# Torque-Vectoring mit Koppelgetrieben und reduzierten Koppelgetrieben in elektrischen Achsantrieben

# Torque Vectoring in electrical drives with coupled planetary gears and complex compound planetary gears

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Schwerpunktthema: Drive Systems – Torque Vectoring

**Kurzfassung**

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Elektrisch angetriebene Achsen in Pkw, die über Torque-Vectoring-Funktionalitäten verfügen, ermöglichen neben der Zusatzfunktion des elektrischen Allradantriebes eines Hybridfahrzeugs auch eine Erhöhung der Fahrdynamik.

Elektrisch angetriebene Achsen mit Torque-Vectoring-Funktion lassen sich durch zwei E-Maschinen, die jeweils ein Rad antreiben, darstellen. Auch heute in Serie befindliche Systeme, bei denen eine Lamellenkupplung oder -bremse durch gezielten Schlupf eine Differenzdrehzahl zwischen den Rädern erzeugt, und in denen prinzipiell ein Überlagerungsgetriebe zwischen den Rädern oder ein Rad und den Differentialsteg geschaltet ist, können zum Umverteilen von Antriebsmomenten in elektrischen Getrieben genutzt werden. Somit ist eine E-Maschine für die elektrische Traktion ausreichend. Wird anstelle eines Schaltelements ein elektrischer Aktor mit dem Überlagerungsgetriebe zusammengeschaltet, entfallen die systemimmanenten Verluste durch Kupplungsschlupf während der Regelung.

Die ZG - Zahnräder und Getriebe GmbH schlägt eine besonders günstige Getriebestruktur für ein elektrisch aktuiertes Torque-Vectoring-Getriebe vor, die es erlaubt mittels einer relativ kleinen (zweiten) E-Maschine („Steuer-E-Maschine“) hohe Differenzmomente zwischen den Rädern aufzubringen, und zwar unabhängig vom Antriebsmoment der Achse, das über die primäre Traktionsmaschine dargestellt wird. Für eine hohe Übersetzung auf die Steuer-E-Maschine, vor allem bei koaxialen Getriebeanordnungen, wird vorgeschlagen, das Differential des Fahrzeugs als Koppelgetriebe oder als reduziertes Koppelgetriebe auszuführen. Die derartige Mehrfachnutzung der Bauelemente des Differentials führt zu einer hohen Leistungsdichte bei gleichzeitig hoher Übersetzung.

Es wird nachfolgend die prinzipielle Funktionsweise des aus Koppelgetrieben bestehenden Differentials anhand eines Prototyps erläutert. Dieser wurde im Rahmen eines Forschungsprojekts mit den Projektpartnern Audi, Hör Technologie, FZG (Forschungsstelle für Zahnräder und Getriebebau) und der ZG GmbH aufgebaut und in Prüfstands- und Fahrversuchen analysiert.
Weiterhin wird ein Achsantrieb vorgeschlagen, bei dem die gesamte Differentialeinheit inklusive des Koppelgetriebes im Leistungsfluss zwischen der E-Maschine und der Achsübersetzung angeordnet ist. Dadurch ist eine besonders gewichts- und bauraumgünstige Integration der Differentialeinheit in den Rotor der Traktionsmaschine möglich. Anhand einer Grobkonstruktion wird das Konzept erläutert.

**Abstract**
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Electrically driven axles in cars equipped with torque vectoring functionality provide enhanced driving dynamics – in addition to the provision of a hybrid vehicle with AWD-system.
Electrical drives with torque vectoring function can be displayed through two electric machines, each driving one wheel. Also the systems, which are currently used in series production, and in which a multi-disk clutch or a multi-disk brake generates differential speed between the wheels through direct slippage, and in which generally a superimposing unit is switched between the wheels or between one wheel and the differential carrier, can be used for the redistribution of driving torques in electrical gearboxes. Thus, one e-machine is sufficient for electric traction. If an electric actor – instead of clutch elements - is interconnected with the superimposing unit, then the losses inherent to the system, caused by clutch traction during control, cease to exist.
The “ZG - Zahnräder und Getriebe GmbH” has proposed a particularly suitable gearbox structure for an electrically actuated torque vectoring gearbox which permits the occurrence of high levels of difference torque between the wheels by means of a quite small (second) superimposing electric machine; this takes place independently from the driving torque of the axle, which is represented via the primary traction machine. For a high ratio to the superimposing electric machine, in particular in coaxial gearbox layouts, it is recommended to execute the differential of the vehicle in the form of a complex compounded planetary gear. This multiple use of the differential’s components results in a high level of performance in combination with a high level of transfer ratio.
With the help of an assembled prototype, the presentation will explain the layout of a differential which is comprised on a complex compounded planetary gear. The prototype was de-

veloped during a research project, funded by “Bayern Innovativ” with the participating project partners Audi, Hör Technologie, FZG (“Gear Research Centre”) and of the ZG GmbH. Audi has successfully tested the electric drive at the test stand and during road tests.

What’s more the presentation also shows the potential of single-gear axle drives which have the entire differential unit, including the coupled gear, arranged in the power-flow between the e-machine and the axle drive ratio. This makes integration into the rotor of the traction machine particularly efficient with regard to weight and construction space. By means of a concept design the principles are explained.

# 1. Basic Information

## 1.1 Torque Vectoring

Torque Vectoring gearboxes enable individual torque distribution to the wheels of a drive axle. Through different peripheral forces at the wheels it is possible to generate yaw moment around the vehicle vertical axis, which enables direct influencing of driving dynamics and driving stability. Unlike the ESP system, the vehicle will not be slowed down by control intervention. In contrast to current ESP, Torque Vectoring gearboxes are capable to effectively prevent understeer and thus to enhance safety and dynamics of the vehicle.

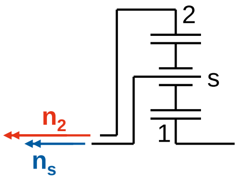
Systems currently used in series production create the required differential speed at the wheels through a superimposing unit and by enforcing fixed speed ratio on the closing of a multi-disk clutch or a multi-disk brake. Differential speeds at the wheels deviating from the fixed transmission ratio in the superimposing unit will be generated through slippage in the control element. This type of clutch-based systems shows drawbacks regarding efficiency, responding behaviour and controllability.

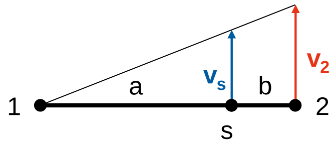
From [2], [3] and [4] systems have been presented, in which both control elements have been replaced by an electric machine. Generally, the wheels' differential speed onto the wheels is transmitted by means of a superimposing unit. In case of inactive system and straight-line drive (i.e. wheels turn with equal speed), the electric machine stands still so that no significant losses will occur. In the following, the basic principle of this gearbox structure will be explained. For analysis and synthesis of such gearboxes, a procedure according to Helfer [1] has turned out adequate. The procedure will be presented in brief in the following.

