of the nominal torque of the measurement system is above 2 kNm.

The second criteria of the torque measurement is the accuracy of the system. Due to the operation within the ±25 Nm range around the nominal value, the system requires an accurate measurement system.

An HBM TB40 3 kNm torque measurement flange is the object of interest. The output of the system is a ±10 V or a frequency output. Cause by an additional analog-digital converter, the accuracy of the system depends on the type of the output signal.

The cardan shaft produces a parasite bending moment on the shaft, which is depending on the angular position, the deflection angle and the load of the cardan shaft. This additional pulsating bending moment will reach twice the maximum (up to 360 Nm) and zero. It can be considered for the analysis and the initial pretension. The calculation of the accuracy is done according to the HBM guide and added in the appendix. [20]

parasite load (min/max)	**Signal type**	**Accuracy in Nm**	**Accuracy in percent of max. torque**
max	±10 V output	±9.12	±0.46 %
max	Frequency output	±7.37	±0.37 %
min	±10 V output	±6.59	±0.33 %
min	Frequency output	±3.78	±0.19 %

Table 9: Accuracy of the different output variants and parasite loads (95.5 % probability)

The check if the pretensioned torque falls during the test has to be always done at the same position of the cardan shaft.

The rotational speed is also measured with the HBM sensor, where an incremental rotary encoder is integrated.

6.3.5 Gearbox for the pretension of the system

The task of the gearbox is to transfer the torque of gear 1 to gear 2 without any change of the rotation direction and any ratio. Three gears are needed because with every interaction of the gears, the direction changes once.

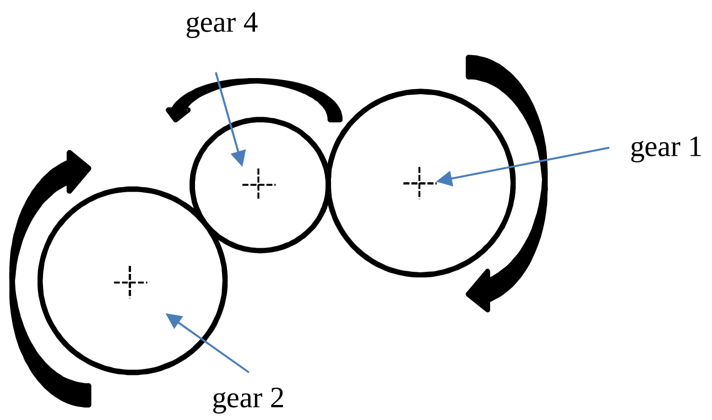

Figure 43: The constellation of the shafts

Both, the drive (gear1) and output gear (gear 2) need to have the same size and in between another gear is mounted, which is responsible for the change to the right rotation direction.

The gears are normal involute gears.

6.3.5.1 Reasons for two shafts outputs

According to reach higher variability of the system, two output shafts are implemented. Always one of this output shafts is connected to the joint shaft. Either gear 2 or gear 3 is connected to the output.

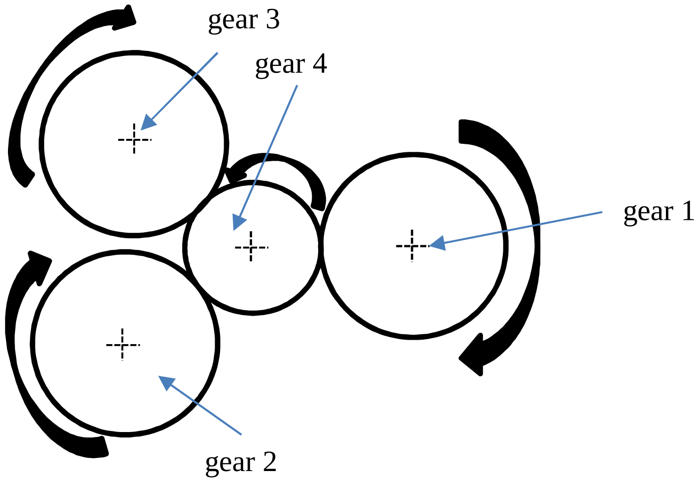

Figure 44: Constellation with two shafts

The main reason for the two shafts is the fact that two types of transfer cases are available - one with the output to the front axis on the right side and one on the left side.

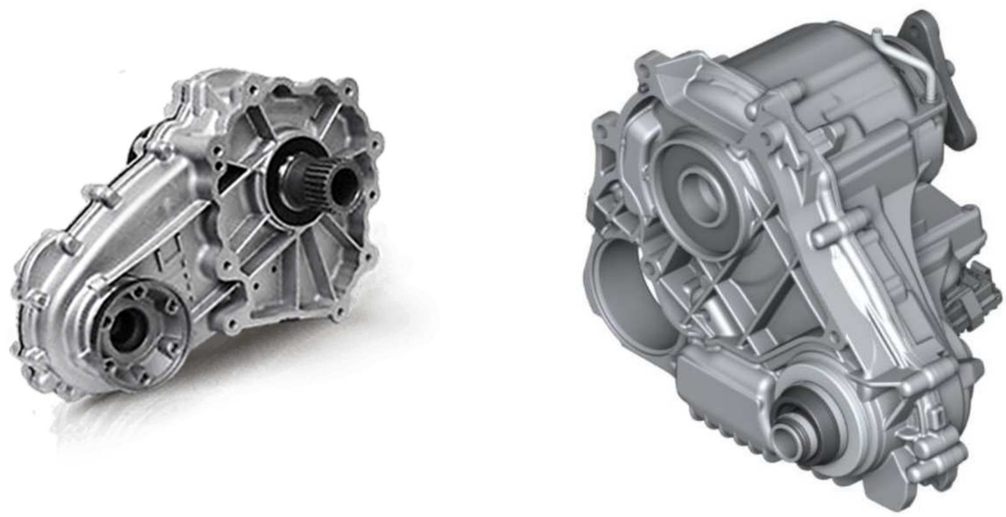

Figure 45: right sided output [21] and left sided output [14]

6.3.6 Calculation of the gears

The initial calculation is done analytically, and the fatigue strength of the gears is calculated. Spur toothed gear sets are used to reduce the forces of the bearings caused by the high transferred torque.

The constellation of Figure 44 is calculated and the result of the calculation shows that four gears are necessary, but the gear 1, gear 2 and gear 3 are equal. Gear 4 is mounted just one time in the tension gearbox. The specifications of the gears are shown in Table 10.