## 1.2 Planetary Gears – Analogue Model according to Helfer

In [1] Helfer has published an analogue procedure basing on the similarity of a planetary gear and a bar. It can be demonstrated in a generally accepted way that torque moments and speeds at the shafts of a planetary gear are in the same ratio toward each other like forces and speeds in the nodal points of a bar. For this, the distances between the nodal points of the bar must have a specific ratio which is exclusively characterised by the transmission in the planetary gear.

Fehler! Verweisquelle konnte nicht gefunden werden. shows the analogue procedure by means of an example: If the sun shaft is determined in the planetary gear according to Figure 1, a fixed transmission ratio between carrier shaft and ring gear shaft will occur. The same applies to the bar: If the displayed bar is deviated around node 1, peripheral speeds will occur at the nodes; these speeds match the speeds in the planetary gear as long as distances a and b between the nodes are set correspondingly.




$$n_1 - n_2 \cdot i_{12} - n_s \cdot (1 - i_{12}) = 0$$

$$n_1 = 0: \; -n_2 \cdot i_{12} = n_s \cdot (1 - i_{12})$$

$$\frac{n_2}{n_s} = \frac{1 - i_{12}}{-i_{12}}$$

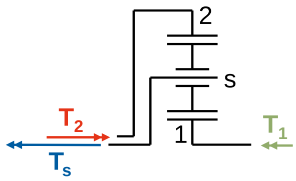
$$\Rightarrow$$

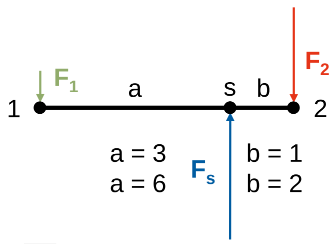
$$v_1 = 0: \quad v_2 = \frac{a+b}{a} \cdot v_s$$

$$\frac{v_2}{v_s} = \frac{\frac{a}{b} + 1}{\frac{a}{b}} = \frac{-i_{12} + 1}{-i_{12}}$$

**Figure 1: Engine Speed- / Speed Analogy according to Helfer [1]**

Also the torque forces developing at the shafts and the forces developing at the nodal points match proportionally if sections a and b have been set correctly. An example of this analogy is displayed in **Figure 2**.




$$T_1 + T_2 + T_s = 0$$

$$T_2 = -i_0 \cdot T_1$$

für $i_0 = -3$

$$T_2 = 3 \cdot T_1$$

$$\Rightarrow$$

$$\sum F = 0: \; F_1 + F_2 + F_s = 0$$

$$\sum M_s = 0: F_1 \cdot a - F_2 \cdot b = 0$$

$$F_2 = \frac{a}{b} \cdot F_1 \quad \left(\frac{a}{b} = -i_0\right)$$

$$F_2 = 3 \cdot F_1$$

**Figure 2: Torque- / Forces Analogy according to Helfer [1]**

This analogy cannot only be applied to individual elementary planetary gears, but also to gearbox systems consisting of several elementary gearboxes, which are connected arbitrarily.

### 1.3 Bar Model for a Differential Gear according to Helfer

In the following, the analogue procedure is applied to a bevel differential gear. **Figure 3 a)** shows a standard differential gear in bevel gear construction. As is known, the task of a differential gear is to distribute torque moments to the driving wheels in equal parts and independently from the speeds.

The respective bar is depicted in Figure 3 b). Both forces $T_{LR}$ and $T_{RR}$ are each half the size of force $T_{an}$ and are opposed to force $T_{an}$. The waves of the differential gear match the nodal points at the bar. In order to keep the bar in balance; that is bar is statically defined, it is required to allocate the same length to both bar sections (a).

It is interesting to know that this bar does not only represent the bevel differential gear, but also many other planetary gears. So, the spur differential gear according to Figure 3 c) does also match the statics and the kinematics of the bar according to Figure 3 b).

Basically it can be said that each planetary gear which can be represented by a bar according to Figure 3 b) can also be used as a differential gear.

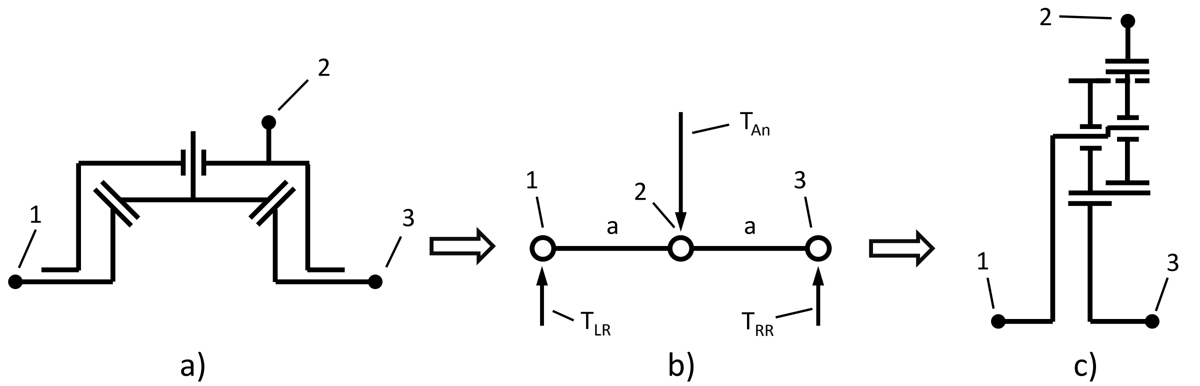



**Figure 3: Analogy for a Differential Gear according to Helfer [1]**

The statics and kinematics of a bar will remain unchanged when a further unstressed node is introduced. At the gearbox this corresponds to a fourth wave which is not connected towards the outside and thus does not lead in or lead away power. A corresponding bar can be seen in **Figure 4 a.** The forces in the bar's nodal points 1 and 3 (they correspond to the torque moments at the planetary gears) continue to be in a 1:1 ratio.

Bars with 4 nodes can be substituted in form of a circuitry with 2 bars and 3 nodes, each. Thus, the bar according to Figure 4 a) can be depicted by two bars according to Figure 4 b. The node in the middle "2" of the upper bar is connected with the outer node "2" of the bottom bar. The same applies to nodes "4". As a bar with 3 nodal points can be interpreted as an elementary planetary gear with 3 shafts, this splitting corresponds to the circuitry of two elementary planetary gears to one single differential gear.

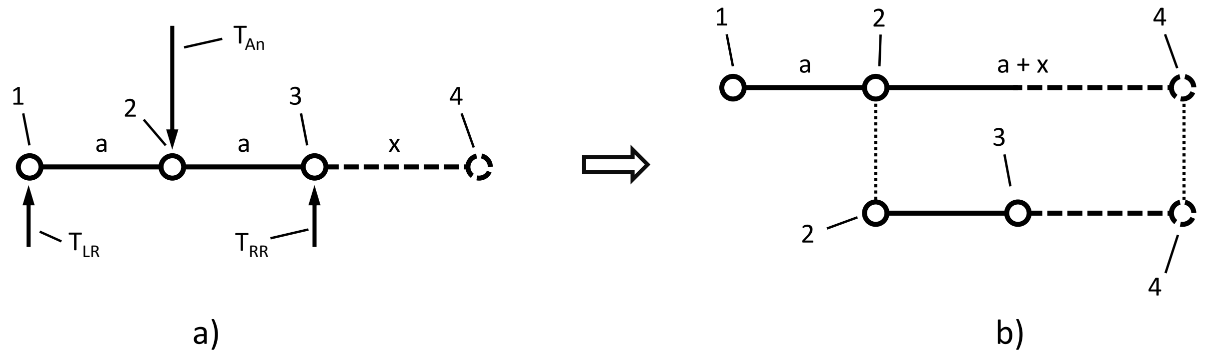



**Figure 4: Splitting of a Bar with 4 Nodal Points into 2 Bars with 3 Nodes, each**

**Figure 5** shows the selected bar circuitry faced by an equivalent coupled planetary gear. The stationary gear ratios $i_0 = -3$ or $i_0 = -2$ can be allocated to the bar lengths as defined in Figure 5 a).

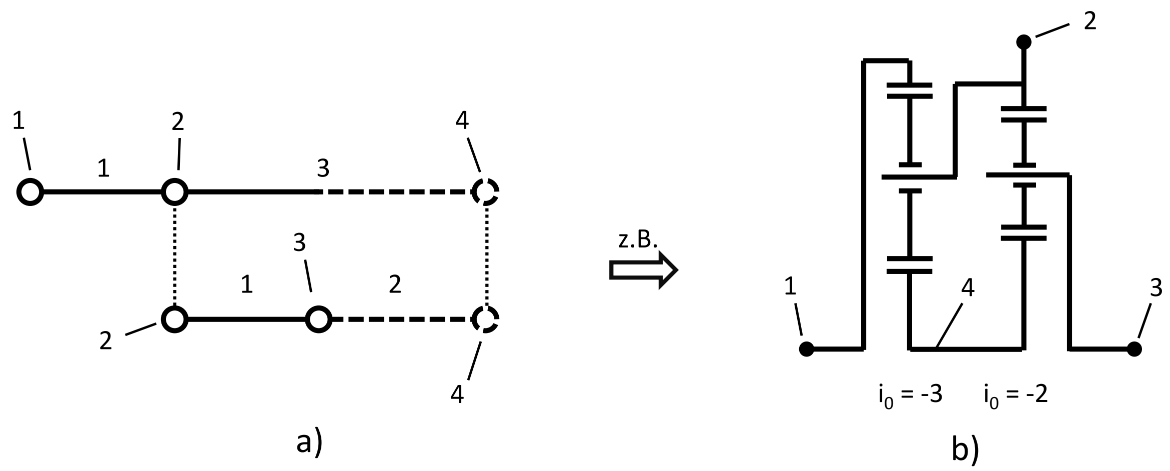



**Figure 5: Differential Gear with 4 Shafts – Derivation according to the Bars**

In the differential gear depicted in Figure 5 b) the drive is positioned at shaft 2 and the outputs to the wheels are at shaft 1 and 3. Although coupled shaft 4 is not connected toward the outside, it produces torque. When wheels revolve with same speed, the entire coupled gear rotates in the so-called coupling case; that is as a block.

At first, the effort required for the construction of a differential gear with two planetary gears is higher; however, there are specific advantages according to the respective circuitry. The depicted circuitry features power partition; that is driving power is partitioned onto two planetary gears. Thus, no element of the gearbox will be loaded with driving torque and therefore little specific stress arises for each planetary gear. Due to the decision to use 5 planets, a very compact design, above all in radial direction, can be achieved.

### 1.4 Bar Model for an active Differential Gear with Torque Vectoring

In the following – just as in [2] - an “active differential gear“ is considered a gearbox assembly, which - according to the needs - is able to generate different driving torque at the wheels. In case of deactivated Torque Vectoring, the gearbox assembly behaves like a “normal differential gear”. As described under [2], a superimposing unit is to be used according to the principle from **Figure 6**. The depicted circuitry of known structure connects two planetary gears via one common shaft (in this case a ring gear). One shaft (here: the sun of the left gearbox) is fixed relative to the housing and the corresponding shaft of the other gearbox is connected to an actuator (e.g. electric motor). Diametrically opposed torque will be produced at the remaining shafts (here: the carriers) as soon as torque occurs at the actuator. The respective differential torque exclusively depends on the actuator torque. It does not depend on the speeds of the shafts.

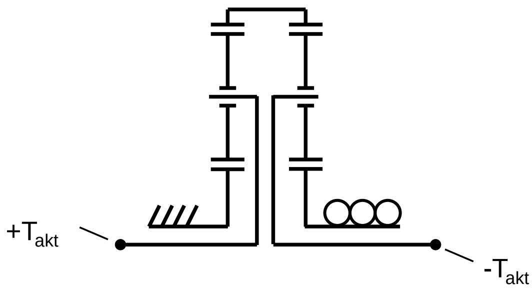
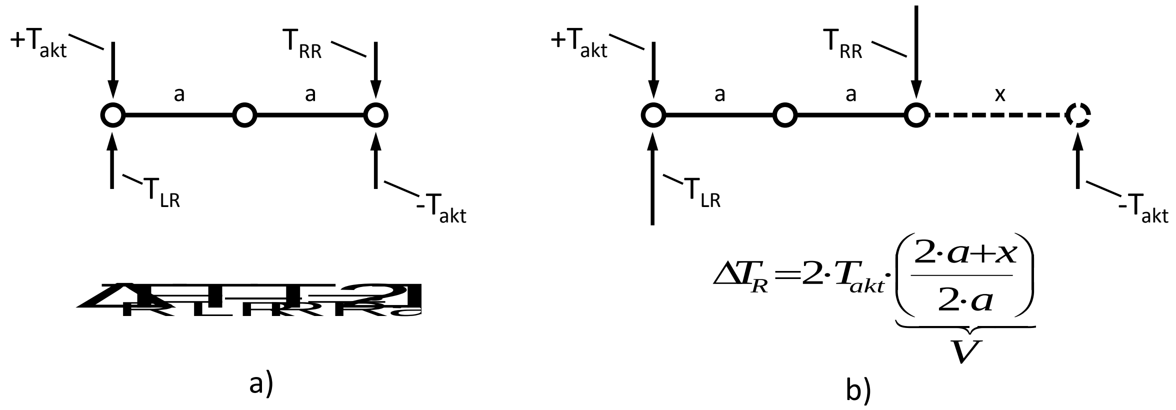



**Figure 6: Superimposing Unit**

The superimposing unit as depicted in Figure 6 and its output shafts can directly be connected to the wheel shafts so that diametrically opposed wheel torque will be generated at the wheels torque. If there is no driving torque at the differential gear, so the forces (or torque) in the schematic bar diagram will occur according to **Figure 7 a)**. The differential torque is defined as the difference of wheels torque and, in this case, it corresponds to double the actuator torque $T_{akt}$.