	Gear 1/2/3	**Gear 4**
Number of teeth's	37	31
Module	4 mm	4mm
Reference diameter	148 mm	124mm
Width	65 mm	68mm
Reference pressure angle	20 °	20°
Helix angle	0 °	0 °
Tip diameter	156 mm	132 mm
Root diameter	138 mm	114 mm
Base circle diameter	139.08 mm	116.52 mm
Material	16MnCr5 case-hardened	16MnCr5 case-hardened

Table 10: Data of the gears

This calculation is an analytical calculation based on the interaction of two gears. A calculation with gear calculation software should be the next step to verify these dimensions and the aim is to optimize the toothing. The lubrication happens with a circulation pressure lubrication.

The calculation of the gears shows the safety of the smaller gear against tooth break and pitting, which is in relation to the maximum permitted Hertzian pressure. The safety against the tooth break is 2.02. The safety against pitting is 1.22.

6.3.7 Variability in positioning with joint shaft

Variability between the possible mounted gearboxes is needed. It can be achieved by a cardan or constant velocity shaft. The mounting of the joint shaft offers a bandwidth to operate with a gearbox. The deflection angle of maximal 10 ° is the limiting factor in the variability. A higher deflection angle increases the variability but it limits the rotational speed and it occurs a shorter life time of the shaft.

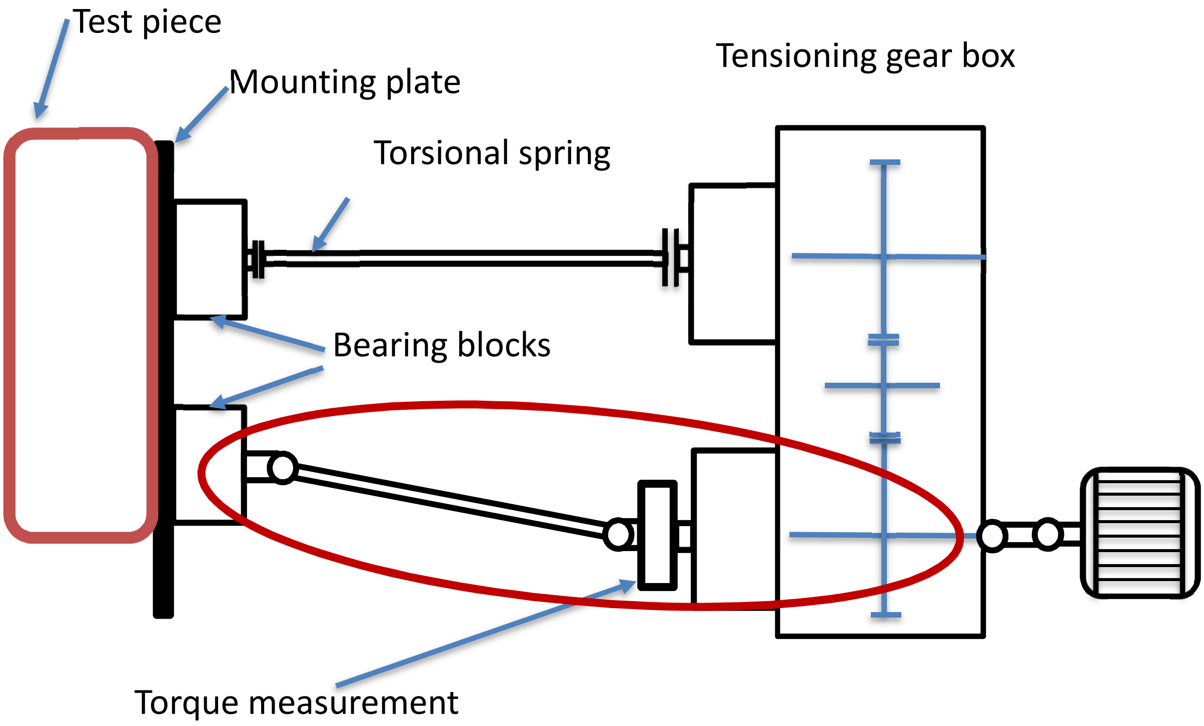

Figure 46: Concept with a cardan shaft

6.3.7.1 Possible areas and speeds with the joint shaft

The variability due to a joint shaft is shown in Figure 47.

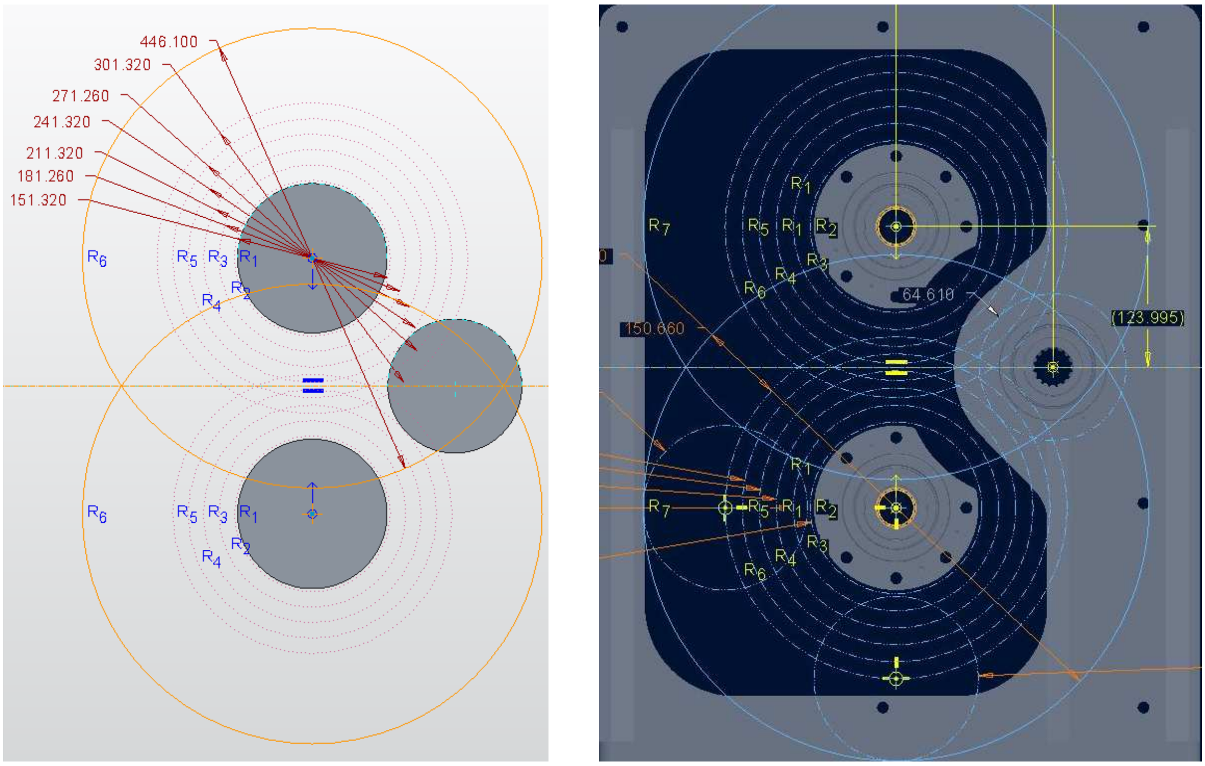

Figure 47: Reached positions with a joint shaft (left →sketch, right→constructed)

The dotted circle lines show the different degrees of the deflection angles starting from the inner circle with 5° and rises by one degree to the outside.

The allowed rotational speed under a certain angle is limited through a temperature limit of the joint. This empirical equilibrium is applicable for a constant velocity shaft.

$$n * \beta <= 18000$$

Equation 5: Empirical equation for the temperature limit of constant velocity joints [22]

The other opportunity is a cardan shaft of Elbe with the joint size 0.113 (declaration of Elbe). The maximum speed for certain deflection angles results out of Figure 48.

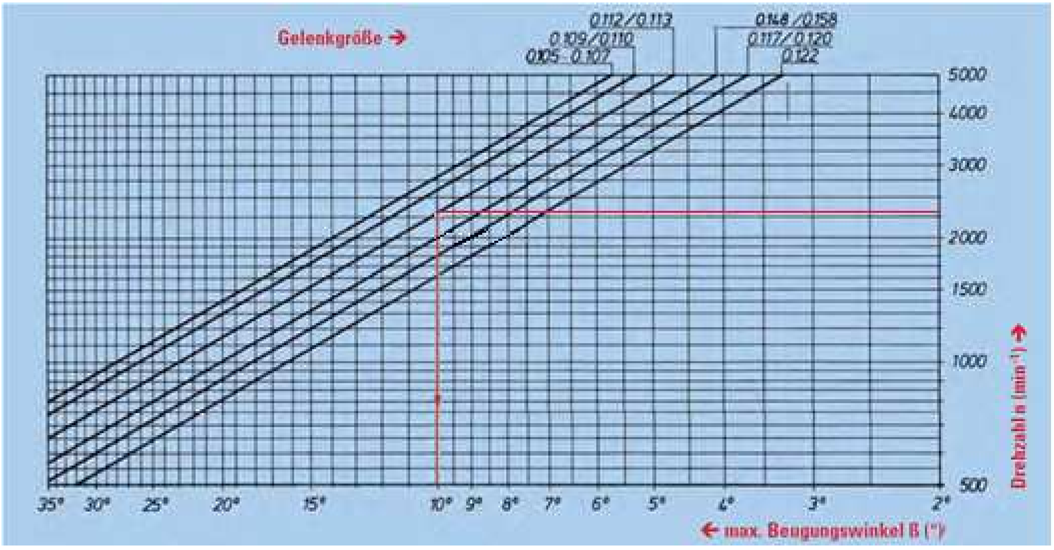

Figure 48: Correlation of deflection angle (10°) and speed for Elbe cardan shaft with a joint size 0.113 [22]

deflection angle	*Cardan shaft speed (0.113 joint size)*	*Constant velocity shaft*
5°	4600 rpm	3600 rpm
6°	3750 rpm	3000 rpm
7°	3250 rpm	2550 rpm
8°	2800 rpm	2250 rpm
9°	2550 rpm	2000 rpm
10°	2300 rpm	1800 rpm

Table 11: Interrelations of speed and deflection angle for both types of shafts

According to Table 11 the cardan shaft offers higher possible speeds than a CV-shaft and in addition the cardan shaft is cheaper as well. A criterium of the construction is also the better fitting flange. The only disadvantage of a cardan shaft is the need of a driven and an output shaft, which have to be parallel, like in the torque test rig.

The critical aspect of the use of a joint shaft is the maintenance interval. The highest torque, maximum speed in combination with the highest deflection angle damage the joint shaft before the next maintenance interval after 500 tests. Therefore, a check of the shaft is needed every 30 test runs or 500h, that depends on which occurs earlier. Usually, this worst case doesn't happen every test cycle. Therefore, the cardan shaft should survive the time until the next maintenance interval.

6.3.7.2 Individual mounting plate for different transfer cases

The transfer case is usually mounted on an existing gearbox of the car, and this connection of housing and the connection of the shafts differs between different OEM's.
The test bench is made to be adaptable to the different types without any issues. An individual mounting plate is to design for every transfer case because the screw pattern and the position of the alignment pins changes. The different lengths of the input shaft and the diverse plug-in toothings will be realized with an adapter which is mounted in the bearing blocks. The only criteria are the existing toothing of the bearing blocks and the position of the threads which are needed for fixation of the adapter.