If a 4 shaft gearbox instead of a 3 shaft gearbox is now used as differential gear, just as shown in Figure 7 b), so the connecting shafts of the superimposing unit can be connected to those shafts of the differential gear which match the outer knots. The force couple at the bar created by the superimposing unit must be compensated by the wheel difference torque in order to meet bar statics. Owing to the lever conditions, higher wheel difference torque than in case a) develops. In the following, the increase through the respective lever lengths is determined as amplification factor V.



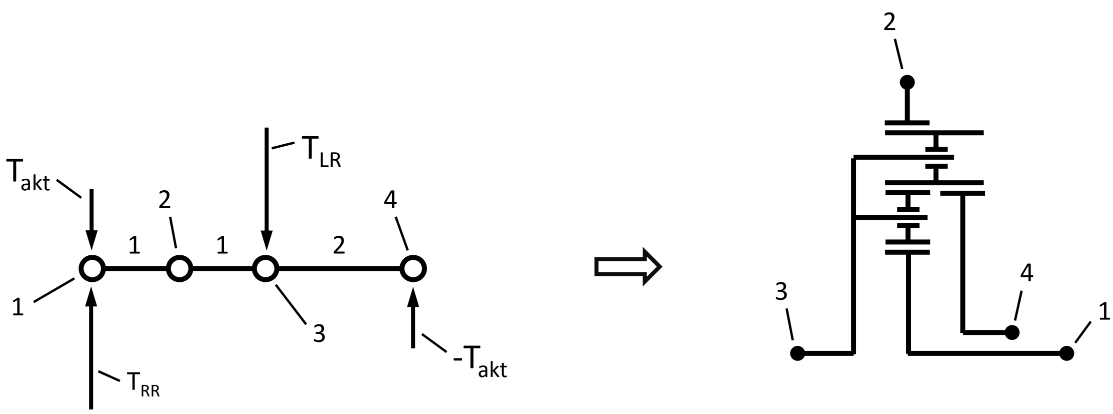

$$\Delta T_R = 2 \cdot T_{akt} \cdot \underbrace{\left( \frac{2 \cdot a + x}{2 \cdot a} \right)}_{V}$$

**Figure 7: Principle of active Differential Gears with Torque Vectoring**

A high amplification factor V reduces the actuator torque required for a specific wheel difference torque. As a rule, the target is to select a high amplification factor. For this, the section length x has to be set as long as possible.

#### 1.4.1 Active Differential Gear by means of Ravigneaux Planetary Gearset

One possibility to execute the differential gear as a 4 shaft gear is – next to the depicted coupled gear according to Figure 5 – the so-called Ravigneaux planetary gearset. **Figure 8** shows the structure and the corresponding bar. The drive of the differential gear is at ring gear 2, the wheel shafts are connected to shafts 1 and 3. The torque forces of the superimposing unit $T_{akt}$ is generated at shafts 1 and 4 and result in an amplification factor of V = 2 under the chosen transmission ratios. Compared to the circuitry according to Figure 7 a) it is possible to halve the actuator torque required for specific torque.

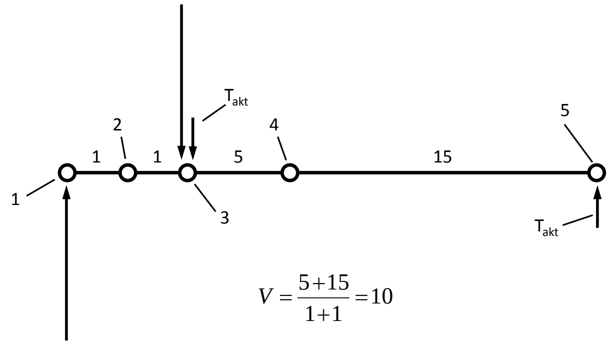



$$V = \frac{2 \cdot 1 + 2}{2 \cdot 1} = 2$$

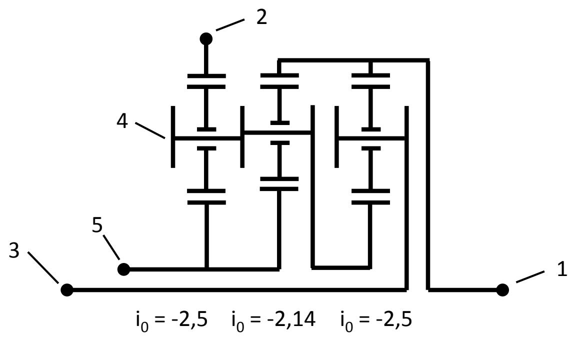
**Figure 8: Active Differential as Ravigneaux Planetary Gearset**

### 1.4.2 Active Differential Gear by means of Coupled Gear

Even higher amplification factors V can be achieved by correspondingly modifying the lever conditions at the bar. By means of a bar according to **Figure 9 a**, an amplification factor V = 10 can be achieved. The superimposition torque $T_{akt}$ is supported at the knots (or shafts) 3 and 5 and must correspondingly be compensated in the form of difference torque at the wheel shafts. Knot 4 is unstressed towards the outside, which corresponds to a shaft unconnected toward the outside. The bar according to Figure 9 can be depicted by the three elementary planetary gears as shown in Figure 9 b. Again, the gearbox rotates in the coupling case while the wheel shafts turn at same speed. All shafts of the gearbox are also subject to load in case of deactivated Torque Vectoring.




**a)**




**b)**

**Figure 9: Bar Plan for active Differential Gear with high Increase (not drawn to scale) and selected Coupled Gear for the Bar according to Figure 9a**

### 1.4.3 Gear Synthesis Programme “PlanGear“

The evaluation of gearbox circuitry which correspond to a given bar can be done considerably more efficient when computer-aided.

“PlanGear”, an in-house gear synthesis programme is mainly based on the above-explained analogue procedure according to Helfer [1]. It is possible to automatically evaluate gearbox circuitry from given bars. Circuitry will be checked for construction capability by means of graph theory.