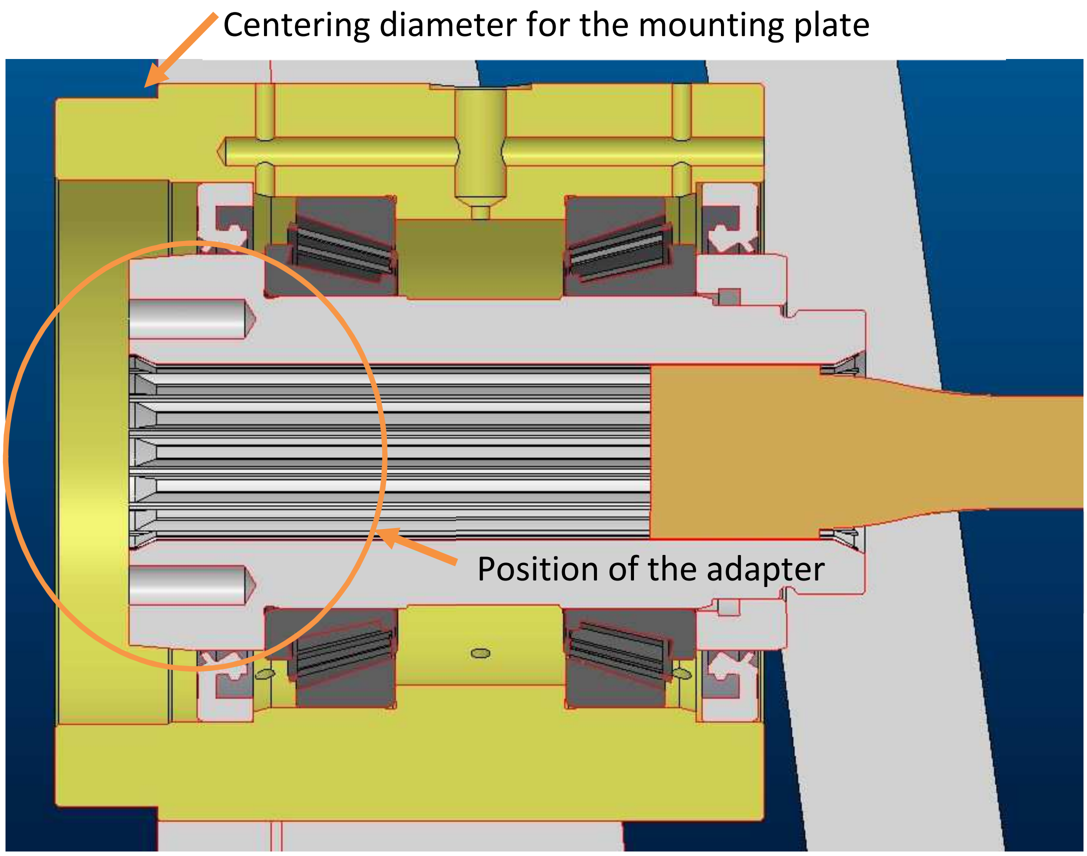

Figure 49: Bearing blocks for variable position and variable adapters

The mounting plate has the additional task of positioning the bearing blocks and the force transfer between them.

6.3.7.3 Calculation of the friction in the system for dimensioning of the electric machine

The whole system is powered with an electric machine which is responsible for creating the power which gets lost through the friction in the system. These maximal losses are the dimension factor of the electric machine.

The efficiency of the transfer case is assumed with less than 3 % and the losses of the gear contacts have been calculated. The bearing and seal losses are the result of SKF experience values. The values in Table 12 refer to the maximal losses at a speed of 3000 rpm and a torque of 2000 Nm.

Component	*Losses*
Transfer case	<3 %
Gear contact (3x)	3.3 %
bearings	~1.2 %
seals	~0.3 %

Table 12: Maximal losses in the power circuit [23]

The sum of the losses is approximately 8.1 % of the power. This value is created from reference value for bearings and seals, therefore a slightly higher-powered engine is recommended.

$$P_{out} = (100\% - \eta_m) * P = (100\% - 91.9\ \%) * 628.3 = 50.89\ \text{kW}$$

Equation 6: Output power of the electric machine

The critical side fact for electric engines is, that this power output has to be a long-term power amount, which is the so-called S1 mode. Additionally, the power unit has to supply this power. An important side fact is also the transformation of this power in heat. The engine is cooled on a water circuit or with air cooling, and the room has a ventilation system. A big amount of the power also will be produced in bearings and the sealing, which afford a lubrication cooling circuit to transfer the heat away.

6.3.7.4 Natural frequencies

The knowledge about the eigenfrequencies gives the information about the rotational speeds, which are restricted for permanent operation and slow passing of them. Changing or modification of the components occurs a change of the natural frequencies.

The critical point for analysis with an multi body simulation tools are the unknown stiffnesses of components and connections, like the clamping element and all the plug-in toothings. Additionally, every change of the transfer case creates a new situation. The solution is a fast speed up until the maximum speed and then a slow down until a standstill. The natural frequency can be found out at the output of the torque measurement flange. An acceleration measurement next to the bearing positions provides information as well.

6.3.7.5 lubrication of the system and lubrication power unit

The gearbox, in detail the gears need some lubrication to resist the contact forces without damage. Damages of the gears could be pitting, wear, fretting or tooth break. For them, an immersion bath lubrication would be sufficient, but all the bearing blocks won’t be affected by this kind of lubrication.

One reason for circulation pressure lubrication is a lubrication of the gears before the start.

The second reason is that several bearing blocks are implemented where taper roller bearings are in use, which have a higher heat production than common ball bearings. The oil should transfer the heat into the tank of circulation pump to the surrounding room. The high loads in the gearbox require a special gearbox oil with a viscosity of 320 mm²/s. A phi oil HC Gold 320 is chosen to fulfill this task.

6.3.8 Consideration to use the test bench for gear sets

The torque test bench is developed for testing transfer cases, but instead of the transfer case, a self-constructed gearbox could be mounted as well. Using this test rig for gears as well has been considered during the whole conception period and construction process. A special mounting box for gears has to be designed. This option is held free for this use, but it is neither calculated nor constructed in the course of this thesis.

6.3.9 Proposals for the further implementations at the test bench

Safety concerns

The operator's safety is the highest aim. Therefore, a housing of the test bench will be necessary to cover all the rotating parts. A housing made of metal sheet could be mounted on the test bench. The width of both, the tensioning gear box and the assembly frame, is equal. Additionally, mounting threads are planned on them to fix the metal sheet easily.
This metal sheet is to remove for the exchange of the transfer case and the mounting plates.

Covering of the torsion shaft

The critical part according to the strength of the part is the torsional shaft. If an overload of torque occurs, the torsional shaft breaks at first. This shaft is pretensioned and rotated with 3000 rpm. If the shaft bursts, a cover prevents the surrounding from flying shaft parts. Therefore, an additional protection is needed in the form of a separable metal pipe.

Noise reduction

The created noise due to the tooth meshing of the straight toothed gearwheels creates a noise of unknown value so far. The reduction of the expected noise is the aim of the final dimensioning of the gears. A high-profile overlap is preferred. Sound isolation mats could reduce appearing noises. Stiffener at big surfaces can help as well to reduce the creation of sound.

Voith Kuesel clutches

The principle of the torsional shaft is long proven for FZG test benches. This principle has been adopted to our test application, and there are unknown factors like the eigenfrequencies. There is the option to use of a Voith Kuesel element instead of the torsional shaft or in addition to the torsional shaft. This standard element could be pretensioned as well. There is an elastomer implemented, which is highly elastic. The low stiffness of this element reduces the initial natural frequencies and it leads to an overcritical operation of the test bench. The creeping of the elastomer has to be almost equal to zero otherwise the pretension of the system gets reduced. [24]

6.4 PLANNED TESTING PROCEDURE

The procedure to test a new sprocket in a transfer case should be executed like the following steps:

6.4.1 Preparation for the mounting:

A custom mounting plate for the new transfer case has to be constructed and manufactured for the individual transfer case. The holes and positioning boreholes have to be adapted. The mounting plate implements the alignments for the positioning of the fixed and variable bearing block. The alignments are manufactured according the center distance and the mounting angle of the transfer case. Besides, the input and output shafts have to be made according to the input toothings and toothings lengths.

6.4.2 Mounting of the mounting plate, adapters and transfer case

The mounting plate is to fix on the testbench. The flexible bearing block is to position on the mounting plate according the alignment. The shaft adapters are screwed onto the shafts of the bearing blocks and afterwards, the transfer box has to be mounted on the plate. The torsional shaft is to choose according to the test torque.

6.4.3 Initial run until the first re-tension steps

1. Pretension the torque shaft with the nominal torque plus 25 Nm.
2. Fix the shaft with the “Ringspann” element.
3. Check the pretension with the signal for the HBM sensor and check if all housings are closed.
4. Run up the test bench until the maximum speed and slow down to zero again. Check the eigenfrequencies in the signal of the torque.
5. Choose an operation rotational speed in the range of 1500 – 3000 rpm outside and away from the natural frequencies.
6. Once the test is started, it runs until the HBM sensor detects a decrease of 25 Nm from the nominal torque or the final number of rotations is reached.

6.4.4 Repetitive steps until the final number of rotations

Step 1 until step 3 deal with the tensioning of the system and they are equal to the initial run.

4. Choose an operation rotational speed like before the test run and start the test.
5. Once the test is started, it runs until the HBM sensor detect a decrease of 25 Nm from the nominal torque or the final number of rotations is reached.
6. Repeat until the final number of rotations.

6.4.5 Immediate stop of the test bench

The criterions to stop the test bench immediately (without consideration of safety criterions):

- The immediate decrease of the torque to zero indicates a breakage.
- The fast decrease of the specific torque per minute (torque gradient) shows damage or wear. The test component has to be examined.
- The increase of the needed power of the electric machine above a limit to drive the test bench indicates a damage of a bearing or fault of any parts.

The values for the stopping of the test cycle like the decrease of the torque and the increase of the input power, have to be determined experimentally during the first test cycles.

7 FINAL SUMMARY OF THE MASTER THESIS

This master thesis has shown up, that parts of clutches of the electrified power trains can be made with powder metallurgy manufacturing. The investigation of the different types of electrified drives shown up several parts, which could be future PM parts, even though, some parts are not advisable to produce right now with PM-technology because of a problem with the material strength or quantity. Manufacturing specific system design makes a production in sinter technology interesting again.

A part of the outer disc carrier is a sprocket, which has been chosen as component able to manufacture with sinter technology. This part is a typical part, which suits best for the sintering technology.

Finally, a testing concept for this sprocket has been developed and the outcome of this thesis is the concept for the tension test rig, which fulfills the criterions to test the fatigue strength and to get an indication about the wear and pitting of the sprocket.

The innovative testing concept, which bases on a proven concept of FZG-gear testing, makes a fatigue test of transfer cases possible. The wide use of the test bench for various transfer cases will make an inhouse endurance test of sprockets possible.

The test bench fits in the in- house facilities and can be initially operated with the existing infrastructure.

The Master Thesis and the outcoming concept is the basis for the continuative work of colleagues, which also cares about the mounting and initial operation.

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9 TABLE OF FIGURES

10 ATTACHMENTS

A.1. Attachment 1: Calculation of the shaft

Verspannungsprüfstand Rechnung mit einem Kettentrieb

Torsionsstab Daten

$D_{Stab} := 22mm$

$l_{Stab} := 1000mm$

$\nu_{Stahl} := 0.3$

$$E_{Stab} := 210000\frac{N}{mm^2}$$

$$G_{Stab} := \frac{E_{Stab}}{2(1+\nu_{Stahl})} = 8.077 \times 10^4 \cdot \frac{N}{mm^2}$$

Kettentrieb Angaben

$A := 173.54mm$

$d_{Kopfkreis} := 94mm$

Aufgebrachtes Moment

$M_{soll} := 1000N \cdot m$

Torsionsträgheitsmoment

$$I_t := \frac{\left(\frac{D_{Stab}}{2}\right)^4}{2} \cdot \pi = 2.3 \times 10^4 \cdot mm^4$$

Torsions- Widerstandsmoment

$$W_t := \pi \cdot \frac{D_{Stab}^{\ 3}}{16} = 2.091 \times 10^3 \cdot mm^3$$

maxiamle Torsionsspannung

$$\tau_{tmax} := \frac{M_{soll}}{W_t} = 478.302 \cdot \frac{N}{mm^2}$$

Daten Federstahl51V4
tau_zulässig_schwellend=585N/mm^2

Vorspannung des Torsionsstabes

$$\alpha := \frac{M_{soll} \cdot l_{Stab}}{G_{Stab} \cdot I_t} = 0.538 \cdot rad$$

$$\alpha = 30.845 \cdot °$$

Torsionale Federkonstante

$$c_t := \frac{M_{soll}}{\alpha} = 1.858 \times 10^3 \cdot \frac{N \cdot m}{rad}$$

Länge der Kette

$$l_{Kette} := 2 \cdot A + d_{Kopfkreis} \cdot \pi = 642.39 \cdot mm$$

Längung der Kette: Annahme 0,5%

$$delta\_l_{Kette} := l_{Kette} \cdot 0.005 = 3.212 \cdot mm$$

Verdrehung des Torsionsstab aufgrund der Kettenlängung

$$\alpha_{Längung} := \frac{delta\_l_{Kette}}{\left(d_{Kopfkreis}\right) \cdot \pi} \cdot 360° = 0.068$$

$$\alpha_{Längung} = 3.916 \cdot °$$

Verhältnis des Moments zur Längung

$$M_{Abweichung}(\varphi) := \frac{\frac{\varphi}{180} \cdot \pi}{l_{Stab}} \cdot G_{Stab} \cdot I_t$$