In total “PlanGear” provides the following functions:

- Gear synthesis via combinatorics of elementary, coupled and complex compound planetary gears
- Assessment of construction capability by means of graph theory

- Calculation of operating data (torque moments, speeds, power flow) and gear load losses
- Taking into account of several drives and outputs (as depicted here)
- Synthesis of multispeed gearboxes and assessment of powershift selection

**3. Research Project: Drives with Torque Vectoring**

In a research project started in September 2010 and still going on at present, an electrical drive with Torque Vectoring function based on an active differential gear has been designed, developed, produced (3 prototypes) and tested at the test stand and in the vehicle. The body responsible for the project is the "Bayerische Staatsministerium für Wirtschaft, Infrastruktur, Verkehr und Technologie (Bavarian Ministry of Economic Affairs, Infrastructure, Transport and Technology)". Together with the "ZG - Zahnräder und Getriebe GmbH", the executing institutions have been the "Audi AG", the "Hör Technologie GmbH" and the "Forschungsstelle für Zahnräder und Getriebebau (FZG) der TU München/Gear Research Centre of the Technical University Munich".

The "ZG GmbH" was in charge of design and construction of the axle drive unit. The manufacturing of all required components was carried out by the "Hör Technologie GmbH". Assembly of the gear units took place at the "FZG" (additional task of the "FZG" is the optimisation of the gear teeth system for a further construction stage in the currently ongoing project.). Subsequently, the "Audi AG" has tested the axle at the test stand and during road tests.

### 3.1. Framework Conditions

The target vehicle of the prototype was an Audi A5 e-tron quattro [5] whose front axle is driven via hybrid drive. In addition to this, the axle drive constructed by “ZG” has been installed at the rear axle. **Table 1** shows the specifications of the gear unit.

| | | |
|---|---|---|
| Nominal Capacity – Drive (temporary) | kW | 60 |
| Maximum driving torque (temporary) | Nm | 300 |
| Maximum wheel difference torque | Nm | 1000 |
| Weight (weighed, including 1l oil) | kg | 85 |

**Table 1: Technical Specifications of the Electrical Drive**

### 3.2. Structure

The basic gear unit structure is depicted in **Figure 10**. The differential has been implemented in the form of a Ravigneaux planetary gearset according to chapter 1.4.1. One shaft of the superimposing gear according to Figure 6 is connected to the second sun of the Ravigneaux planetary gearset via a hollow shaft, the other shaft is connected to a wheel shaft.

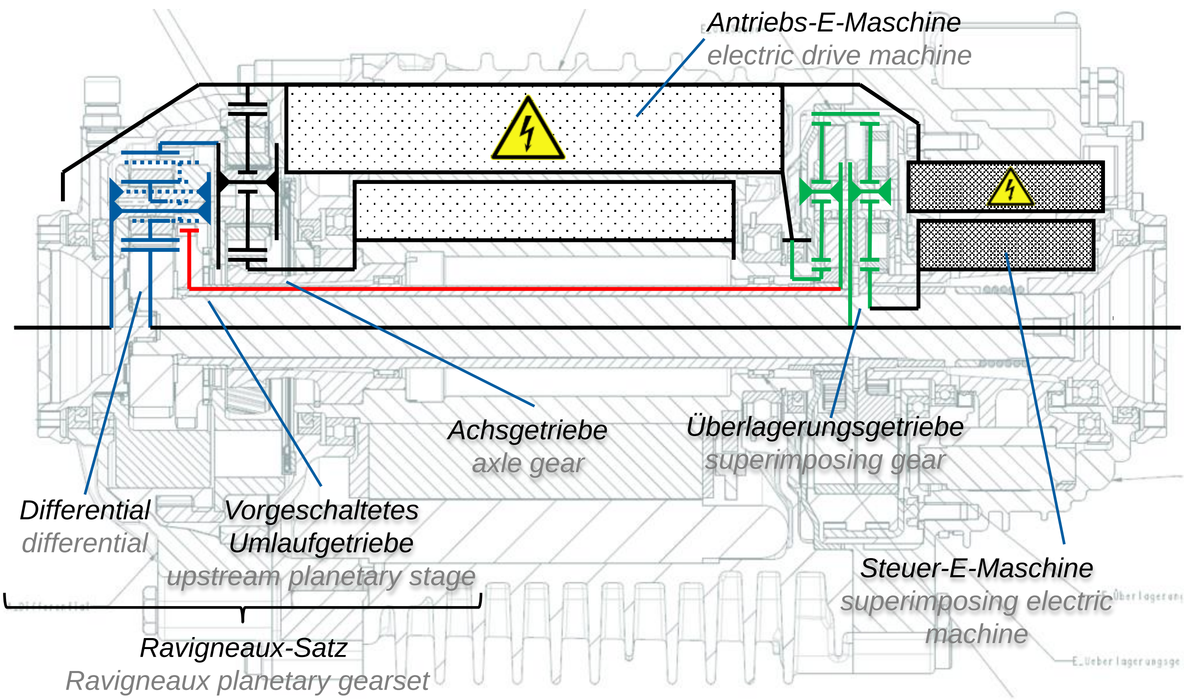



**Figure 10: Gear Unit Structure**

### 3.3. Control Concept and Responding Behaviour

The control concept used is depicted in **Figure 11**. The input requirements which are transferred to the vehicle dynamics controller are yaw rate, lateral and longitudinal acceleration, wheel speed and steering angle.

Using these input parameters and depending on the driving situation, the controller calculates the set value difference torque at the driving wheels of the rear axle. Resorting to this, the current required for power electronics can be defined via the characteristic curve of the drive which has been determined at the test rig before. Classification owing to different system behaviour of the components, like in clutch systems, is not necessary. Difference torque can be controlled and modified according to the needs.