Verlauf des Moments in der Torsionswelle

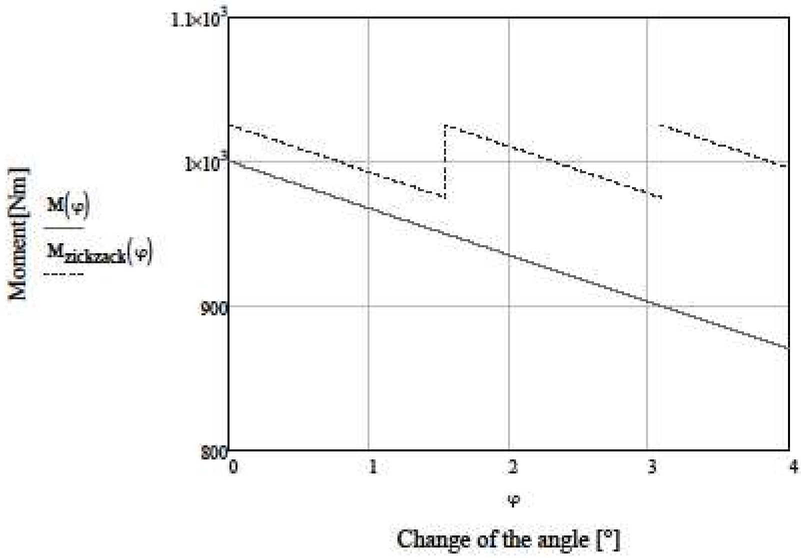
$$M(\varphi) := M_{soll} - M_{Abweichung}(\varphi)$$

$M_{tol} := 50 N \cdot m$ erlaubte Abweichung des Moments von dem Bezugsmoment ist +-25Nm

Verdrehung in Grad bis zur erneuten Verspannung

$$\varphi_{zickzack} := \frac{M_{tol} \cdot l_{Stab} \cdot 180}{I_t \cdot G_{Stab} \cdot \pi} = 1.542$$

$$M_{zickzack}(\varphi) := \begin{cases} \left(M(\varphi) + \frac{M_{tol}}{2}\right) & \text{if } \varphi_{zickzack} > \varphi \\ \left(M(\varphi - \varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } \varphi_{zickzack} \le \varphi < 2 \cdot \varphi_{zickzack} \\ \left(M(\varphi - 2\varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } 2\varphi_{zickzack} \le \varphi < 3\varphi_{zickzack} \\ \left(M(\varphi - 3\varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } 3\varphi_{zickzack} \le \varphi < 4\varphi_{zickzack} \\ \left(M(\varphi - 4\varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } 4\varphi_{zickzack} \le \varphi < 5\varphi_{zickzack} \\ \left(M(\varphi - 5\varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } 5\varphi_{zickzack} \le \varphi < 6\varphi_{zickzack} \\ \left(M(\varphi - 6\varphi_{zickzack}) + \frac{M_{tol}}{2}\right) & \text{if } 6\varphi_{zickzack} \le \varphi < 7\varphi_{zickzack} \end{cases}$$



A.2. Attachment 2: Calculation of the losses of the bearings

Reibungsverluste der Lager von SKF Bearing Calculator

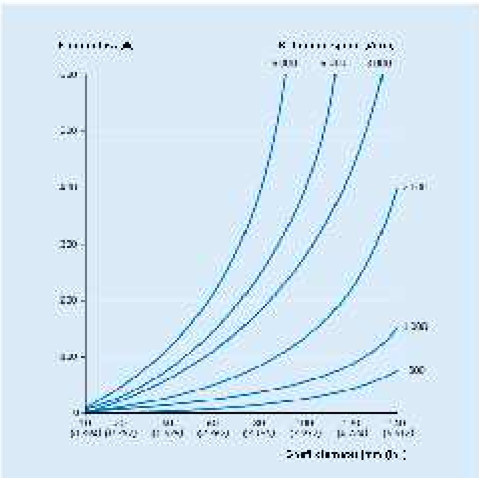
Bezeichnung	Anzahl	Radiallast	Querlast	Beölung	Ergebnis SKF bearing calculator[W]	Gesamt
SKF 32014	1	9025	1000		770	770
SKF 32012	1	9025	1000		580	580
SKF 32015	2	9025	1000		830	1660
SKF 32013	2	9025	1000		640	1280
SKF 32012	1	5000	1000		550	550
SKF 32011	1	5000	1000		500	500
SKF 22209EK	1	17786	0		360	360
SKF 32011	1	5000	1000		500	500
SKF 32011	1	5000	1000		500	500

Summe	6,7 kW
gerundet	7 kW

Reibverluste Dichtungen nach Diagramm von SKF

Bezeichnung	Anzahl	Durchmesser	Ergebnis SKF	Gesamt
CR 55x100x10 HMS	1	55	110	110
CR 85x115x12 HMS	1	85	210	210
CR 90x120x12_ HM	2	90	230	460
CR 85x105x12_ HM	2	85	210	420
CR 70x100x10_HM	1	70	140	140
CR 70x95x10_HMS	1	70	140	140
CR 70x95x10_HMS	1	70	140	140
CR 70x95x10_HMS	1	70	140	140

Summe	1,76 kW
gerundet	2 kW



A.3. Attachment 3: Uncertainty of the measurement TB40

Berechnung mit maximalen parasitären Lasten und +- 10 V Ausgang

		Faktor	plus/minus	bezogen auf	Max torque / FSO / Detle_M	Rechteckverteilungsfaktor	Normalverteilungsfaktor	Einzelbeträge der Messunsicherheit[Nm]
Kennwerttoleranz		1	0,100%	*max. Torque*	2000	0,577350269	nicht angwer	1,154700538
Linearitätsabweichung and hysteresis Spannungsausgang		1	0,030%	*FSO*	3000	0,577350269	nicht angwer	0,519615242
Rel. Standardabweichung der Wiederholbarkeit		1	0,030%	*FSO*	3000	0,577350269	nicht angwer	0,519615242
Temperatureinfluss pro 10k auf Ausgangssignal	hier 10°C	1	0,200%	*max. Torque*	2000	0,577350269	nicht angwer	2,309401077
Temperatureinfluss pro 10k auf Nullsignal bezogen auf Nennkennwert	hier 10°C	1	0,100%	*FSO*	3000	0,577350269	nicht angwer	1,732050808
parasitäre Fehler (0,3% X anteil davon)		1	0,183%	*FSO*	3000	0,577350269	nicht angwer	3,173811419
Wiederholbarkeit		1	0,030%	*Delta_M*	0	nicht angewendet	1	0
Langzeitdrift		1	0,030%	*max. Torque*	2000	0,577350269	nicht angewe	0,346410162

Gesamter Fehler (68,3% Vertrauenswahrscheinlichkeit)	4,52	*Nm*

Auftreender Fehler bei max. parasitären Lasten und+-10V Ausgang		
Gesamter Fehler (95,5% Wahrscheinlichkeit)	9,0332	*Nm*
% von max. Drehmoment	0,45%	

Berechnung mit minimalen parasitären Lasten und +- 10 V Ausgang

		Faktor	plus/ minus	bezogen auf	Max torque / FSO / Detle_M	Rechteckverteil- ungsfaktor	Normalverteil- ungsfaktor	Einzelbeträge der Messunsicherheit[Nm]
Kennwerttoleranz		1	0,100%	max. Torque	2000	0,577350269	nicht angwer	1,154700538
Linearitätsabweichung and hysteresis Spannungsausgang		1	0,030%	FSO	3000	0,577350269	nicht angwer	0,519615242
Rel. Standardabweichung der Wiederholbarkeit		1	0,030%	FSO	3000	0,577350269	nicht angwer	0,519615242
Temperatureinfluss pro 10k auf Ausgangssignal	hier 10°C	1	0,200%	max. Torque	2000	0,577350269	nicht angwer	2,309401077
Temperatureinfluss pro 10k auf Nullsignal bezogen auf Nennkennwert	hier 10°C	1	0,100%	FSO	3000	0,577350269	nicht angwer	1,732050808
parasitäre Fehler (0,3% X anteil davon)		1	0,007%	FSO	3000	0,577350269	nicht angwer	0,119738526
Wiederholbarkeit		1	0,030%	Delta_M	0	nicht angewendet	1	0
Langzeitdrift		1	0,030%	max. Torque	2000	0,577350269	nicht angwe	0,346410162

Gesamter Fehler (68,3% Vertrauenswahrscheinlichkeit) 3,22 Nm

Auftreender Fehler bei max. parasitären Lasten und+-10V Ausgang		
Gesamter Fehler (95,5% Wahrscheinlichkeit)	6,4315	Nm
% von max. Drehmoment	0,32%	

Berechnung mit minimalen parasitären Lasten und Frequenzausgang

		Faktor	plus/minus	bezogen auf	Max torque / FSO / Detle_M	Rechteckverteil-ungsfaktor	Normalverteil-ungsfaktor	Einzelbeträge der Messunsicherheit[Nm]
Kennwerttoleranz		1	0,100%	*max. Torque*	2000	0,577350269	nicht angwen	1,154700538
Linearitätsabweichung and hysteresis Spannungsausgang		1	0,030%	*FSO*	3000	0,577350269	nicht angwen	0,519615242
Rel. Standardabweichung der Wiederholbarkeit		1	0,030%	*FSO*	3000	0,577350269	nicht angwen	0,519615242
Temperatureinfluss pro 10k auf Ausgangssignal	hier 10°C	1	0,050%	*max. Torque*	2000	0,577350269	nicht angwen	0,577350269
Temperatureinfluss pro 10k auf Nullsignal bezogen auf Nennkennwert	hier 10°C	1	0,050%	*FSO*	3000	0,577350269	nicht angwen	0,866025404
parasitäre Fehler (0,3% X anteil davon)		1	0,007%	*FSO*	3000	0,577350269	nicht angwen	0,119738526
Wiederholbarkeit		1	0,030%	*Delta_M*	0	nicht angewendet	1	0
Langzeitdrift		1	0,030%	*max. Torque*	2000	0,577350269	nicht angewe	0,346410162

Gesamter Fehler (68,3% Vertrauenswahrscheinlichkeit) 1,76 *Nm*

Auftreender Fehler bei max. parasitären Lasten und Frequenzausgang		
Gesamter Fehler (95,5% Wahrscheinlichkeit)	3,5163	*Nm*
% von max. Drehmoment	0,18%	

A.4. Attachment 4: Calculation of the gears

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Verspanngetriebe Auslegung
Verzahnungsauslegung

Angaben

Eingangsdrehzahl:	$n_e := 3000 \cdot \frac{1}{min}$
Eingangsmoment:	$M_e := 2000 \cdot N \cdot m$
Schrägungswinkel:	$\beta := 0°$
Eingriffswinkel:	$\alpha_n := 20°$
Betriebsfaktor:	$c_b := 1.2$
Nennmoment:	$M_N := c_b \cdot M_e = 2400 \cdot N \cdot m$
Übersetzung:	$i_{Soll} := 1.5$
Verzahnungsqualität:	$VZQ := 7$
Zähnezahl:	$z_1 := 31$
Modul:	$m_n := 4 \cdot mm$
Zahnradbreite:	$b_1 := 65 \cdot mm$
ges. Profilverschiebung:	$x_{ges} := 0$

Übersetzung

geählte Zähnezahl Großrad: $z_2 := 37$

neue Übersetzung: $i_1 := \frac{z_2}{z_1} = 1.194$

Übersetzungsabweichung: $\frac{i_{Soll} - i_1}{i_{Soll}} = 20.43 \cdot \%$

$i_{Soll} = 1.5 \pm 5 \cdot \%$ *Die Übersetzung ist im Tolerenzbereich.*

Profilverschiebungsfaktoren

laut Roloff / Matek: TB 21-6:

mittlere Zähnezahl: $z_m := \frac{z_1 + z_2}{2} = 34$

Profilverschiebung Ritzel: $x_1 := 0$

Profilverschiebung Großrad: $x_2 := x_{ges} - x_1 = 0$

Es tritt keine Profilverschiebung auf!

Verzahnungsgeometrie

Betriebseingriffwinkel: $inv\alpha_{tw} = 2 \cdot \frac{x_1 + x_2}{z_1 + z_2} \cdot \tan(\alpha_n) + inv\alpha_t$

$inv\alpha_{tw} = inv\alpha_t$

$\alpha_{tw} = \alpha_t$

Stirneingriffswinkel: $\alpha_t := \operatorname{atan}\left(\frac{\tan(\alpha_n)}{\cos(\beta)}\right) = 20 \cdot °$

$\alpha_{tw} := \alpha_t$ *da Profilverschiebungssumme* $x_{ges} = 0$

Kopfspiel (Richtwert): $c_z := 0.25 \cdot m_n = 1 \cdot mm$

Verzahnungsgeometrie Zahnrad 1: Ritzel

Teilkreisdurchmesser: $d_1 := z_1 \cdot m_n = 124 \cdot mm$

Betriebswälzkreisdurchmesser: $d_{w1} := d_1 \cdot \frac{\cos(\alpha_n)}{\cos(\alpha_{tw})} = 124 \cdot mm$