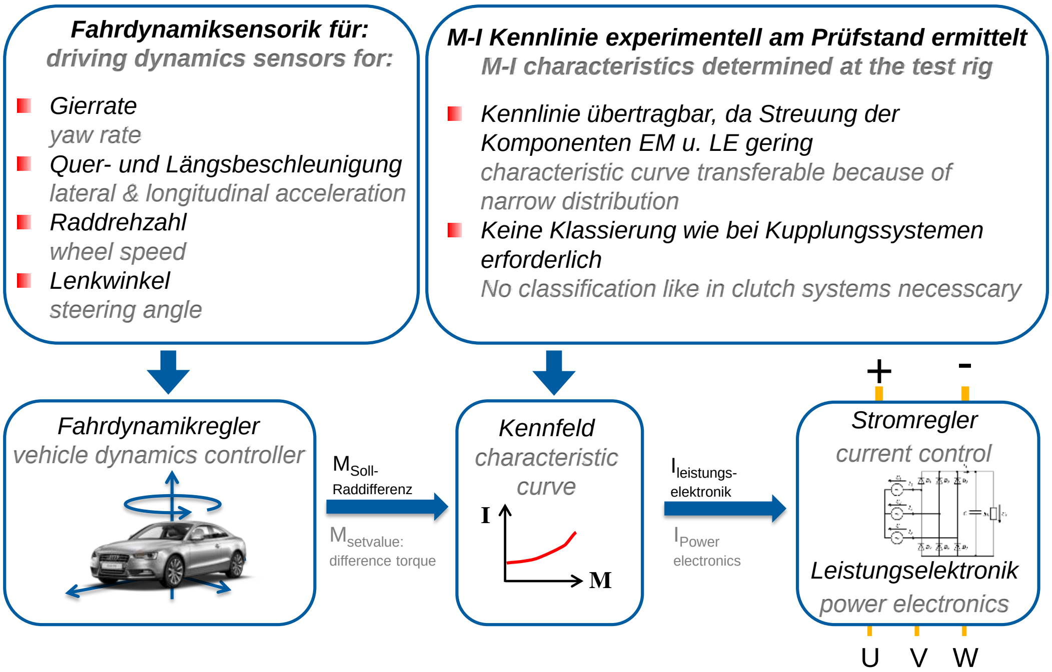



**Figure 11: Control Concept of Torque Vectoring Function**

The responding behaviour of the Torque Vectoring Function is illustrated in **Figure 12** by means of a torque leap. Wheel difference torque of $\Delta T_R$ = 850 Nm has been given. After a delay of 20 ms through the CAN bus, torque will be built up. After another 55 ms required maximum torque will be reached.

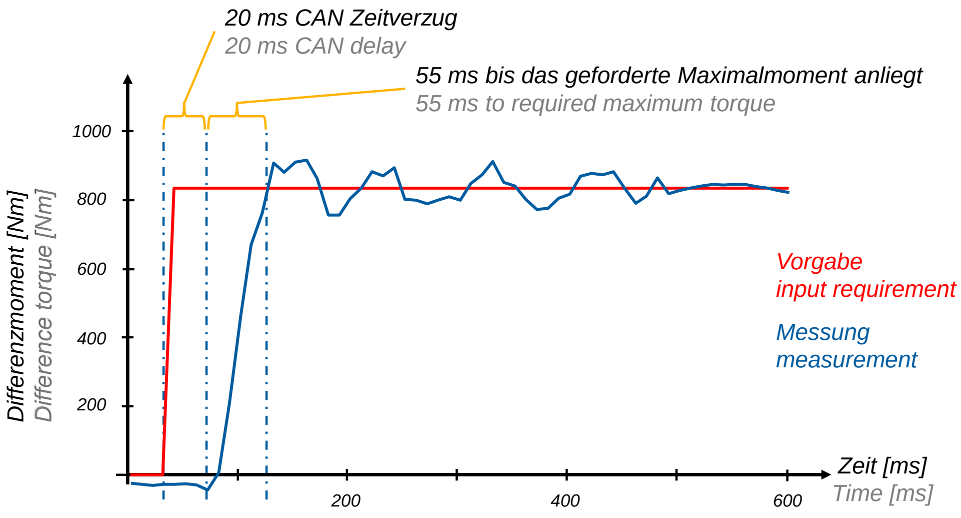



**Figure 12: Responding Behaviour of Torque Vectoring Function**

The control quality of the system is shown in **Figure 13**. Compared with clutch systems, the gear unit provides very high control accuracy. During initial application, however, system was susceptible to oscillation of mechanical components in case of high torque gradients set-values. Here, significant enhancement was attained by implementing the respective measures regarding control.

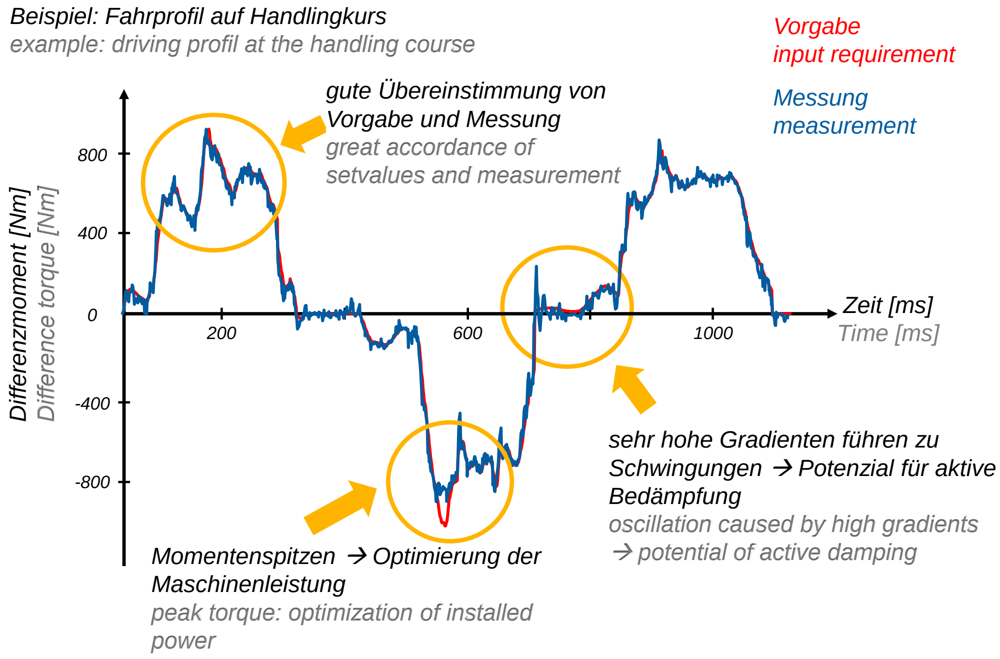



**Figure 13: Control Accuracy**

## 4. Drive Concept with High-Speed Differential

### 4.1 Target

Inherent to the design, electric motors in modern drives possess a relatively large rotor interior diameter which can be used for functional components. The target of the concept, which will be described in the following, is to integrate the differential within the rotor of the electric machine. Positioning of the differential in the power-flow between electric machine and axle drive ratio requires two axle drive ratio steps. As a result, the multiple-speed gear unit can only be realised under extreme effort so that the concept examinations are restricted to the one-speed gear unit.

### 4.2 Execution Example

The example shown in the following will examine one execution with Torque Vectoring and one execution without Torque Vectoring. If possible, the Torque Vectoring Function is supposed to be feasible in modular form with slight modifications at the gearbox.