Kopfkreisdurchmesser: $d_{a1} := d_1 + 2 \cdot m_n + 2 \cdot x_1 \cdot m_n = 132 \cdot mm$

Fußkreisdurchmesser: $d_{f1} := d_1 - 2 \cdot m_n - 2 \cdot c_z + 2 \cdot x_1 \cdot m_n = 114 \cdot mm$

Grundkreisdurchmesser: $d_{b1} := d_1 \cdot \cos(\alpha_t) = 116.522 \cdot mm$

Verzahnungsgeometrie Zahnrad 2: Großrad

Teilkreisdurchmesser: $d_2 := z_2 \cdot \frac{m_n}{\cos(\beta)} = 148 \cdot mm$

Betriebswälzkreisdurchmesser: $d_{w2} := d_2 \cdot \frac{\cos(\alpha_n)}{\cos(\alpha_{tw})} = 148 \cdot mm$

Kopfkreisdurchmesser: $d_{a2} := d_2 + 2 \cdot m_n + 2 \cdot x_2 \cdot m_n = 156 \cdot mm$

Fußkreisdurchmesser: $d_{f2} := d_2 - 2 \cdot m_n - 2 \cdot c_z + 2 \cdot x_2 \cdot m_n = 138 \cdot mm$

Grundkreisdurchmesser: $d_{b2} := d_2 \cdot \cos(\alpha_t) = 139.075 \cdot mm$

Achsabstand: $a_1 := \frac{d_{w1} + d_{w2}}{2} = 136 \cdot mm$

Profilüberdeckung

$$\varepsilon_1 := \frac{\sqrt{\left(\frac{d_{a1}}{2}\right)^2 - \left(\frac{d_{b1}}{2}\right)^2}}{\pi \cdot m_n \cdot \cos(\alpha_t)} = 2.626$$

$$\varepsilon_2 := \frac{\sqrt{\left(\frac{d_{a2}}{2}\right)^2 - \left(\frac{d_{b2}}{2}\right)^2}}{\pi \cdot m_n \cdot \cos(\alpha_t)} = 2.992$$

$$\varepsilon_a := \frac{a_1 \cdot \sin(\alpha_{tw})}{\pi \cdot m_n \cdot \cos(\alpha_t)} = 3.939$$

$\varepsilon_\alpha := \varepsilon_1 + \varepsilon_2 - \varepsilon_a = 1.679$ $\qquad$ $\varepsilon_\alpha > 1.15$

Kontrolle auf Unterschnitt

Ritzel		***Großrad***	
Zähnezahl:	$z_1 = 31$	Zähnezahl:	$z_2 = 37$
Profilverschiebungsfaktor:	$x_1 = 0$	Profilverschiebungsfaktor:	$x_2 = 0$

Nach Prüfung Mit Abb(Z4) besteht weder Gefahr für Unterschnitt noch Spitzenbildung

Kontrolle Kopfspiel

min. Kopfspiel: $c_{min} := 0.12 \cdot m_n = 0.48 \cdot mm$

bestehendes Kopfspiel: $c_{ist} := a_1 - \frac{d_{a1} + d_{f2}}{2} = 1 \cdot mm$

$c_{ist} > c_{min}$ *Es besteht genügend Kopfspiel.*

Zahnfußspannung

Tangentialkraft am Ritzel: $F_t := \frac{2 \cdot M_N}{d_1} = 38709.677\,N$

Radialkraft amRitzel: $F_r := F_t \cdot \tan(\alpha_n) = 14089.17\,N$

Axialkraft am Ritzel: $F_a := F_t \cdot \tan(\beta) = 0\,N$

Zahnformfaktor: $Y_F := 2.57$

Lastanteilsfaktor: $Y_\varepsilon := \frac{1}{\varepsilon_\alpha} = 0.595$

Schrägungswinkelfaktor: $Y_\beta := 1 - \frac{\beta \cdot 180}{120 \cdot \pi} = 1$

Faktor für $K_{F\alpha}$ $\frac{F_t}{b_1} = 595.533 \cdot \frac{N}{mm}$

Zahnradqualität: $VZQ = 7$

Profilüberdeckung: $\varepsilon_\alpha = 1.679$

Stirnlastverteilfaktor: $K_{F\alpha} := 1$

Zahnfußspannung: $\sigma_F := \frac{F_t}{b_1 \cdot m_n} \cdot Y_F \cdot Y_\varepsilon \cdot Y_\beta \cdot K_{F\alpha} = 227.84 \cdot \frac{N}{mm^2}$

Ritzelwerkstoff: 16MnCr5
Einsatzgehärtet
max. Zahnfußfestigkeit: $\sigma_{Fl} := 460 \frac{N}{mm^2}$

Sicherheit gegen Zahnbruch:

$S_F := \frac{\sigma_{Fl}}{\sigma_F} = 2.019$

Hertz'sche Pressung

E-Modul:	$E_{16MnCr5} := 2.1 \cdot 10^5 \frac{N}{mm^2}$
Materialfaktor:	$Z_M := \sqrt{0.35 \cdot E_{16MnCr5}} = 271.109 \cdot \sqrt{\frac{N}{mm^2}}$
Flankenformfaktor:	$Z_H := \frac{1}{\cos(\alpha_t)} \cdot \sqrt{\frac{1}{\tan(\alpha_{tw})}} = 1.764$
Überdeckungsfaktor:	$Z_\varepsilon := \sqrt{\left(\frac{4 - \varepsilon_\alpha}{3}\right)} = 0.88$
Zähnezahlverhältnis:	$u_1 := \frac{z_2}{z_1} = 1.194$
Stirnlastverteilfaktor:	$K_{H\alpha} := \frac{1}{Z_\varepsilon} = 1.137$
Herz'sche Pressung:	$\sigma_{H1} := Z_M \cdot Z_H \cdot Z_\varepsilon \cdot \sqrt{\frac{u_1 + 1}{u_1} \cdot \frac{F_t}{b_1 \cdot d_1} \cdot K_{H\alpha}} = 1332.42 \cdot \frac{N}{mm^2}$
Ritzelwerkstoff: 16MnCr5 einsatzgehärtet max. Hertz'sche Pressung:	$\sigma_{HP} := 1630 \frac{N}{mm^2}$

Sicherheit gegen Hertz'sche Pressung

$$S_H := \frac{\sigma_{HP}}{\sigma_{H1}} = 1.223$$