#### 4.2.1 Overall Structure without Torque Vectoring Function

**Figure 14** displays the basic structure of the axle drive without Torque Vectoring. The differential has been integrated within the rotor in the form of a coupled planetary gear. Two identical axle drive ratios are arranged laterally to the electric machine. The drive of the axle gears is carried out via the sun, the output at the wheel end is carried out at the carrier under a ring gear fixed relative to the housing. By means of this circuitry, high transmission ratio at low constructional effort will be attained. In common gear outside diameters, axle drive ratios of up to approximately $i = 9$ are possible in this form of structure. Since the sun of the axle drive ratio does not have to be executed in the form of a hollow shaft, small sun diameter and therefore also small teeth numbers can be realised. This favours the acoustic behaviour of the gear through lower tooth meshing frequencies as compared to those coaxial concepts with axle gear following-up the axle gear.

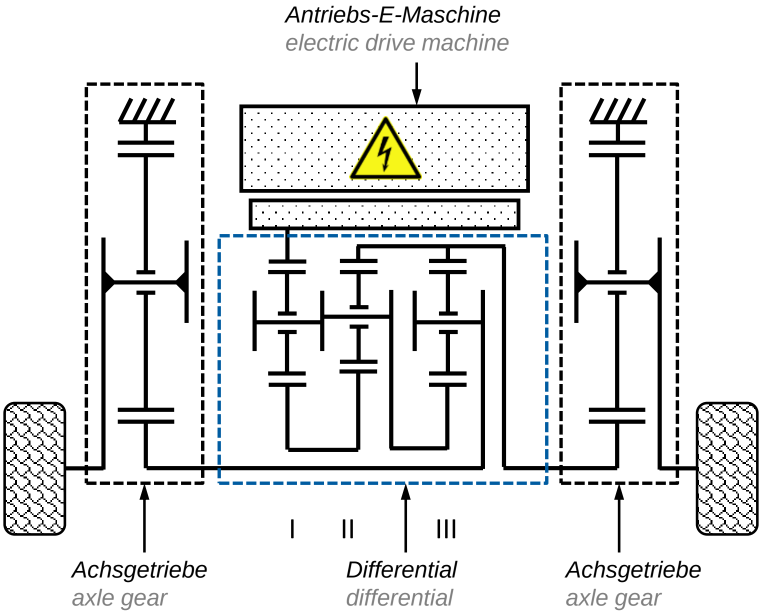



**Figure 14: Structure of Axle Gear with High-Speed Differential without Torque Vectoring**

The differential of the gearbox corresponds to the concept described in chapter 1.4.2. The selected stationary gear ratios of the planetary gears enable a structure with suns executed in the form of hollow shafts, and sufficiently large planet diameters for optional rolling bearing.

### 4.2.2 Gear Structure with Torque Vectoring Function

The structure of a gear with Torque Vectoring is shown in **Figure 15**. The superimposing unit with the 2$^{nd}$ electric machine serving as actuator for the generation of wheel difference torque is integrated additionally between an axle ratio and the electric machine. The suns of the superimposing unit will be connected to an output shaft as well as to a free shaft of the differential gear. The functionality of the superimposing unit matches with the description as described in chapter 1.4; however, the structure has been modified. The coupled gear of the differential already produces a sufficiently high amplification factor, and thus, it is possible to execute the superimposing gear the way it has been displayed. On the one hand torque moments $T_{akt}$ at the connecting shafts will be lower as compared to the circuitry according to Figure 6; on the other hand, however, in total the required transmission ratio from the electric machine torque to the difference torque can be achieved. The benefits of this structure are the simple construction method with only one carrier and the speeds of the planetary roller bearings, which do not exceed the range common in practice even under high-speed of the electric machine.

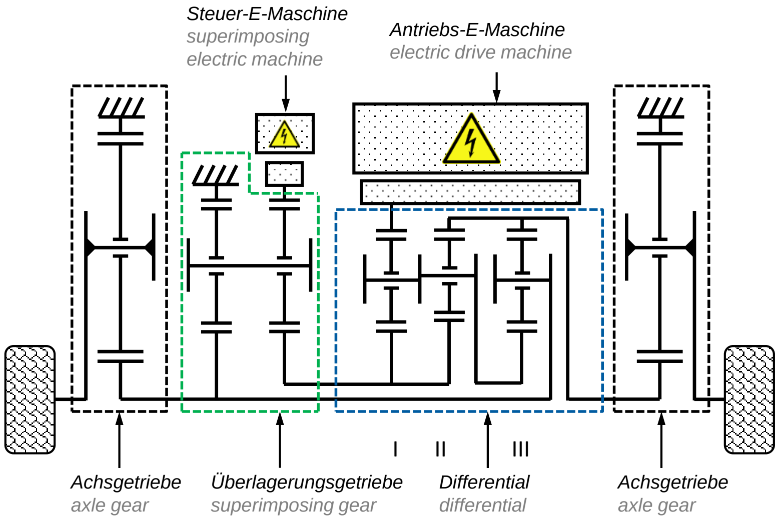



**Figure 15: Structure of Axle Gear with High-Speed Differential with Torque Vectoring**

Torque forces of the gear when operated with active Torque Vectoring are displayed in **Figure 16**. Torque at the electric machine has been selected adequately in order to ensure that wheel difference torque of 1000 Nm will develop.

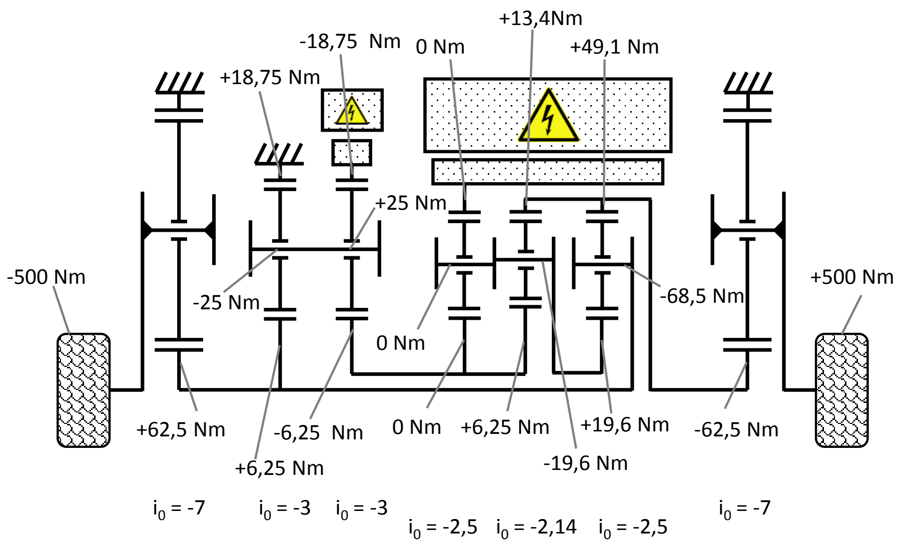



**Figure 16: Torque at the Components under active Torque Vectoring**

### 4.2.4 Structure of Axle Drive

**Figure 17 a** displays the sectional view of a gear with Torque Vectoring. With regard to construction space, the electric machine and the superimposing gear can be integrated into the gear very efficiently.

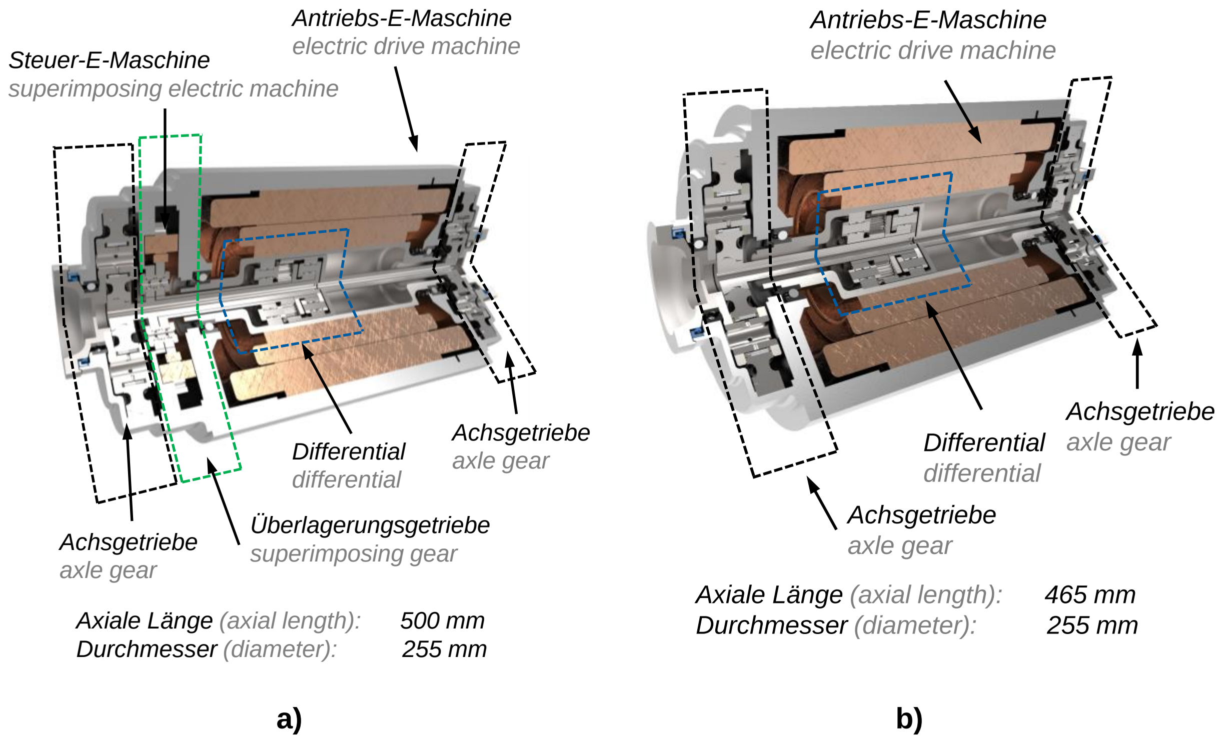



**Figure 17: Sectional View of Axle Drive with and without Torque Vectoring**

In order to refit the gear system with a drive without Torque Vectoring Function, it is required to remove the superimposing unit and to replace those shafts of the gear which are connected to the Torque Vectoring Unit. Figure 17 b depicts the result; that is the same electrical drive without Torque Vectoring.

## 5. Summary

Since its foundation in 2008 the "ZG GmbH" has intensively been focussing on Torque Vectoring gearboxes which are equipped with a so-called "electric machine" for the generation of wheel difference torque. In order to generate wheel difference torque, the electric machine requires approximately 5 kW as a maximum and it is relatively small compared to the traction electric machine. Basically, due to the structure of the gears, exclusively wheel difference speeds are transmitted to the electric machine and the respective torque is proportional to the wheel difference torque. In order to realise the highest possible transmission ratio between electric machine and wheel difference torque, the "ZG GmbH" recommends coupled or complex compound planetary gears to be used as differential.

The functionality of the differential, particularly with regard to its amplifying effect, has been explained by means of the analogue procedure according to Helfer. With the help of this procedure it is very easy to comprehend the kinematic and static interrelations of complex

planetary gears. By means of a given “Analogue Bar”, gear synthesis is possible. This process is part of the gear synthesis programme of the “ZG GmbH”.
Together with the project partners “Audi AG”, “Hör Technologie GmbH” and the “Forschungsstelle für Zahnräder und Getriebebau (TU München) /Gear Research Centre (Technical University Munich)” an electric drive system with Torque Vectoring Function was developed. The characteristics of the axle drive were tested at the test rig of the “Audi AG”. It turned out that the Torque Vectoring system was able to meet the expected potentials: regarding control dynamics and controllability it performed superior to clutch-based solutions. Since no losses occur in open control elements, no relevant disadvantages compared to axle drives without Torque Vectoring turn out with regard to efficiency behaviour.
For efficient use of the construction space within the rotor of the traction machine, another concept for an electric drive with Torque Vectoring is proposed. With the help of the synthesis programme, a gear structure has been determined which – on the one hand – enables differential function and – on the other hand – high transmission ratio between the electric machine and the wheel difference torque. This results in a relatively small electric machine and a very compact gear structure. Further benefits are smaller tooth meshing frequencies in the axle drive ratio and thus the enhanced excitation behaviour as well as the possibility to easily derive a gear derivative without Torque Vectoring.

## 6. References


[1] Helfer, F.: Eine Analogie zur Untersuchung von Planetengetrieben. ATZ 69 (1967), Nr. 5 S.149-152.

[2] Höhn, B.-R.; Wirth, Ch.; Kurth, F. Aktives Differential mit Torque-Vectoring-Funktion. Automobitechnisches Kolloquium 2009, TU München.

[3] Höhn, B.-R.; Stahl, K.; Lienkamp, M.; Wirth, Ch.; Kurth, F.; Wiesbeck, F.: Der elektromechanische Antriebsstrang mit Torque-Vectoring-Funktion des E-Fahrzeugs MUTE der TU München. VDI-Kongress: Getriebe in Fahrzeugen 2011, Friedrichshafen.

[4] Smetana, T.; Biermann, T.; Höhn, B.-R.; Kurth, F.; Wirth, Ch.: Schaefflers aktives Elektrodifferenzial. 9. Schaeffler Kolloquium (2010), S. 178-189.

[5] Schwarz, R.; Strasser, S.; Wein, M.; Maerkl, J.
Der elektrische quattro – Chancen und Herausforderungen
10. Grazer Allradkongress, Graz, 2011